Automotive Seat Vibration Control via Hydraulic Semiactive Vibration Absorbers (SAVA)

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The paper examines the ride performance enhancement that can be obtained by applying hydraulic semiactive vibration absorbers (SAVA) to alter the compliance characteristics of the seat/wheel suspension system. The work relies on a consistent model of the (nonlinear) hydraulics of the SAVA. A recently developed Lyapunov control scheme is used for regulation. The performance is examined assuming a quarter car with a seat/seat mounted mass. The paper then employs a quarter car/seat with a two mass ISO model of the seated human. Two road conditions are employed in the simulations; a ride swell and a road surface with a white noise velocity profile. The results show 45% reduction of of the vertical acceleration.

Key Words: Semiactive, Vibration Absorber, Ride Comfort, Modeling, Lyapunov Control

Nomenclature

$A_{P1,2}$	: Effective area of the actuator
$A_{V1,2}$	: Valve opening area
$A_{vmax1,2}$	: Maximum value opening area
$A_{vmin1,2}$	: Minimum value opening area
$C_{d1,2}$	: Discharge coefficient
c_a	: Damping coefficient of the seat
$c_{b,h}$	: Damping coefficient of the lower/upper mass
k_a	: Stiffness of the seat
$k_{b,h}$	: Stiffness of the lower/upper mass
k_s	: Stiffness of the sprung mass
k_u	: Stiffness of the unsprung mass
m_a	: Occupant/seat mass
$m_{b,h}$	: Lower/upper mass of a human
m_s	: Sprung mass
m_u	: Unsprung mass
$V_{1,2}$	: Volume of the actuator chamber 1,2
x_a	: Displacement of the seat
x_d	: Road input, displacement
$x_{s,u}$	: Displacement of the sprung/ unsprung mass
β	: Bulk modulus of the fluid
$\Delta P_{1,2}$	: Differential pressure of the actuator 1,2

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1. Introduction

Ride comfort is a key objective in the design of an automobile. Vibration in a vehicle can cause discomfort, annoyance, and even chronic health problems for automobile occupants. The principal interface between occupants and the vehicle is the automobile seat. It is becoming increasingly clear to automobile manufacturers that as the real cost of the auto rises relative to the earning power of the customer, there is an increased expectation of performance and comfort. Whole body vibration experienced during traveling in an auto is very often perceived as a source of discomfort. There are now extensive efforts by researchers to understand the nature of ride vibration and its effect on perceived comfort. Traditionally, auto makers have relied on the optimized passive design to achieve acceptable levels of ride comfort. But what was acceptable a decade ago is not now. There also is an emerging interest in reengineering chassis and seat suspensions in order to provide increased comfort.

It is generally held that if the transmissibility of the seat is reduced, then comfort is increased. Transmissibility is the amplitude rate of accelerations measured at different points on the anatomy, or the seat track. Increased ride isolation and reduced seat transmissibility can be accomplished by introducing automatically adjustable dampers into the load path between the seat and the road. This paper examines the effect on seat transmissibility when semiactive hydraulic dampers are integrated into either the seat suspension or the chassis suspension or at both points. The work relies on a linear model of the seats compliance and a linear two mass model of the seated human. Recent research by the authors has led to the development of a new and more complete characterization of the dynamics of semiactive dampers. (Patten, et al., 1998) The model which has been experimentally verified is used in the presented work.

2. Background

Passive, active and semiactive control system may be used as a seat damper between the seat and the body as well as a suspension between the body and the axle to isolate the vibration disturbance. (Cho and Yi, 1997a ; Ryba, 1993 ; Ryba, et al., 1993 ; Stein, 1995) It is noted that the sensibility of the human body to vibration is dependent on both amplitude and frequency. (Zhao, 1995) A great deal of research has demonstrated that semiactive suspensions can improve ride quality of an automobile if the hardware is capable of range of force generating appropriate with sufficient bandwidth. (Ivers and Miller, 1989 ; Wu, et al., 1993 ; Tseng and Hedrick, 1994) But there are very few articles discussing the use of semiactive dampers as actuators in a seat. He(1994) researched a means of reducing the transmission of the vibration to the human body by using viscoelastic(VE) material as the cushion suspension. His research and testing results showed that the peak value of the vertical vibration on the seated occupant could be reduced significantly by using a three-layer sandwich structure of VE material. Rakheja(Rakhaja et al., 1994) investigated the vibration attenuation performance of a behind-the-seat suspension structure based on Gouw seat suspension model. (Gouw, 1990) Patten(Patten and Pang, 1996) demonstrated the nonlinear automobile seat modeling with a sport car seat. The locally linear seat model utilized in this paper was obtained from his proposed nonlinear seat model. The seat modeling is essential to understand the characteristic of a seat and to design the seat vibration controller. The modeling of the human as a two mass system was suggested by Park(1976) and Wambold. (1986) Their human model was adopted in this work. Ryba(1993) presented an electronically controlled rotational damper which worked as a semiactive damping element for a sprung seat to improve poor isolation of truck vibrations. The sky-hook principle was used for his controller. Ryba(Ryba et al., 1993) also introduced a pneumatic fully active suspension for a sprung seat. Stein(1995) presented an electropneumatic active vibration control system for the driver seats of heavy earth-moving machines and trucks. Up to authors best knowledge, there has been no attempt to investigate the benefit to be obtained by integrating chassis and seat suspension controls.

3. Modeling

This section describes the basic seat/suspension models used in the analysis. Two configuration were considered. First a standard quarter car model is shown outfitted with a compliant linear seat(Fig. 1). A lumped mass representation of the human is used in that model. This first configuration is helpful when designing a controller. The system is shown equipped with linear springs and dampers representing the nominal compliance of the tire, chassis suspension and seat. The system is also shown outfitted with a semiactive hydraulic damper mounted both in the rattle space and between the sprung mass and the seat. The work here will examine the performance of the system

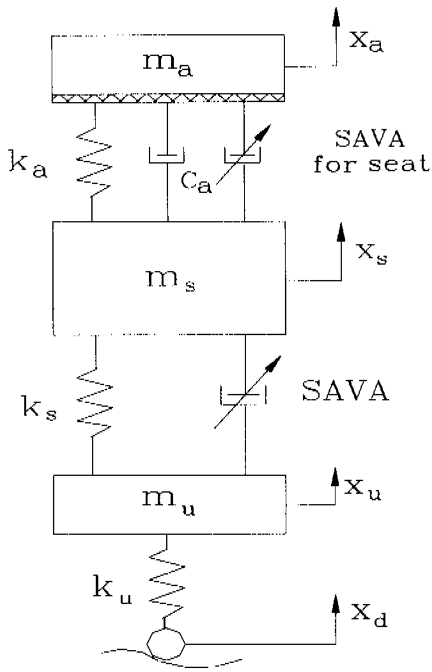

Fig. 1 A three DOF seat/suspension system with lumped human mass.

with a) both SAVA mounted, b) the seat SAVA mounted and the chassis suspension SAVA demounted c) the seat SAVA demounted and the chassis suspension SAVA mounted, d) both SAVA demounted.

The equations of motion for the model shown in Fig. 1 are;

$$m_a\ddot{x}_a = -k_a(x_a - x_s) - c_a(\dot{x}_a - \dot{x}_s) - A_{p1}\Delta P_1 \quad (1)$$

$$m_s\ddot{x}_s = k_a(x_a - x_s) + c_a(\dot{x}_a - \dot{x}_s) - k_s(x_s - x_u) + A_{p1}\Delta P_1 - A_{p2}\Delta P_2 \quad (2)$$

$$m_u\ddot{x}_u = k_s(x_s - x_u) - k_u(x_u - x_d) + A_{p2}\Delta P_2 \quad (3)$$

The parameters employed in the study represent a mid sized automobile with $k_u = 300{,}000.0$N/m, $k_s = 25{,}000.0$N/m, $k_a = 23{,}424.8$N/m, $c_a = 350.0$Ns/m, $m_u = 60.0$kg, $m_s = 270.0$kg.

The model assumes that 70% of the weight of the rider is supported by the seat cushion, then $m_a = 54.5$kg. A discussion of an extended model of the seat/suspension/occupant is offered in a later section. The dynamics of the actuator are developed next.

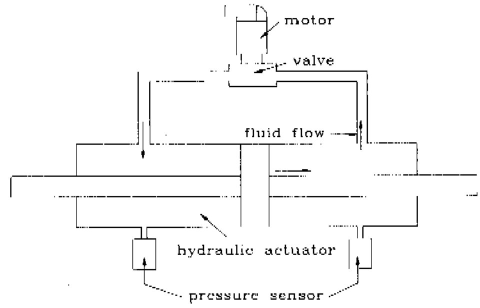

Fig. 2 Components of a hydraulic SAVA.

3.1 SAVA model

The essential components of a SAVA actuator are shown in Fig. 2.

The reduced order model of the hydraulic SAVA with the fixed bulk modulus β is given by (McCloy and Martin, 1980 ; Patten et al., 1996)

$$\Delta\dot{P}_i = \alpha_i A_{Pi} V_{reli} - \alpha_i C_{di} A_{vi} \cdot \mathrm{sing}(\Delta P_i)\sqrt{\frac{2|\Delta P_i|}{\rho}} \quad (4)$$

where $\alpha_i = \dfrac{\beta_i(V_{i1} + V_{i2})}{V_{i1}V_{i2}}$ and V_{reli} is relative velocity across the actuator. The subscripts $i = 1, 2$ indicate respectively the seat and chassis suspension. The state space form of a three DOF (degree of freedom) quarter car model with lumped human mass shown in is

$$\dot{X} = AX + Bg(x)U + d \quad (5)$$

where $x = [x_a, x_s, x_u, \dot{x}_a, \dot{x}_s, \dot{x}_u, \Delta P_1, \Delta P_2]$, $u = [A_{v1}, A_{v2}]^T$

$$A = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ -\frac{k_a}{m_a} & \frac{k_a}{m_a} & 0 & -\frac{c_a}{m_a} & \frac{c_a}{m_a} & 0 & -\frac{A_{P1}}{m_a} & 0 \\ \frac{k_a}{m_s} & -\frac{(k_a+k_s)}{m_s} & \frac{k_s}{m_s} & \frac{c_s}{m_s} & -\frac{c_a}{m_s} & 0 & \frac{A_{p1}}{m_s} & -\frac{A_{P2}}{m_s} \\ 0 & \frac{k_s}{m_u} & -\frac{(k_s+k_u)}{m_u} & 0 & 0 & 0 & 0 & \frac{A_{P2}}{m_u} \\ 0 & 0 & 0 & \alpha_1 A_{P1} & -\alpha_1 A_{P1} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \alpha_2 A_{P2} & -\alpha_2 A_{P2} & 0 & 0 \end{bmatrix}$$

$$g(X) = \begin{bmatrix} \mathrm{sgn}(X_7)\sqrt{\frac{2|X_7|}{\rho}} & 0 \\ 0 & \mathrm{sgn}(X_8)\sqrt{\frac{2|X_8|}{\rho}} \end{bmatrix}$$

$$B=\begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ -\alpha_1 C_{d1} & 0 \\ 0 & -\alpha_2 C_{d2} \end{bmatrix} \quad d=\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{k_u}{m_u}x_d \\ 0 \\ 0 \end{bmatrix}$$

The design of a generic control law for the closed loop operation of the SAVA actuators is discussed next.

4. Controller Design

The nonlinear actuator dynamics in Eq. (4) make the design of an effective control law for the system problematic. Other goal is also to keep the tire force as close to the nominal condition as possible. Another goal is the reduction of rattle space deflection. A recent paper (Mo, et al., 1997) gives an in depth discussion of the design issues raised by the last two goals when a SAVA is used in the chassis. The presented work is focused on the first goal; ride isolation.

The challenge to the control designer is to select a valve orifice area A_V at each moment in time that affords the sought after performance. One can utilize feedback linearization to linearize the actuator dynamics and then construct a suboptimal LQR "clipped" optimal control law. (Wu, et al., 1993) But that approach fails to capitalize the potential advantage inherent in the make up of the SAVA. Recalling the actuator dynamics, it is clear (Patten, et al., 1998, 1996a) that the SAVA can both store energy (by virtue of the compressibility of the fluid) and dissipate energy. To treat the device as a strictly dissipative element is to disregard the significant possibilities that one is able to manage the release of that stored energy.

A Lyapunov approach (Alleyne and Hedrick, 1993 ; Leitmann, 1994) is used here to craft a suitable control law that not only achieves the global goal at improving the ride isolation, capitalizes but also inherent to the operation of a SAVA, the feature which can store and dissipate the energy at the same time. The approach, which has been used to successfully control hardware versions of the SAVA design, is an outgrowth of a well known technique used to provide regulation for saturating controllers. (Patten, et al., 1996a, 1996b, 1998)

The procedure is initiated by selecting a candidate function or functional that is a mapping into a scalar. The following function is typically used;

$$V=\frac{1}{2}X^TQX,\ Q>0 \tag{6}$$

The matrix Q is an arbitrarily selected, positive definite penalty matrix. As will be demonstrated below, certain elements of Q are in fact control gains that can be tuned to achieve a desired performance criteria. The first time derivative of V is;

$$\dot{V}=\frac{1}{2}\dot{X}^TQX+\frac{1}{2}X^TQ\dot{X} \tag{7}$$

which in terms of the state space model developed above becomes;

$$\dot{V}=\frac{1}{2}X^T[A^TQ+QA]X+X^TQBg(X)u \\ +X^TQd \tag{8}$$

$\dot{V}$ is a dissipation function. The objective of the control design is to maximize the negativity of $\dot{V}$. Normally, if the eigenvalues of A are negative, a Q could be selected in a routine manner to assure that;

$$A^TQ+QA=P\leq 0 \tag{9}$$

The eigenvalues of A are all negative except one, which is 0, indicating at best Eq. (9) yield a semidefinite expression. This suggests that the search for a best Q would not produce any guarantee of asymptotic stability. It is also noted that the semidefiniteness of Eq. (9) confirms the fact that the system may have more than one equilibrium point, which is the case if the valve is closed permanently while the system is under load. the design then disregards the influence of the first term on the right of Eq. (8).

The second term on the right of Eq. (8) does provide a means of enhancing the rate of system energy dissipation. Noting the required structure of Q and the system dynamics, and writing Q as;

$Q=[\hat{q}_1,\hat{q}_2,\hat{q}_3,\hat{q}_4,\hat{q}_5,\hat{q}_6,\hat{q}_7,\hat{q}_8]$, $\dot{V}$ can take the following form;

$$\begin{aligned}\dot{V} = & -\frac{1}{2}X^T P X - \alpha_1 C_{d1}\sqrt{\frac{2|X_7|}{\rho}}X^T\hat{q}_7 \\ & \cdot \operatorname{sign}(X_7)A_{V1} - \alpha_2 C_{d2}\sqrt{\frac{2|X_8|}{\rho}}X^T\hat{q}_8 \\ & \cdot \operatorname{sign}(X_8)A_{V2} + X^T\hat{q}_6\frac{k_u}{m_u}x_d \end{aligned} \qquad (10)$$

An examination of the second and third terms reveals a control policy that provides the maximum negativity of the two terms;

$$\begin{cases} A_{V1}=A_{V1min} & \text{if } X^T\hat{q}_7\operatorname{sgn}(X_7)<0 \\ A_{V1}=A_{V1max} & \text{if } X^T\hat{q}_7\operatorname{sgn}(X_7)\geq 0 \end{cases} \qquad (11)$$

$$\begin{cases} A_{V2}=A_{V2min} & \text{if } X^T\hat{q}_8\operatorname{sgn}(X_8)<0 \\ A_{V2}=A_{V2max} & \text{if } X^T\hat{q}_8\operatorname{sgn}(X_8)\geq 0 \end{cases} \qquad (12)$$

The two vectors $\hat{q}_7$ and $\hat{q}_8$ are in fact control gains that can be tuned to provide the desired performance. The last term in Eq. (10) is a measure of the disturbance effect on the instantaneous change of $\dot{V}$. Noting that x_d is not known a priori, then there is nothing to be done to insure the negativity of the last term. The presented work relies on the selection of P, Q, $\hat{q}_7$ and $\hat{q}_8$ using a heuristic iterative procedure; P is specified first, Q is determined, and the performance using $\hat{q}_7$ and $\hat{q}_8$ is determined for a step input and a random input. The process (which is clearly indirect) is repeated until an acceptable performance is achieved. Once Q has been established, then the Q is used in all the subsequent simulations. It is interesting to note that it may be possible to select $\hat{q}_6$ on line in order to minimize the magnitude of $X^T\hat{q}_6$, which would, in turn, reduce the effect of the disturbance on the negativity of $\dot{V}$. That approach is now the subject of a research effort that is ongoing.

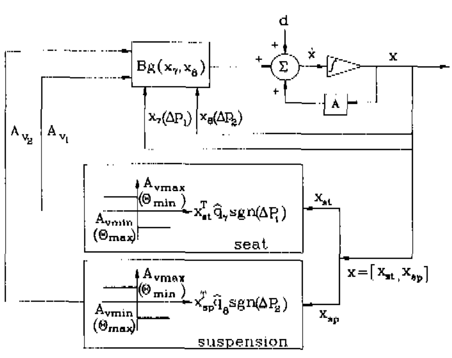

Fig. 3 Block diagram of a bistate controller.

The control law in Eqs. (11) and (12) define a bistate logic for the operation of the two valves. A block diagram of the feedback control system is shown in Fig. 3. The controller assumes the full state feedback. It is entirely feasible to cast the problem as an output feedback problem. In that case, an estimator, or reduced order estimator would then be required.

5. Simulation Results and Analysis

5.1 TEST 1: A seat/suspension system with a lumped human mass

In order to benchmark the performance of the design, a simulation of the three mass model given in Fig. 1 was conducted.

This approach will provide a clear indication of the effect the semiactive system might have on variation of transmissibility at the seat cushion. the tests provide comparisons of the performance for four basic test configurations;

1) Passive suspension/Passive seat
2) Passive suspension/SAVA seat
3) SAVA suspension/Passive seat
4) SAVA suspension/SAVA seat

In order to demonstrate the variation of performance with input, each of the four cases was examined first for a ride swell (a transient) and next for a road surface that was characterized by a random velocity profile.

Recent work by the authors has made it clear that any meaningful comparison of control strategies must first be preceded by a selection of SAVA static design parameters that produce an effective passive damper design if the valve is held

Table 1 SAVA Parameter values for a seat and a suspension.

Parameter	Quantity	
	Seat	Suspension
$A_{p1,2}(m^2)$	6.95e-4	1.2e-3
Stroke(m)	0.0381	0.0762
C_d	0.842	0.842
$A_{vmax1,2}(m^2)$	0.9048e-5	1.8096e-5

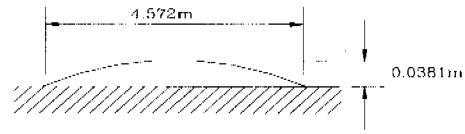

Fig. 4 profile of a ride swell used in simulation.

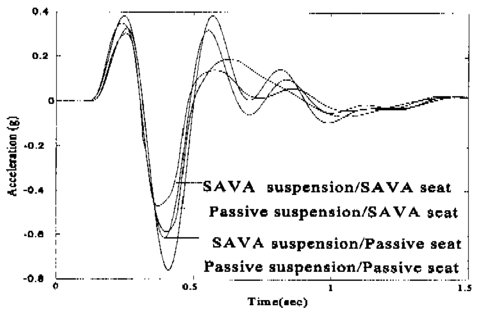

Fig. 5 Seat acceleration response to a ride swell with the vehicle traveling at 50Km/hr.

at a fixed position. The work here relies on a physical description of the SAVA system that provided approximately 22% of critical damping of the quarter car sprung mass model and the seat vibration model. Those physical parameters are listed in Table 1.

Figure 4 depicts the geometry of the ride swell used in the study. We assume that the vehicle is traveling at 50Km/hr.

The time domain response of the acceleration at the seat cushion is shown in Fig. 5. If the case 1 (Passive suspension/Passive seat) is used as a bench mark then one can compute the percentage reduction of peak amplitude as well as the RMS reduction of amplitude for each case. Those values are given in Table 2. The table indicates that the peak amplitude of the vertical acceleration is significantly reduced by 38% for the case 4, 23% for the case 2 and 19% for the case 3 comparing with the case 1. RMS comparison also indicates that the isolation performance is increased in the order of the cases 4, 3 and 2. The results indicate that SAVA seat can improve ride quality as much as SAVA chassis suspension.

Table 3 is used to compare the time response of the vertical seat relative displacement for the same ride swell disturbance. The relative seat vertical deflection is reduced by more than 20% for all the case 2, 3 and 4 comparing with the case 1. It is noted that the case 4 produces up to 42% reduction.

The next simulation employed a generic long road profile with random velocity character. The frequency component of the road profile is shown

Table 2 Reduction of seat acceleration amplitude response (Fig. 5).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	23%	18%
3. SAVA suspension /Passive seat	19%	20%
4. SAVA suspension /SAVA seat	38%	36%

Table 3 Reduction of seat vertical deflection (%) for the ride swell excitation (Fig. 6).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	26%	22%
3. SAVA suspension /Passive seat	20%	21%
4. SAVA suspension /SAVA seat	42%	41%

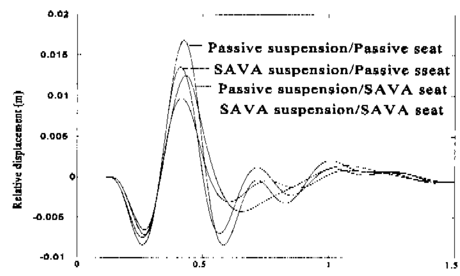

Fig. 6 Seat relative displacement response to a ride swell with the vehicle traveling at 50Km/hr.

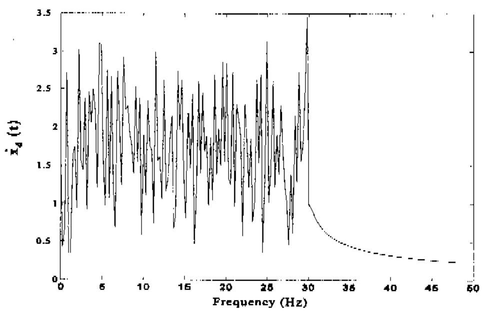

Fig. 7 FFT of the white noise velocity input.

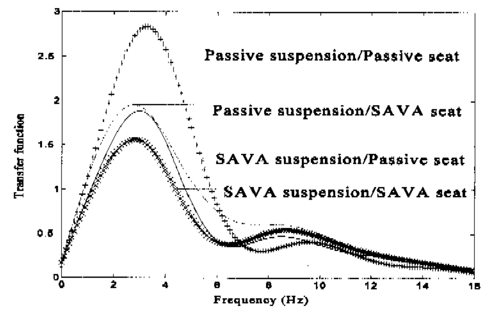

Fig. 8 Comparison of transfer functions for seat acceleration to a velocity input.

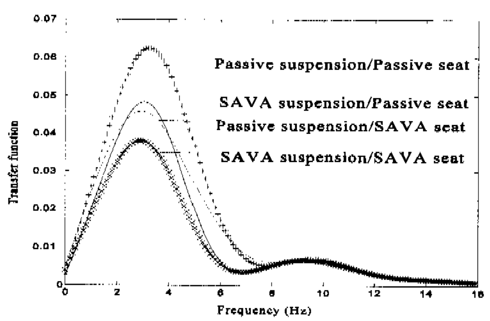

Fig. 9 Comparison of transfer functions for seat relative displacement to a velocity input.

in Fig. 7. Reduction rate of the acceleration and displacement with the random road profile are given in Table 4 and 5, respectively.

The results indicate that the SAVA design with the Lyapunov control is capable of producing twice as much isolation as a recent report has suggested to be possible for semiactive chassis suspension (Besinger, et al., 1995). The results here also make it clear that the reduction of acceleration transmitted to the seat butt is significant enough to be easily detected by a jury of untrained riders.

It is important to note that while the emphasis of the design is to reduce the transmissibility at the seat, the synthesis also seeks to maintain the

Table 4 Reduction of seat transmissibility (seat acceleration vs. road profile velocity)

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	31%	24%
3. SAVA suspension /Passive seat	33%	33%
4. SAVA suspension /SAVA seat	45%	41%

Table 5 Reduction of seat vertical displacement transmissibility.

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	27%	23%
3. SAVA suspension /Passive seat	23%	27%
4. SAVA suspension /SAVA seat	39%	41%

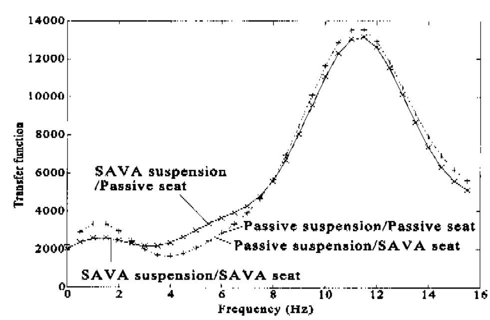

Fig. 10 Comparison of transfer functions for the variation of tire force to a velocity input.

degree of handling provided by the nominal (uncontrolled) design. The degree of handling is commonly gauged by measuring the variation of the tire force. Figure 10 depicts the variation of the tire force over the range of operating frequencies studied here. The effect of SAVA seat on the variation of the tire force is negligible compared with the SAVA chassis suspension.

The results clearly indicate that the proposed design does achieve the reduction of seat transmissibility while maintaining the handling performance.

5.2 TEST 2:Isolation using the ISO human model

Many researchers point out the possibility that the seated anatomy of a human can be described as a two mode system. While various models have been conjectured, the presented work explores the two mass human model for seat vertical vibration studies that was proposed by ISO.(1981) The model is shown in Fig. 11. The equation of motion of the seat/occupant model(Eq. (1)) can be replaced with;

$$m_h \ddot{x}_h = -k_h(x_h - x_a) - c_h(\dot{x}_h - \dot{x}_a) \qquad (13)$$
$$m_b \ddot{x}_b = -k_b(x_b - x_a) - c_b(\dot{x}_b - \dot{x}_a) \qquad (14)$$

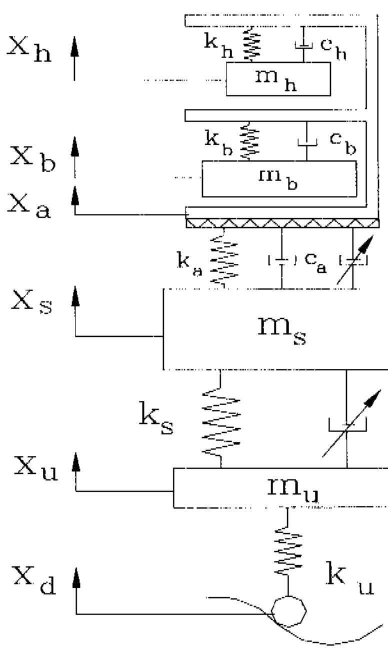

Fig. 11 A seat/suspension system with the ISO human model.

$$(c_h + c_b + c_a)\dot{x}_a = k_b x_b - (k_h + k_b + k_a)x_a + k_h x_h + c_b \dot{x}_b + k_h x_h + k_a x_s + c_a \dot{x}_s - A_{pl}\Delta P_1 \qquad (15)$$

Table 6 Reduction of seat acceleration amplitude response(Fig. 12).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	20%	15%
3. SAVA suspension /Passive seat	26%	23%
4. SAVA suspension /SAVA seat	40%	35%

Table 7 Reduction of seat acceleration amplitude response(Fig. 13).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	17%	16%
3. SAVA suspension /Passive seat	22%	22%
4. SAVA suspension /SAVA seat	36%	35%

Table 8 Reduction of seat transmissibility(Fig. 14).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	17%	11%
3. SAVA suspension /Passive seat	20%	18%
4. SAVA suspension /SAVA seat	32%	24%

Table 9 Reduction of seat transmissibility (Fig. 15).

Design case	Peak reduction	RMS reduction
1. Passive suspension /Passive seat	—	—
2. Passive suspension /SAVA seat	17%	15%
3. SAVA suspension /Passive seat	20%	16%
4. SAVA suspension /SAVA seat	33%	30%

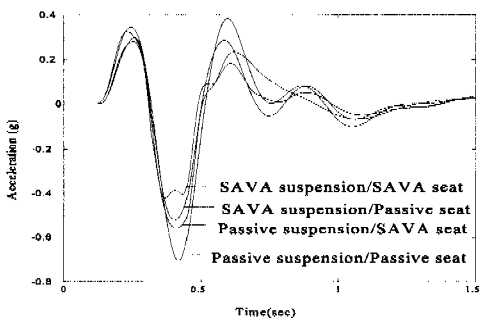

Fig. 12 Acceleration response of the upper mass to a ride swell with the vehicle traveling at 50 Km/hr.

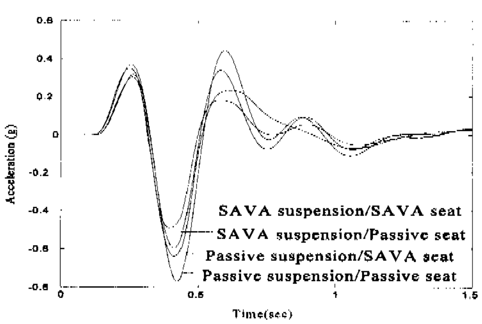

Fig. 13 Acceleration response of the lower mass to a ride swell with the vehicle traveling at 50 Km/hr.

Following the ISO standards, and assuming that the seated human weight 77kg, then the values of the anatomical parts used in the model were; $m_h=7$kg, $k_h=24{,}000$N/m, $c_h=190$Ns/m, $m_b=47$kg, $k_b=68{,}000$N/m, $c_b=1{,}540$Ns/m.

The presented work utilized the same two

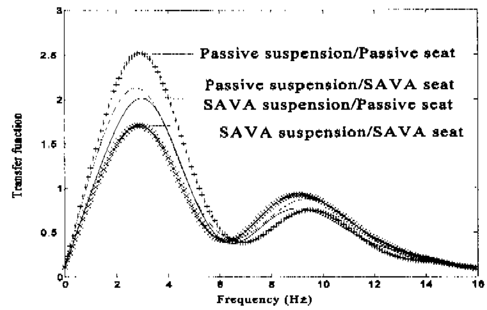

Fig. 14 Comparison of transfer functions for acceleration of the upper mass to a velocity input.

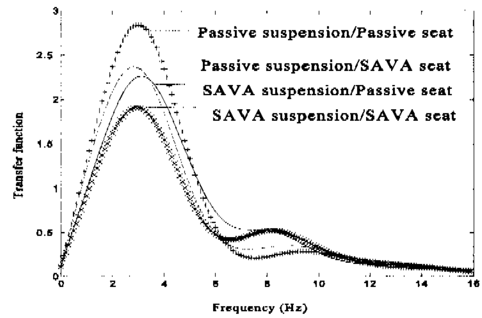

Fig. 15 Comparison of transfer functions for acceleration of the lower mass to a velocity input.

inputs used in the previous section to judge the effectiveness of the design. The results are listed in Tables 6, 7, 8 and 9. Figs. 12 and 13 depict the time domain response of the acceleration at the upper and the lower mass.

The transfer functions for the acceleration of the upper and the lower mass are shown in Figs. 14 and 15. As described in the TEST 1, the results in this simulation indicate that the case 4 provides the best ride comfort.

6. Conclusion and Discussion

The design analysis presented here makes it clear that a significant reduction of ride vibration in the critical area of spinal resonance (3-5Hz) is achieved by utilizing semiactive vibration absorbers both at the seat and at the chassis suspension. The design also offers attenuation at higher frequencies as well. The simulation of the control system in the time domain also indicates a significant increase in isolation. The analysis presented here relies on an extended experimental effort to validate the physical model used for the semiactive vibration absorber. The work also reflects an analytical and experimental effort to validate the performance of a SAVA chassis suspension design. The results demonstrate the theoretical feasibility of the design, which should be tested in a prototype configuration to prove its practicality.

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