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Automotive Seat Vibration Control with a Nonlinear Seat Cushion Model
비선형 시트 쿠션 모델을 고려한 자동차 시트의 진동 제어

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<요 약>

이 논문에서는 반능동 진동 흡수기를 통합 시트/섀시 현가장치에 확대, 적용하여 그 성능을 조사하였다. 통합 현가시스템의 성능분석을 위해 집중 인체질량과 함께 실험적으로 입증된 한 비선형 시트 쿠션 모델을 도입하였다. 또한 3 자유도 시트/섀시 현가시스템의 효과적인 진동제어를 위해 리아푸노브 바이스테이트 제어법칙을 사용하였다. 시뮬레이션결과 반능동 통합 현가장치는 시트 쿠션 모델과 관계없이 운전자의 승차감과 관련 있는 시트의 절대가속도 크기와 시트쿠션의 시트 트랙에 대한 상대변위를 상당히 감소시킬 수 있음을 알 수 있었다. 그러나, 주로 사용되어온 선형 쿠션 모델을 사용한 경우보다 비선형 쿠션 모델을 사용한 경우의 제진성능이 약간 저조함을 알 수 있었다. 따라서, 자동차 시트 설계시 성능분석을 위해서는 실제의(비선형의) 시트 쿠션 특성을 적용해야 함을 알 수 있다.

Key Words : Semiactive Vibration Absorber(SAVA), Ride Comfort, Nonlinear Seat Cushion Model, Lyapunov Bistate Control

1. INTRODUCTION

Ride comfort is one of main objectives in the design of an automobile as increases in expectation of performance and comfort when purchasing an auto. 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. Passive, active and semiactive control system may be used as a seat damper[1-3] between the seat and the body as well as a suspension[4-6] between the body and the axle to isolate the vibration disturbance. A great deal of research has demonstrated that semiactive suspensions can improve ride quality of an automobile, however, there are relatively few articles discussing the use of semiactive dampers as actuators in a seat to isolate the transmission of vibration to occupants. Ryba[7] 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. He also introduced a pneumatic fully active suspension for a sprung seat.[8] Stein[9] presented an electro-pneumatic

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active vibration control system for the driver seats of heavy earth-moving machines and trucks. Rakheja[10] investigated the vibration attenuation performance of a behind-the-seat suspension structure based on Gouw seat suspension model.[11]

This paper examines the effect on seat transmissibility when semiactive vibration absorbers (SAVA) are integrated into either the seat suspension or the chassis suspension or at both points. Mo[12] demonstrated the performance of the semiactive seat/chassis suspension system with a linear seat cushion model. But Pang, Patten,[13] and Mo[14] have shown that seat cushion dynamics exhibits significant nonlinearity. Results of a seat test they conducted are reprinted here to make that point. The seat testing was conducted on an electro-hydraulic motion simulator to determine the vibration characteristics of an automobile seat. Accelerations were measured at the seat track (input) and the interface between a sandbag and the seat surface (the seat butt). The input consisted of a white noise acceleration with an root mean square(RMS) average value of 0.1g to 0.4g in increments of 0.1g. The smoothed transfer functions for a sports car seat are shown in Fig. 1. The data indicates that as the seat track acceleration increases, both the

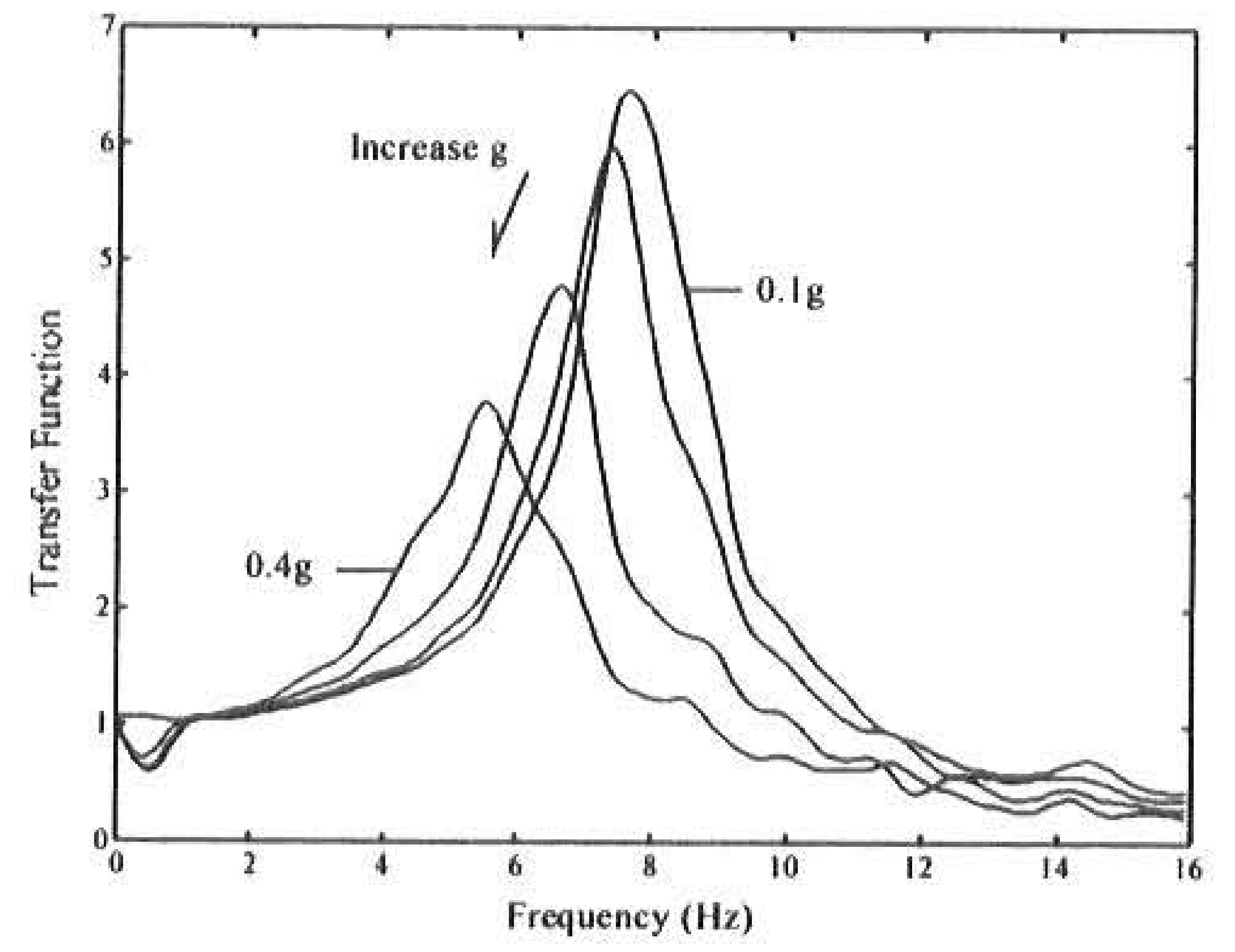

Fig.1 Measured transfer function between seat butt acceleration and seat track acceleration for a 37kg sandbag on a sports car seat cushion (Smoothed data).

peak amplitude of the transfer function and the resonance frequency decrease. In this work, an experimentally verified low order nonlinear seat cushion model was employed to analyze the effectiveness of the integrated semiactive suspension system. The performance is established in terms of acceleration transmissibility at the seat butt.

2. SYSTEM MODELING

2.1 SEAT CUSHION MODEL

The seat model includes linear stiffness and damping effects and a nonlinear dissipation term to account for Coulomb damping that arises when the gas in the open celled foam cushion flows through the matrix under load.[13-14] Fig. 2 depicts a lumped parameter model of an automobile seat cushion.

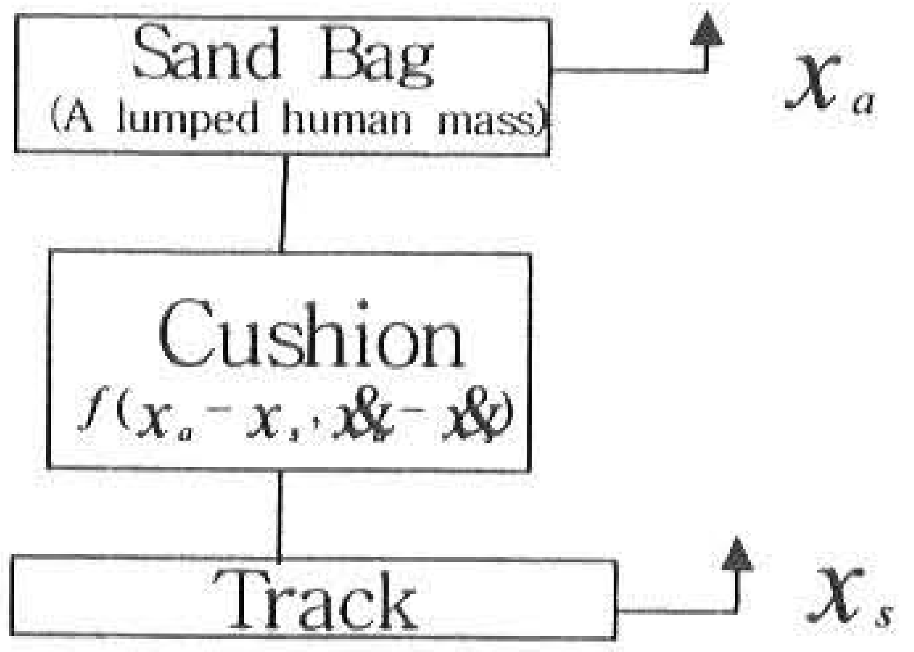

Fig. 2 Seat cushion/sandbag system

The function represents a generalized seat compliance.

The seat cushion model has the generic form;

$$M\ddot{x}_a = K(z)z + C(\dot{z})\dot{z} \tag{1}$$

where, $z(=x_a-x_s)$ is the relative displacement between the supported mass and the seat track, x_a is the inertial position of the mass center of gravity, and x_s is the inertial position of the seat track. The stiffness has the following form;

$$K(z) = K_0 + \frac{K_1}{1+K_2|z|} \tag{2}$$

Liber and Epstein[16] originally proposed the following damping characteristic;

$$C(\dot{z}) = C_0 + C_1|\dot{z}| \tag{3}$$

The empirically identified parameters[13-14] were K_0= 21,000N/m, K_1= 126,000N/m, K_2= 3333.33m^{-1}, C_0 = 230Ns/m, C_1 = 3,000kg/m, and M = 37kg.

A simulation was conducted to validate the proposed model. Fig. 3 depicts the simulated seatbutt/seat track vertical acceleration transfer functions of the nonlinear seat model of the sports car seat for the entire range of white noise acceleration inputs. A comparison of Fig. 1 with Fig. 3 indicates that the modeled and the measured transfer functions are in close agreement.

2.2 INTEGRATED SUSPENSION MODEL

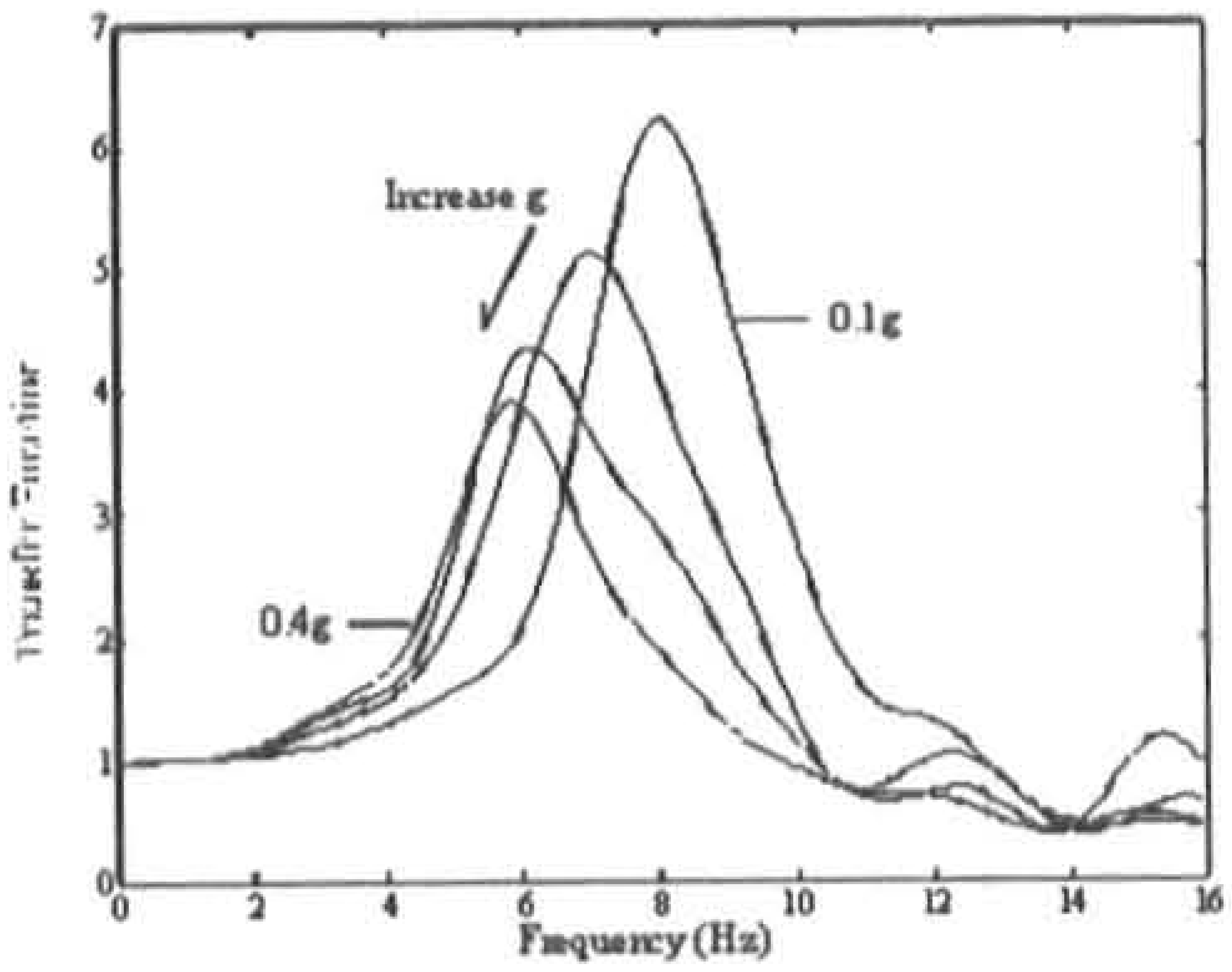

Fig. 3 Simulated transfer functions for the seat /sandbag(37kg) for white noise seat track accelerations of 0.1g to 0.4g RMS in increments of 0.1g (Smoothed data).

Introducing the nonlinear seat cushion model into the 3-DOF (degree of freedom) semiactive seat/chassis suspension shown in Fig. 4 equations of motion for the extended model of the occupant/seat/suspension are;

$$
\begin{aligned}
m_a\ddot{x}_a = &-(K_o+\frac{K_1}{1+K_2|x_a-x_s|})(x_a-x_s) \\
&-C_o(\dot{x}_a-\dot{x}_s)-C_1|\dot{x}_a-\dot{x}_s|(\dot{x}_a-\dot{x}_s) \\
&-A_{p1}\Delta P_1
\end{aligned} \tag{4}
$$

$$
\begin{aligned}
m_s\ddot{x}_s = &(K_0+\frac{K_1}{1+K_2|x_a-x_s|})(x_a-x_s) \\
&+C_0(\dot{x}_a-\dot{x}_s)-k_s(x_s-x_u)-c_s(\dot{x}_s-\dot{x}_u) \\
&+C_1|\dot{x}_a-\dot{x}_s|(\dot{x}_a-\dot{x}_s)+A_{p1}\Delta P_1-A_{p2}\Delta P_2
\end{aligned} \tag{5}
$$

$$m_u\ddot{x}_u = k_s(x_s-x_u)-k_u(x_u-x_d)+A_{p2}\Delta P_2 \tag{6}$$

where

$A_{P1,2}$: effective area of the actuator
k_a: stiffness of the seat
k_s: stiffness of the sprung mass
k_u: stiffness of the unsprung mass
m_a: occupant/seat mass
m_s: sprung mass
m_u: unsprung mass
x_a: displacement of the seat
x_d: road input, displacement
$x_{s,u}$: displacement of the sprung/unsprung mass
$\Delta P_{1,2}$: differential pressure of the actuator 1,2

The subscripts 1,2 indicate respectively the seat and chassis suspension. The essential components and the reduced order model of the hydraulic SAVA were described in Mo's work.[12]

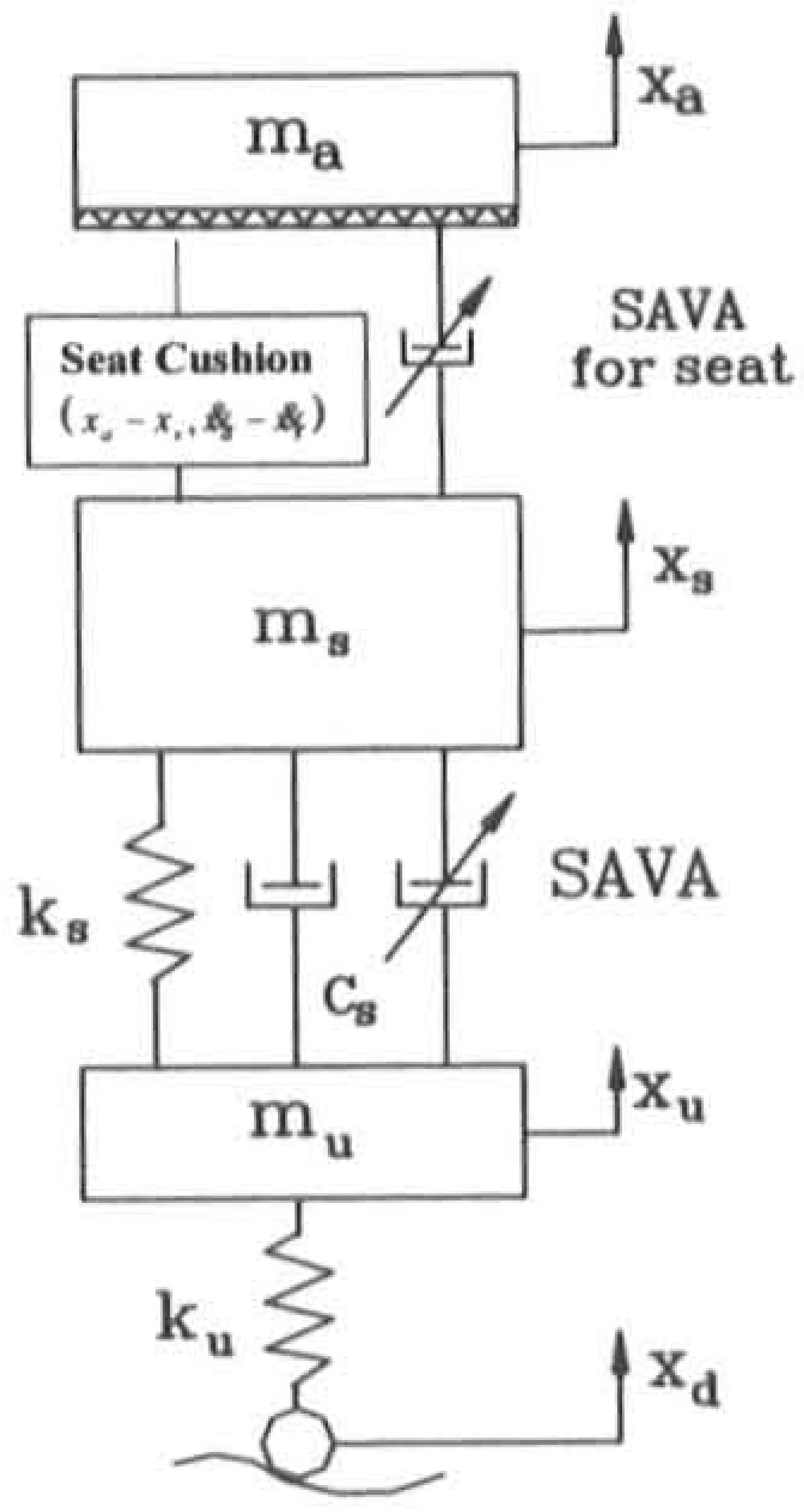

Fig. 4 A 3-DOF seat/chassis suspension with a lumped human mass.

A simulation was also conducted to determine the open loop response. A road surface with a white noise velocity was assumed. Fig. 5 depicts the response.

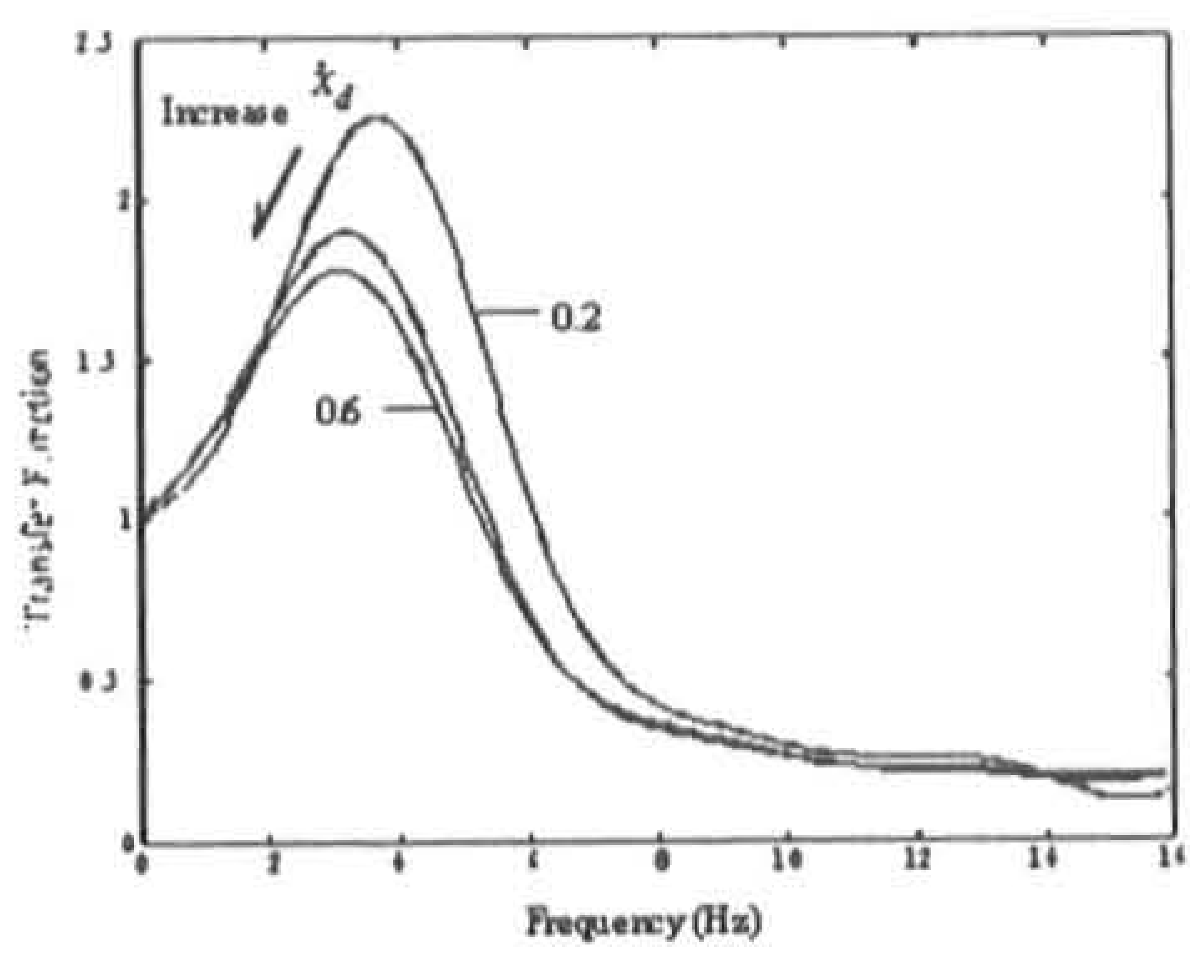

Fig. 5 Simulated transfer function of seat butt acceleration to the road input $\dot{x}_d$. $\dot{x}_d$ was assumed to be white noise with RMS amplitudes of 0.2, 0.4 and 0.6 (Smoothed data).

The results indicate that as the seat track acceleration increases, both the peak amplitude of the transfer function and the resonance frequency decrease.

3. SIMULATION RESULTS AND ANALYSIS

The simulation was next conducted to examine the effect of the SAVA on variation of transmissibility at seat cushion with the nonlinear characteristic. The tests provide comparisons of the performance for the four basic 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 and next for a road surface that was characterized with a random velocity.

The bistate controller which was developed when a SAVA was used in the chassis[17] as well as the seat/wheel suspension with a linear seat cushion model[12] was also employed to regulate each valve orifice area A_v. Those two articles[12) 17)] give an in depth discussion of the controller design issues.

The parameters for the 3-DOF quarter car model were m_s = 270kg, m_u= 60kg, k_s = 25,000N/m, c_s = 2,000N/m, k_u = 300,000N/m, m_a = 54kg, A_{p1}= 6.95e-4m^2, and A_{p2} = 1.2e-3m^2.

Fig. 6 depicts the seat vertical acceleration responses to a ride swell. The SAVA suspension /Passive seat system produces a 22% RMS reduction while the Passive suspension/SAVA seat system produces less than 10% RMS reduction. The SAVA suspension/SAVA seat system yields 25% RMS reduction. The RMS amplitude of the Passive suspension/Passive seat was reduced by 36% for SAVA suspension /SAVA seat system with a linear seat cushion model.[12]

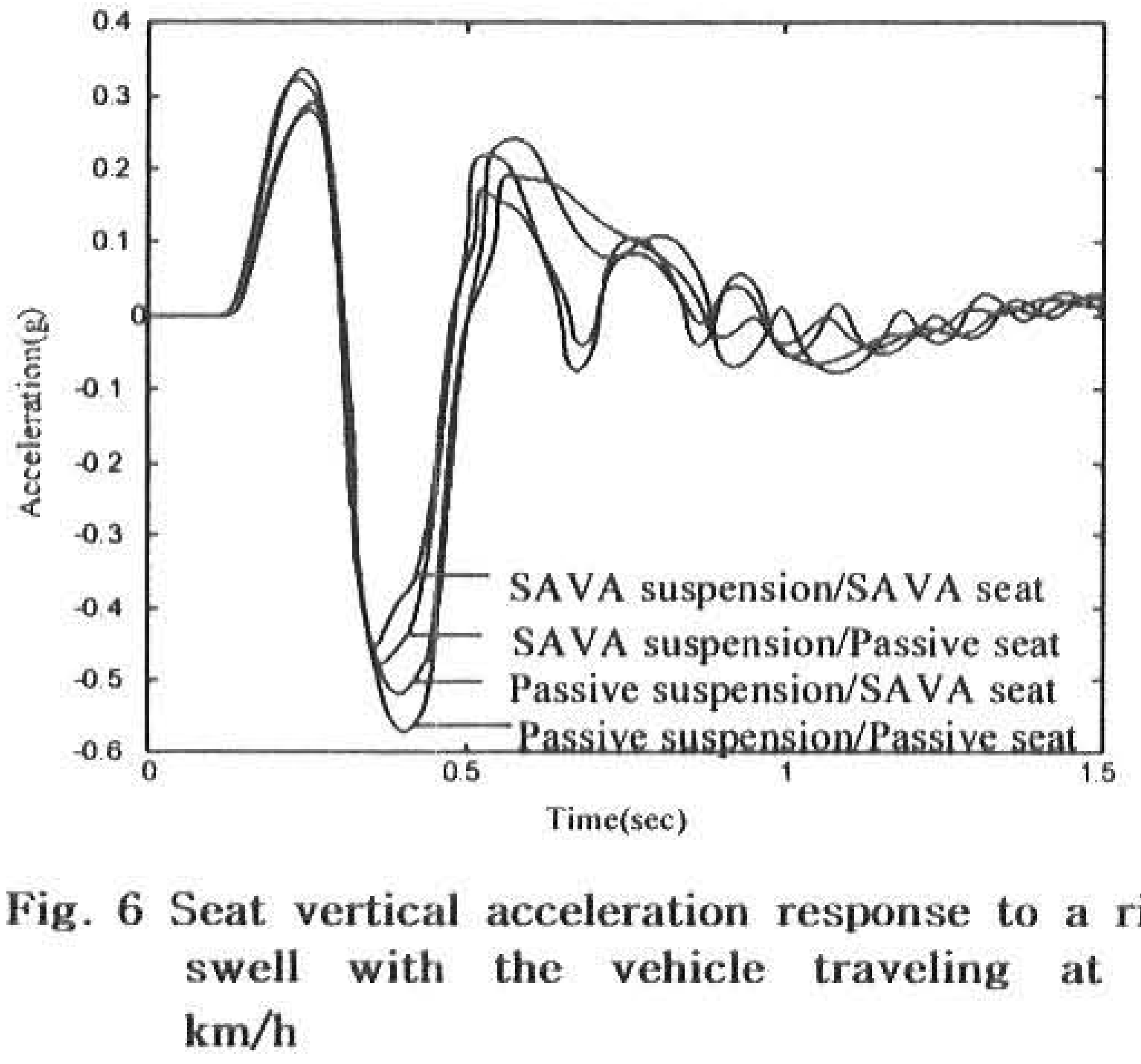

Fig. 6 Seat vertical acceleration response to a ride swell with the vehicle traveling at 50 km/h

Fig. 7 depicts the relative difference between the motion at the seat butt and the seat track to the ride swell. The RMS amplitude of the Passive suspension/Passive seat is reduced by 24% for SAVA suspension /Passive seat system, 12% for Passive suspension/SAVA seat system and 35% for SAVA suspension/SAVA seat system.

Fig. 8 depicts the performance of the designs when the input is treated as a white noise velocity. Comparing the transfer functions shown there to those shown in Fig. 9, it is clear that there is not a great deal of difference between the linear and nonlinear system when comparing seat butt response. SAVA suspension/Passive seat system provides better performance than Passive suspension /SAVA seat system at most frequency range. The SAVA suspension/SAVA seat on the

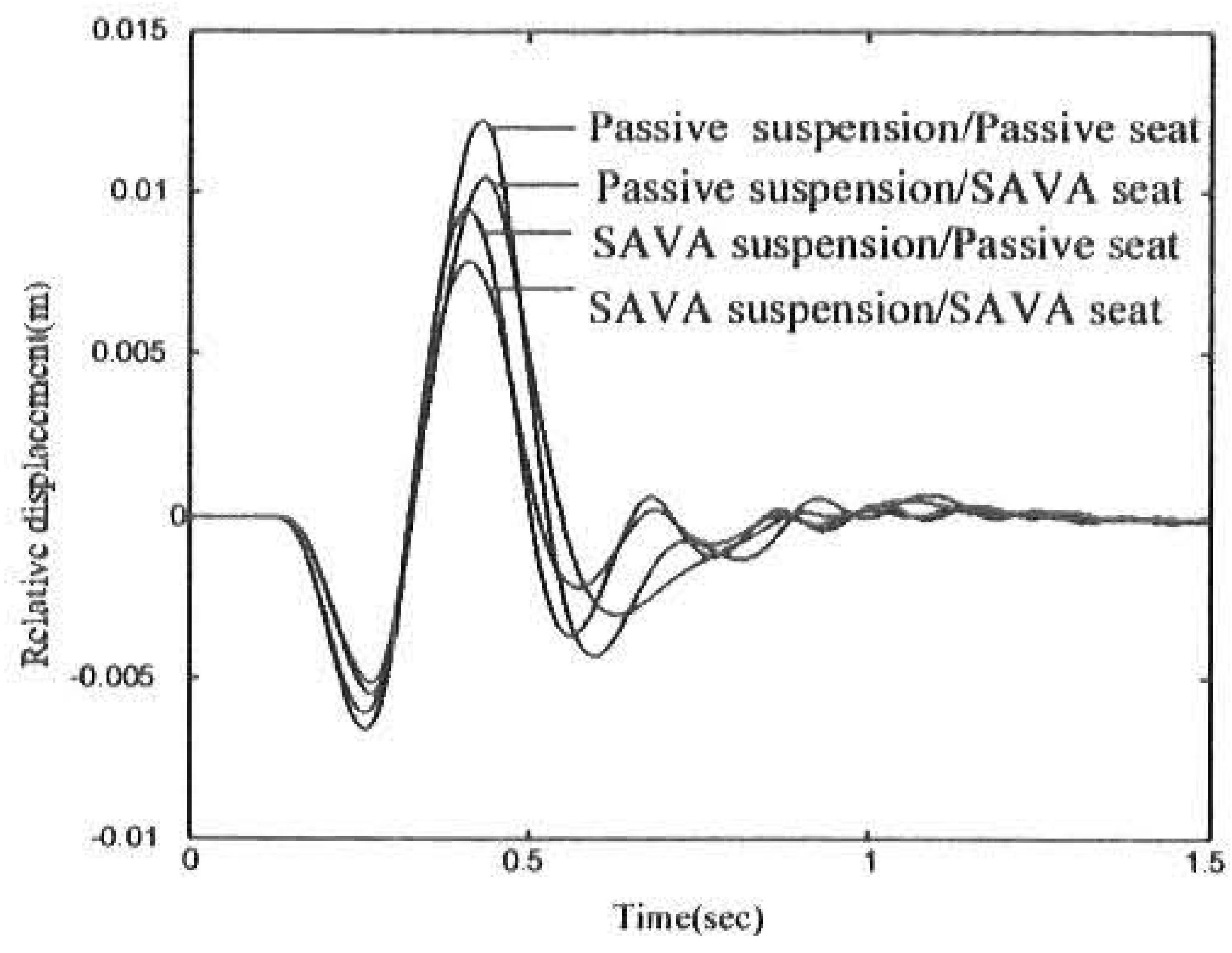

Fig. 7 The relative difference between the motion at the seat butt and the seat track to a ride swell with the vehicle traveling at 50 km/h

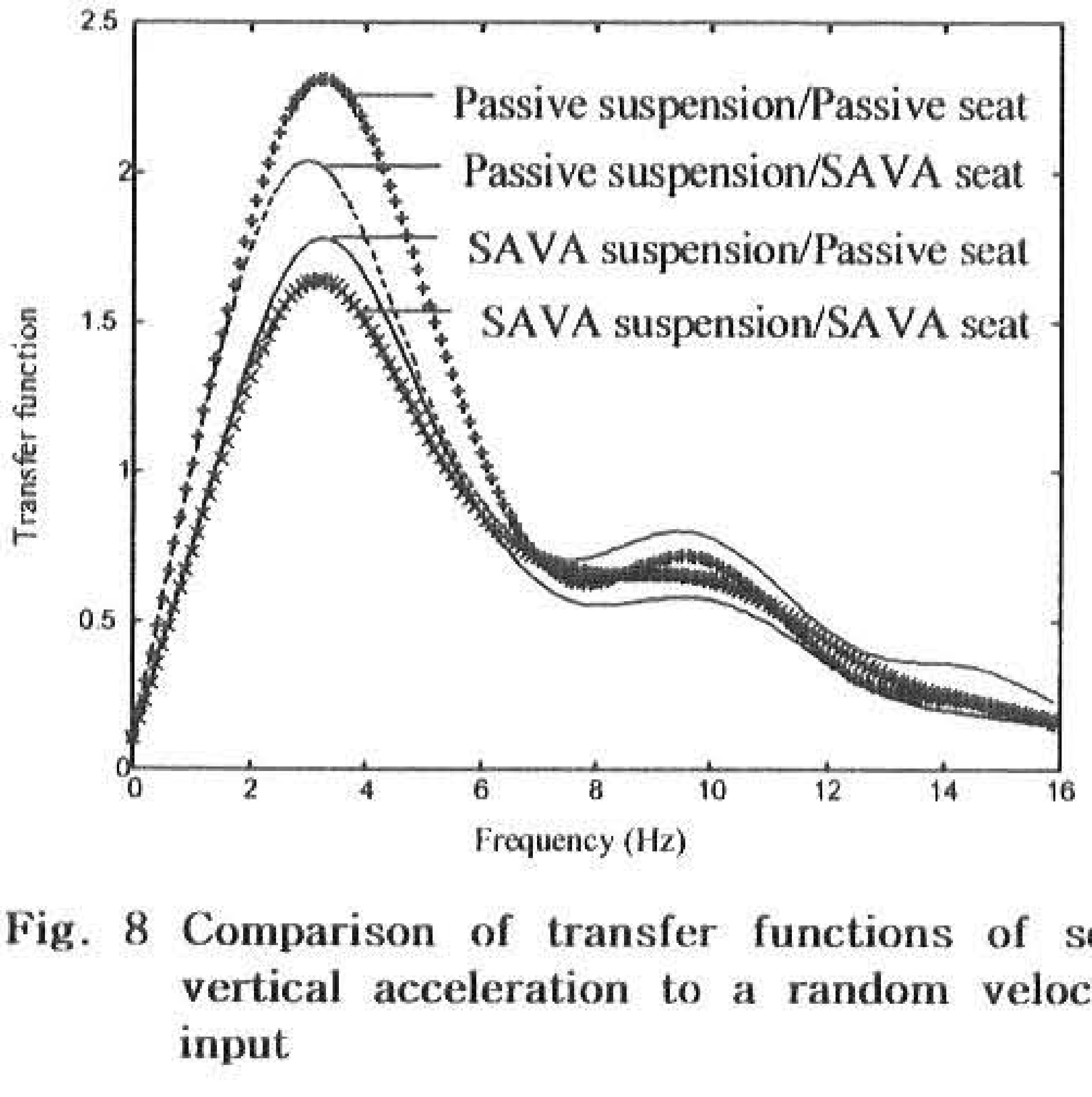

Fig. 8 Comparison of transfer functions of seat vertical acceleration to a random velocity input

other hand turns in the best overall performance. The peak amplitude at the first modal frequency is reduced by 28% for SAVA suspension/Passive seat system, 18% for Passive suspension/SAVA seat system and 34% for SAVA suspension/SAVA seat system. The SAVA suspension/SAVA seat system yielded 45% RMS reduction with a linear seat cushion model.[12)]

Fig. 10 depicts that comparison of transfer functions of the relative difference between the motion at the seat butt and the seat track to a random velocity input. The peak amplitude at the first modal frequency is reduced by 31% for SAVA suspension /Passive seat system, 16% for Passive suspension/SAVA seat system and 40% for SAVA suspension/SAVA seat system.

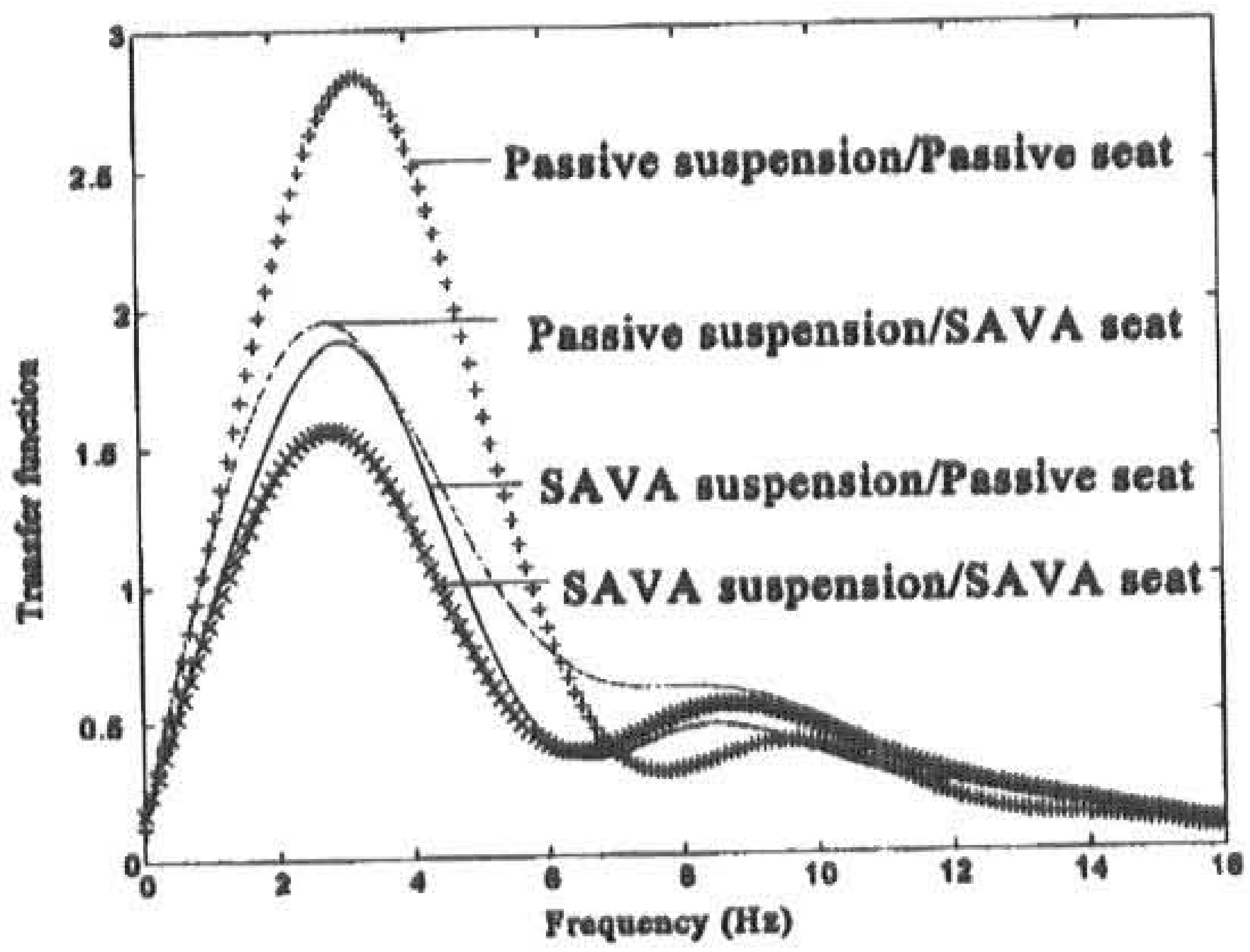

Fig. 9 Comparison of transfer functions of seat vertical acceleration to a random velocity input with a linear seat cushion model[12)]

It is finally noted that the performance of the linear seat is slightly better than that of nonlinear model in terms of reduction in the seat vertical acceleration. It is clear then that if the performance of a seat/suspension control system is determined correctly, then the actual (nonlinear) model of the cushion is essential to the work. A linear seat cushion model suggests more benefit than in fact is possible.

4. CONCLUSION

A 3-DOF quarter car model with a nonlinear seat cushion model was introduced in this paper to design a integrated semiactive seat/chassis suspension control system. The simulation results indicate that the SAVA can reduce ride vibration regardless of characteristic of the seat cushion. The SAVA suspension/SAVA seat system provides a noticeable reduction in the vertical seat butt acceleration, while the expected gains are somewhat diminished when a nonlinear model of the seat cushion's compliance is employed. The results also indicate that the semiactive seat vibration controller can be examined by a seat/suspension system with a lumped human mass without considering actual human subjects. It can be a means of simplified and efficient procedure to design automobile seats.

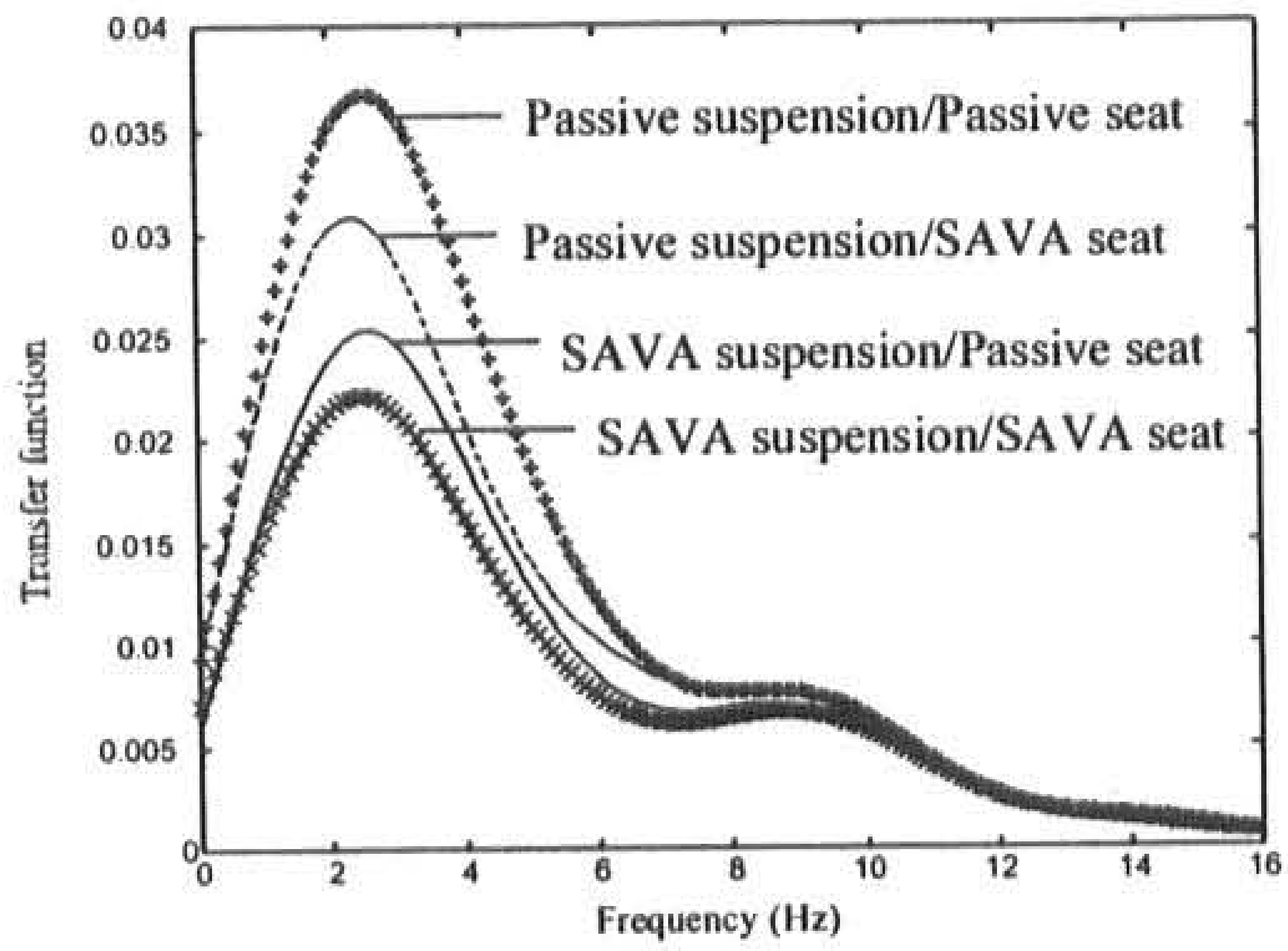

Fig. 10 Comparison of transfer functions of the relative difference between the motion at the seat butt and the seat track to a random velocity input

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