Virtual Reality Simulation of Active Car Suspension System

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***Abstract*---This paper presents the design of a Nonlinear Energy Sink (NES) controller and its application to active suspension systems in the Virtual Reality Environment. In this environment, the design engineers are immersed in an audio-visually coupled tele-operated environment whereby direct interaction with and control of the design process is achieved in real time. In this manner, the behavior of synthetic models of the full car can be monitored by literally walking around the car and adjusting the design parameters of the suspension as needed to ensure optimal performance while satisfying design and operational requirements.**

The control actuators which provide forces equivalent to nonlinear stiffness and damping elements are attached to the vehicle in order to actively isolate it from road excitation. The effect of the parameters of the NES controller on the vehicle performance is studied both in the frequency and time domain. The effectiveness of the NES controller is validated by numerical simulation. The robustness of the nonlinear energy pumping process is studied by varying the magnitude of road excitation. The simulation results in the Virtual Reality Environment show that under certain conditions, the nonlinear energy pumping can be induced and significant vibration isolation can be achieved. The performance of vehicle including the ride comfort and road holding capability can be improved significantly. When the magnitude of road excitation is increased, the capacity of the NES to absorb energy from the main system is also enhanced. This is very important to achieve vibration isolation objectives. The virtual reality simulation results also show that the nonlinear NES controller performs better than the classical LQR controller particularly as the road condition becomes worst.

***Index Terms*---Design in virtual reality environment, nonlinear energy sinks, LQR, controllable suspension, control strategy.**

I. INTRODUCTION

In order to improve the vehicle performance, computer-controlled suspensions such as active or semi-active suspension have been considered. Several control strategies have been proposed. Examples of these strategies include: sky-hook model-following [1], adaptive [2], linear quadratic Gaussian (LQG) [3], neural network [4], nonlinear H_∞ [5 and 6], the mixed H_2/H_∞ [7], sliding mode [8], fuzzy [9], and preview [10] control strategies.

Recently, versions of these strategies have been adopted by many automobile makers which offer controllable suspension systems as an option for their high-end passenger cars. Some examples include the suspensions offered by Mercedes Benz, Toyota, and Jaguar (www.jaguarusa.com/us/en/vehicles). Also, Cadillac Seville and Chevrolet Corvette, are currently employing Delphi's MagneRide semi-active suspension technology (www.delphi.com/pdf/e/magneride.pdf).

In this paper, another control strategy is considered which relies in its operation on **N**onlinear **E**nergy **S**inks (**NES**) as means for achieving shock isolation of broadband unwanted disturbances. These NES have been extensively analyzed by Vakakis [11 and 12], Gendelman *et al*. [13], and Ma *et al*. [14]. Unlike linear passive sinks, as dynamic absorbers, which can operate only in the neighborhood of a single frequency, and are incapable of effectively attenuating transient disturbances, the NES have been shown to suppress transient disturbances in an irreversible manner [11 and 12].

In this paper, the NES are coupled to the car suspension (primary system) by means of nonlinear springs and dampers. Such nonlinear coupling elements are necessary in order to achieve energy pumping and dissipation of unwanted disturbances in a sufficiently fast timescale. Specifically, the idea of the NESs is used to control the vertical dynamics of a seven degree-of-freedom vehicle in order to achieve a desired vehicle body and wheel performance that ensures a balance between the ride and handling performance.

This paper is organized in six sections. In section 1, a brief introduction is given. Section 2 presents the dynamic equations of a full car model with seven degrees of freedom. Section 3 describes the concept of the NES control design strategy and architecture. A numerical example is given in section 4 and implementation in the virtual reality environment is outlined in Section 5. Section 6 summarizes the main results of this work and gives some recommendation for future applications.

II. DYANMICS OF A FULL-CAR MODEL

Assume a vehicle driven on a straight road in a steady-state condition, i.e., with constant thrust and without brake action. In this case, the vertical dynamics of the vehicle including the car body heave, roll, and pitch motions, and the bounce motions of the four wheels. This is a typical seven degrees of freedom characterization of the vertical dynamics used for the computer controlled suspension. The vertical dynamic model is shown in Fig. 1.

Manuscript received on 30 June 2008
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Let $z_w = [z_{w1} \quad z_{w2} \quad z_{w3} \quad z_{w4}]^T$, where z_{wi} is the displacement of the i^{th} wheel of the vehicle, where i =1, 2, 3, and 4 correspond to the front-left, front-right, rear- left and rear-right wheels. Also, the body motion vector q is defined as

$$q = [h \quad r \quad p]^T \tag{1}$$

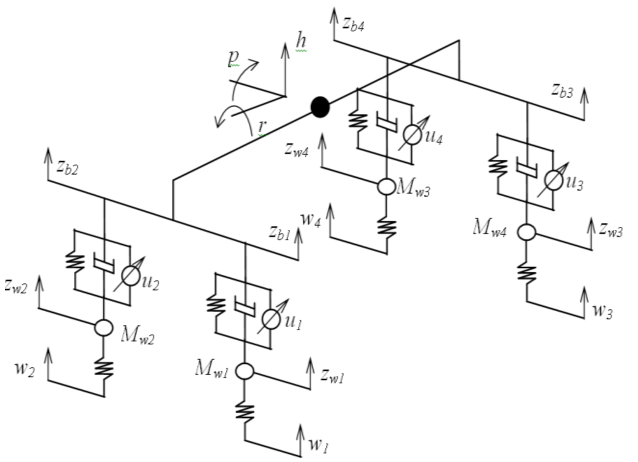

Fig. 1. Seven degrees of freedom full-car model

where h is the heave displacement of the center of gravity of the car body (sprung mass), r is the car body's roll angle, and p is the car body's pitch angle. Both r and p are the global angular displacements with respect to a perfect flat road surface.

Let $Z_b = [\, Z_{b1} \quad Z_{b2} \quad Z_{b3} \quad Z_{b4}]^T$

, where z_{bi} is the vertical displacement of the car body in the i^{th} corner, where i = 1, .., 4 corresponds to the front-left, front-right, rear- left and rear-right corners. Note that z_b and q are related by:

$$Z_b = Hq \tag{2}$$

where H is a transformation matrix given in the appendix. Also, let the relative positions of the suspension be denoted by z_{rp}, then:

$$z_{rp} = z_b - z_w \tag{3}$$

Further, let u be the vector of control forces such that:

$$u = [u_1 \quad u_2 \quad u_3 \quad u_4]^T \tag{4}$$

where u_i is the control force at the i^{th} corner.

Hence, the equations of motion of the full-car model are:

$$M\ddot{\hat{z}} + D\dot{\hat{z}} + K\hat{z} = E_1 w + E_2 u \tag{5}$$

Where $\hat{z} = [q \quad z_w]^T$. Also, the matrices M, D, K, E_1, and E_2 are given in the appendix.

In a state-space representation, the equations of motion reduce to:

$$\dot{\hat{x}} = A\hat{x} + B_1 w + B_2 u \tag{6}$$

where $\hat{x} = [\hat{z} \quad \dot{\hat{z}}]^T$. Also, the matrices A, B_1, and B_2 are defined in the appendix.

III. NONLINEAR ENERGY SINK CONTROL

The NES controller is designed such that the control action u_i is given by:

$$u_i = -N_i g_i \tag{7}$$

where $g_i = [(z_{bi} - z_{wi})^3 \quad \dot{z}_{bi} \quad z_{wi} - w_i \quad \dot{z}_{wi}{}^3]^T$ and N_i is the feedback gain matrix of the i^{th} NES controller which is given by:

$$N_i = [\alpha_{1i}, \alpha_{2i}, \alpha_{3i}, \alpha_{4i}] \tag{8}$$

It can be seen that nonlinear stiffness element $(z_{bi} - z_{wi})^3$ and nonlinear damping element $\dot{z}_{wi}{}^3$ have been introduced into the subsystems of the vehicle.

Note that in the proposed control strategy, four independent NES controllers have been used for each quarter car. The parameters α_i for the gain matrix are selected to optimize the system performance.

IV. PERFORMANCE OF THE NES CONTROLLER

The performance of the NES controller is evaluated for a vehicle with the parameters listed in Table 1. The optimal control gain matrices of the NES are:

$$N_1 = N_2 = [20000, 2203.5, 205.2, -1500]$$

and $$N_3 = N_4 = [20000, 2803.5, 805.2, -1500]$$

TABLE 1: VEHICLE PARAMETERS

Parameter	Unit	Value
M_s	*kg*	1583
I_{xx} / I_{yy}	*kg m^2*	531/2555
$M_{w1}/M_{w2}/M_{w3}/M_{w4}$	*kg*	48/48/74/74
$K_{s1}/K_{s2}/K_{s3}/K_{s4}$	*KN/m*	35/35/34/34
$C_{s1}/C_{s2}/C_{s3}/C_{s4}$	*Ns/m*	400/400/200/200
$K_{t1}/K_{t2}/K_{t3}/K_{t4}$	*KN/m*	220/220/220/220
l_{xf} / l_{xr}	*m*	1.116/1.438
$l_{ylf} / l_{yrf} / l_{ylr} / l_{yrr}$	*m*	0.77/0.77/0.765/0.765

In this paper, chirp signal road profiles are considered. Typical profiles are shown in Fig. 2, indicating dominant frequency content from 0 to 15 Hz [7]. Also, the signals for the left and right wheels are assumed to have a frequency shift in order to excite the pitch and roll modes of the car body. In addition, the vehicle is assumed to be driven straight at a constant speed of 60km/h.

Fig. 3 shows comparisons between the performance of a passive suspension system and an active system controlled by an LQR controller.

The gain matrices of the LQR controller are:

$$k_1 = k_2 = [-1762.6, 846, 789.6, 3.3]$$

and

$$k_3 = k_4 = [-2716.8, 1078.2, 1486.1, -58.9]$$

Fig. 4 and 5 show comparisons between the performance of the NES and LQR controllers both in the time and frequency domains.It can be clearly seen that the NES control strategy is more effective in attenuating the vibration of the vehicle body and improving the handling performance. Furthermore, the suspsion deflections are also attenuated at the same time

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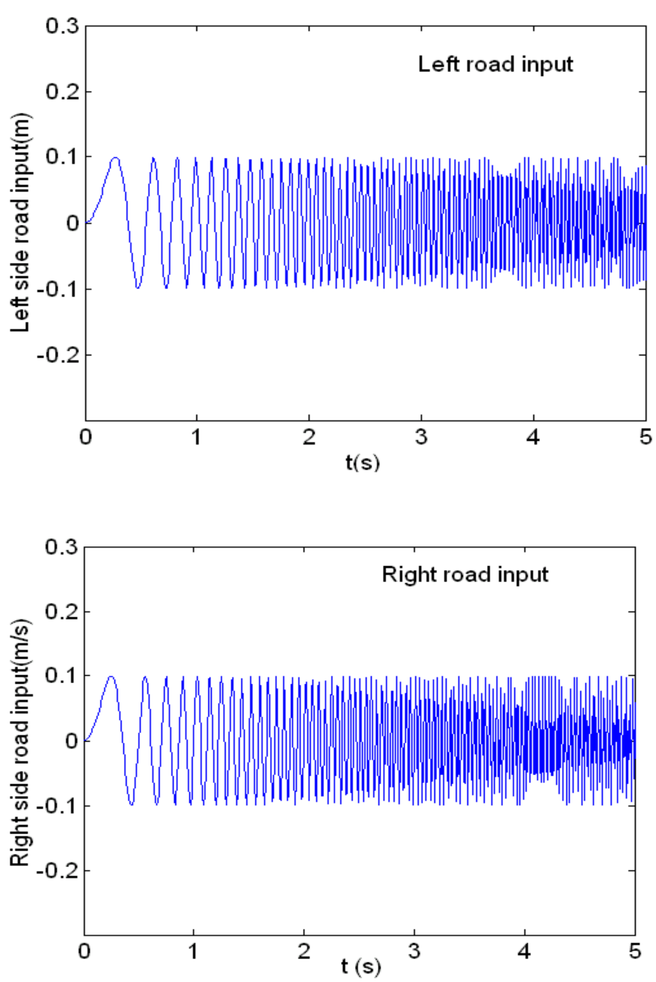
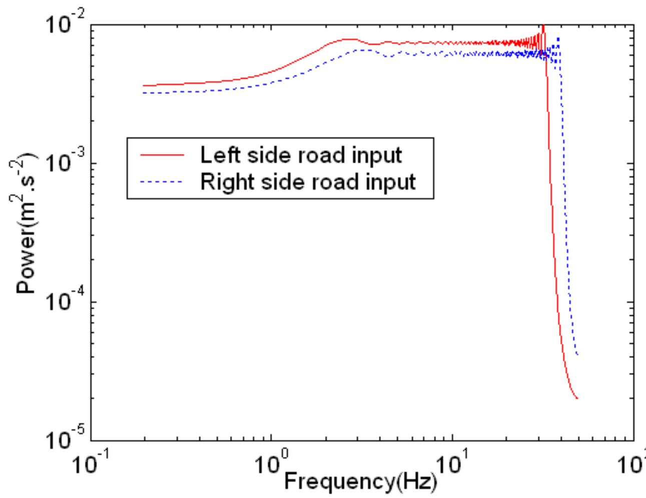


(a) time domain

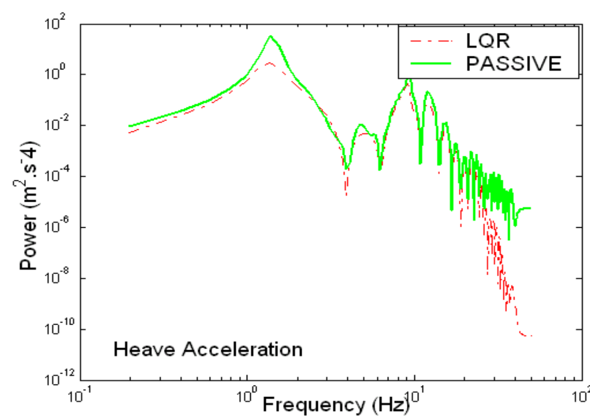


(b) frequency domain

Fig. 2. Time profile and frequency content of chirp road inputs

V. IMPLEMENTATION IN THE VIRTUAL REALITY ENVIRONMENT

The dynamics of the full-car model with the NES controller are implemented in the University of Maryland Virtual Reality (CAVE) Laboratory which is controlled by an eight parallel processor Silicon Graphics Infinite Reality (ONYX2) computer as shown in Fig. 6. In the virtual reality environment, the designer becomes an integral part of the computer-aided design process and the design loop. This is very important as the vehicle designer by being immersed in the virtual reality environment can view the attributes of each of the control strategies in real time and gain considerable insight about the physics of the system. He can favor one controller design over another or modify the design parameters to capture the physics of the problem. Other examples of virtual reality simulation of vehicle dynamics are reported by Ozana [6], Lehner and DeFanti [15], Belyaev and Aranov [16], and Kumar *et al*. [17].



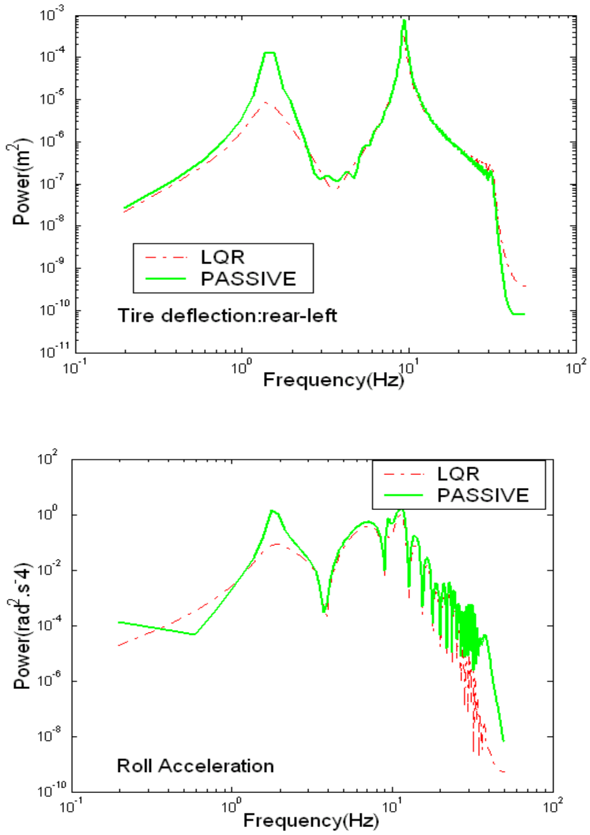
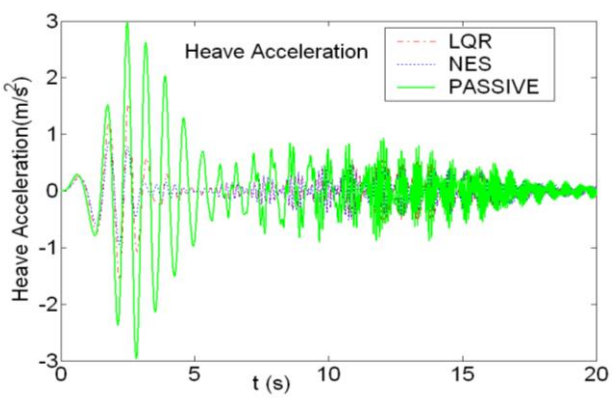
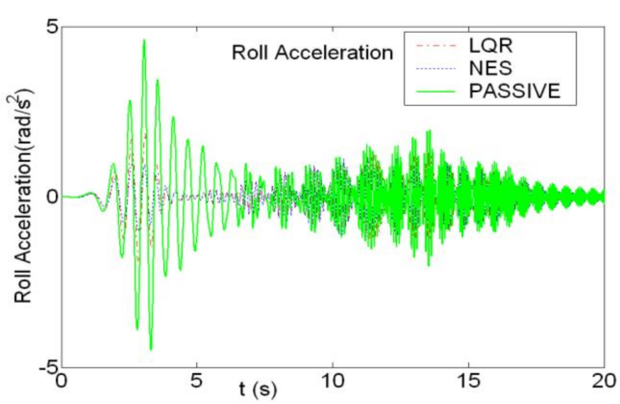


Fig. 3. Frequency response of vehicle with LQR



(a)



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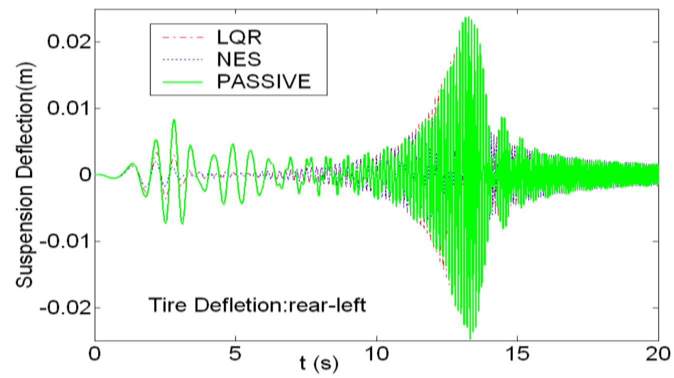


(c)

Fig. 4. Time domain

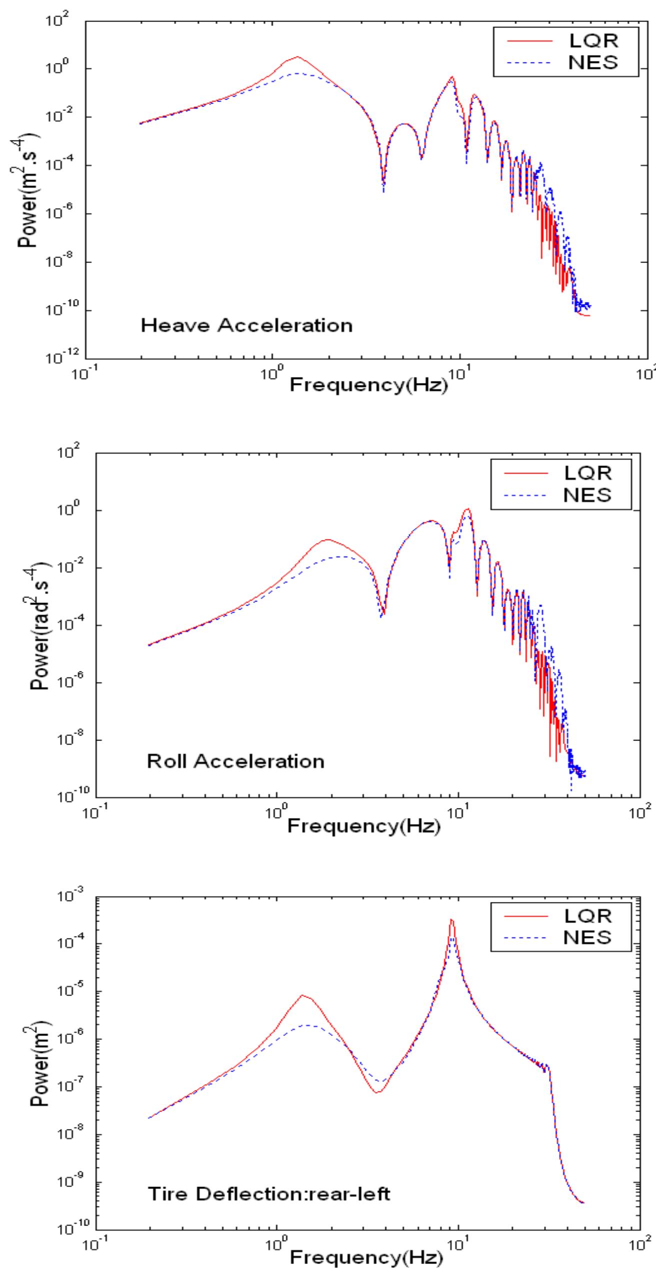
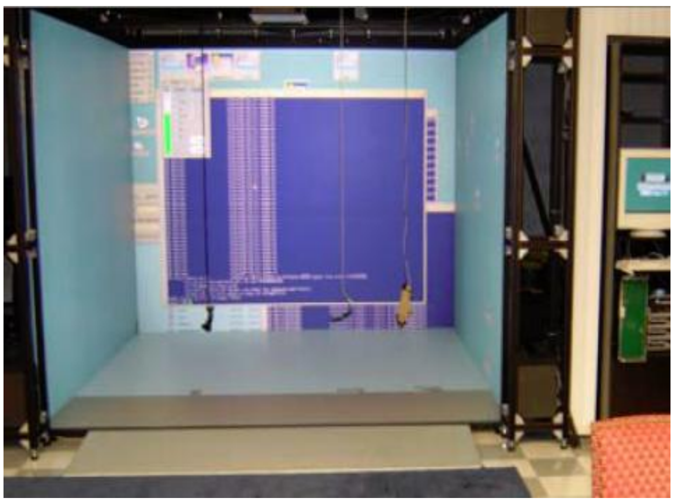


Fig. 5. Frequency domain

(a) SGI-ONYX Computer



(b) CAVE

Fig. 6. Virtual reality laboratory at University of Maryland

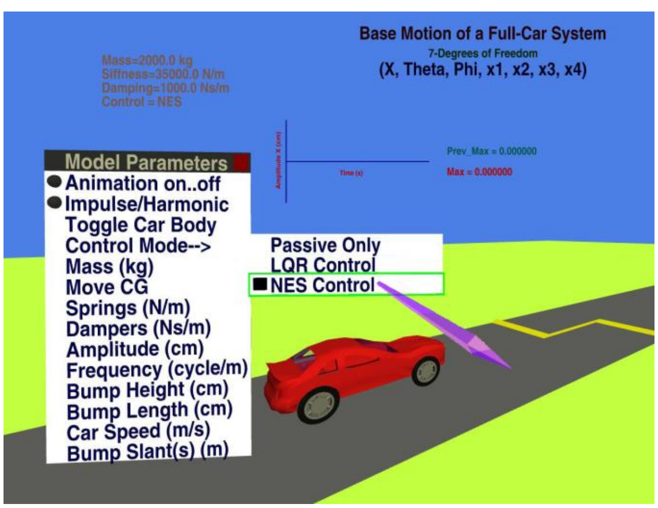


Fig. 7. The main menu of analysis and design of vehicle suspension in virtual reality environment

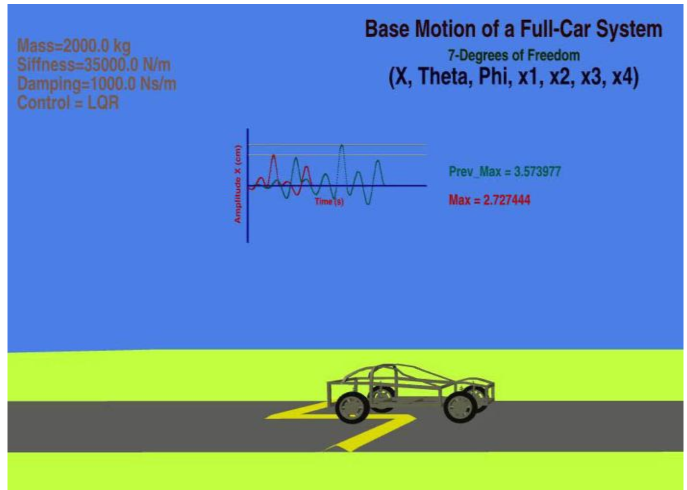


Fig. 8. Display of the design parameters and the corresponding performance characteristics

Fig. 7 shows the main menu of the analysis and design of active vehicle suspension systems in the virtual reality environment. In such an environment, the designer can investigate the effect of the car suspension parameters, the road excitation, car speed, and control strategy on the car dynamics in real time.

Fig. 8 shows, in real time, a comparison between the time response of a vehicle with passive suspension (in green) and a vehicle with an NES controller (in red) when the car passes over a speed bump. Attenuation of the peak amplitude of vibration is evident.

Fig.s 9 through 11 display comparisons between the dynamics of a passive (uncontrolled) vehicle and vehicles controlled with the LQR and NES controllers for heave, pitch, and roll degrees of freedom.

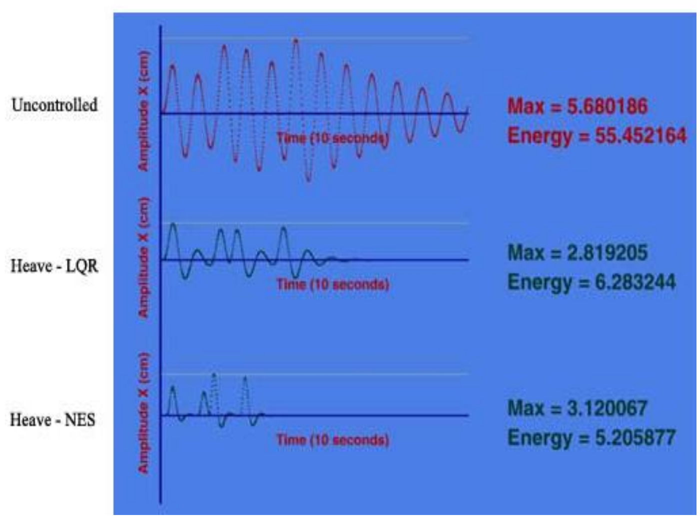


Fig. 9. Comparison of heave dynamics

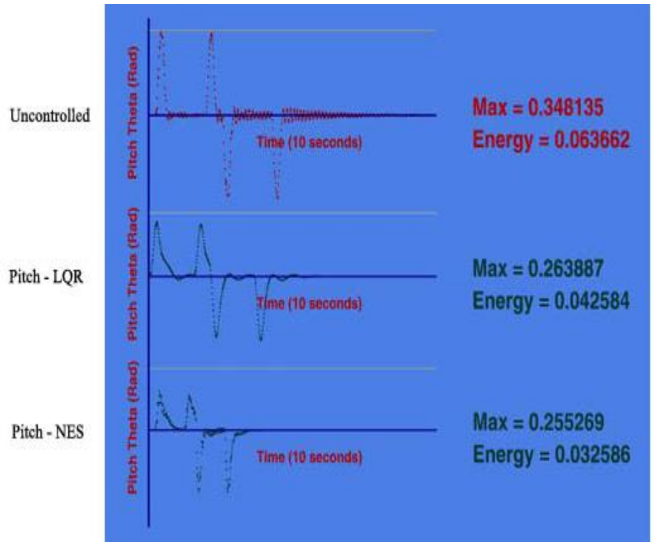


Fig. 10. Comparison of pitch dynamics

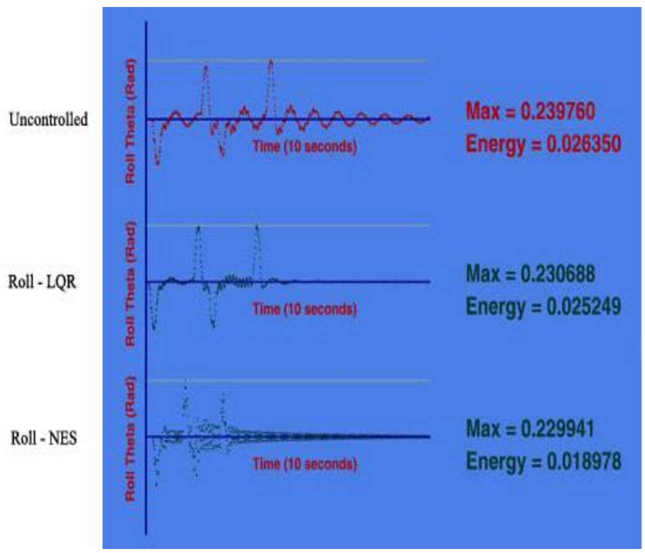


Fig. 11. Comparison of roll dynamics

VI. CONCLUSIONS

This paper has presented the implementation of the dynamics of a seven-degree of freedom vehicle with a nonlinear energy sink controller in the virtual reality environment. The implementation aims at demonstrating the effectiveness of a nonlinear energy sink (NES) controller as compared to the classical LQR controller and uncontrolled suspension.

The simulation results show that NES controllers are more effective than LQR controllers and passive system whether for ride comfort (body performance) or handling performance (wheel performance). With proper selection of the control gains, the NES can generate effective nonlinear energy pumping over a wide frequency range whereby the vibration energy produced by the road disturbance are absorbed into nonlinear attachments and thus an effective vibration isolation is obtained for vehicle.

It is important to note that the design of car suspension systems in the virtual environment enables the design engineer to be fully immersed in an audio-visually coupled tele-operated environment where the physical performance of the suspension can be monitored by literally walking around the car and adjusting its design parameters as needed to ensure optimal performance while satisfying design and operational requirements. Under such conditions, the With such an addition, the designer becomes an integral part of the computer aided design (CAD) process and the design loop where he can make decisions based on the insight gained during the direct interaction with virtual model of the car. Further benefits can be gained when the virtual reality environment is used in the collaborative design mode where several engineers at different sites can share their inputs without physically be at one place.

However, one must emphasize that extensions of the present work are now in progress in order to enhance its practicality and potential. For example, a coupled control system is being developed to improve the performance as compared to the independent wheel controllers considered in this paper. Furthermore, the dynamics of the actuators and the sensors are in the process of being accounted for to ensure practicality of the suspension system. Validation of the optimized configuration experimentally is a natural extension of the present work.

The presented work can also be integrated with driving simulators such as that developed by Sun *et al*. [18] and cognitive response capabilities as reported by Lin *et al*. [19] in order to establish comprehensive virtual reality facilities for car/driver systems. Also, the presented large scale virtual reality simulation in CAVE environment can be equally extended to other vehicle systems such as bicycles [20] and Unmanned Air Vehicle (UAV) [21].

APPENDIX

MATRICES OF CAR MODEL

i. Transformation Matrix

$$H = \begin{bmatrix} 1 & l_{ylf} & -l_{xf} \\ 1 & -l_{ylf} & -l_{xf} \\ 1 & l_{ylr} & l_{xr} \\ 1 & -l_{ylr} & l_{xr} \end{bmatrix}$$

where l_{xf} and l_{xr} are the distances from the front and rear axle to car body CG, respectively, l_{ylf} and l_{ylr} are half of the front and rear wheel tracks.

ii. Mass, Stiffness, and Damping Matrices

$$M_b = \begin{bmatrix} M_s & 0 & 0 \\ 0 & I_{xx} & 0 \\ 0 & 0 & I_{yy} \end{bmatrix}, M_w = \begin{bmatrix} M_{w1} & 0 & 0 & 0 \\ 0 & M_{w2} & 0 & 0 \\ 0 & 0 & M_{w3} & 0 \\ 0 & 0 & 0 & M_{w4} \end{bmatrix}$$

$$K_s = \begin{bmatrix} K_{s1} & 0 & 0 & 0 \\ 0 & K_{s2} & 0 & 0 \\ 0 & 0 & K_{s3} & 0 \\ 0 & 0 & 0 & K_{s4} \end{bmatrix}, C_s = \begin{bmatrix} C_{s1} & 0 & 0 & 0 \\ 0 & C_{s2} & 0 & 0 \\ 0 & 0 & C_{s3} & 0 \\ 0 & 0 & 0 & C_{s4} \end{bmatrix}$$

and

$$K_t = \begin{bmatrix} K_{t1} & 0 & 0 & 0 \\ 0 & K_{t2} & 0 & 0 \\ 0 & 0 & K_{t3} & 0 \\ 0 & 0 & 0 & K_{t4} \end{bmatrix}.$$

where M_s is sprung mass, I_{xx} and I_{yy} are the roll and pitch moments of inertia of the car body. M_{wi} is the unsprung mass at the i^{th} corner, K_{ti} is the i^{th} tire stiffness. K_{si} as the passive suspension spring rate at the i^{th} corner, and C_{si} as the passive damping rate at the i^{th} corner.

$$M = \begin{bmatrix} M_b & 0 \\ 0 & M_w \end{bmatrix}, K = \begin{bmatrix} H'K_sH & -H'K_s \\ -K_sH & K_t+K_s \end{bmatrix}$$

and $$D = \begin{bmatrix} H'C_sH & -H'C_s \\ -C_sH & C_s \end{bmatrix}$$

Also, $E1 = \begin{bmatrix} 0 \\ K_t \end{bmatrix}$ and $E2 = \begin{bmatrix} H \\ -I \end{bmatrix}$

iii. Matrices of state-space model

$$A = \begin{bmatrix} 0 & I \\ -M^{-1}K & -M^{-1}D \end{bmatrix}, B_1 = \begin{bmatrix} 0 \\ M^{-1}E_1 \end{bmatrix}$$

and $$B_2 = \begin{bmatrix} 0 \\ M^{-1}E_2 \end{bmatrix}$$

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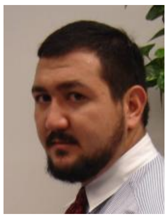
SYMBOLS

C_{si}	passive damping rate at the i^{th} corner
H	transformation matrix
h	heave displacement of CG of car body (sprung mass)
I_{xx}, I_{yy}	roll and pitch moments of inertia of the car body
K_{ti}	Stiffness of the i^{th} tire
K_{si}	passive suspension spring rate at the i^{th} corner

l_{xf} l_{xr}	distances from the front and rear axle to car body CG
l_{ylf} l_{ylr}	half of the front and rear wheel tracks.
M_{wi}	unsprung mass at the i th corner
M_s	sprung mass
N_i	Control gain matrix of NES controller
p	car body's pitch angle.
u_i	control force at the i th corner
z_{bi}	vertical displacement of car body in i^{th} corner
z_{rp}	relative positions of the suspension
z_{wi}	the displacement of the i th wheel of the vehicle



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