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MULTI OBJECTIVE OPTIMIZATION OF VEHICLE SUSPENSION DESIGN

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Abstract

This paper demonstrates the applicability of a multi objective genetic algorithm to analyze the effect of suspension parameters on the ride comfort and road holding capability of vehicles. The maximum vertical acceleration at driver's seat and maximum roll angle are minimized, keeping the suspension working space constrained. The effect of seat position is also considered in the optimization. An 8- degree of freedom (DOF) vehicle model with different suspension systems, such as passive, active and semi active is used for the present study. The optimization is carried out using Non-Dominated Sorting Genetic Algorithm (NSGA- II), developed by Deb (2001). Deterministic road input is applied to the vehicle model. A good pareto optimal solution is achieved in the optimization for all three suspension systems. The results show that the use of active and semi active suspensions is beneficial compared to passive suspension system.

INTRODUCTION

Various types of suspension systems have been developed to reduce the vibrations in an automobile due to road disturbances which are the primary source of vibrations. Redfield (1991) studied different types of suspension systems including semi active and low bandwidth semi active suspension systems. Gobbi *et al.* (1997) reviewed the latest methods in vehicle subsystem optimization. Baumal *et al.* (1998) studied the applicability of genetic algorithms to the design optimization of active vehicle suspension system. The objective was to minimize the maximum acceleration felt by the driver, subject to the constraints representing the required road holding ability and suspension working space. Mohamed Bouazara and Richard (2001) optimized the suspension system using sequential unconstrained minimization technique (SUMT). Els and Uys (2003) optimized the suspension of a sports utility vehicle by Dynamic Q-algorithm to minimize both the maximum vertical acceleration of the sprung mass and the maximum roll angle using Dynamic Analysis and Design System package.

Most of the works in literature on optimization of suspension systems have considered optimization of a single objective only. In the present work, two objectives are considered from the point of ride comfort and handling, with constraints imposed on road holding and suspension working space. The study encompasses passive, active and semi active suspension systems.

SUSPENSION SYSTEMS

The function of the suspension system is to reduce or eliminate the vibrations caused by road disturbances and braking forces. There are different types of suspension systems: passive, active and semi active. Most vehicles have passive suspension systems. A normal passive shock absorber is usually stiffer in rebound than in compression in order to minimize the transmission of energy to the vehicle body from the road input. The passive damping force saturates at high levels of both compression and rebound velocity, ensuring that the maximum force transmitted by the damper is limited. This protects the suspension components from damage or failure resulting from large velocity inputs due to speed bumps and pot holes.

When active damping is included, the total damping force is considered as the sum of the *active* and *passive* contributions. The total damping force is a function of both sprung mass and suspension velocities. The active component of force is a function of sprung mass velocity. This type of suspension is characterized by the need for an external energy source that powers a control system, which continuously controls the force generated by the suspension system. Though it improves the system performance, it has the disadvantage that it cannot be applied without a host of parametric measurements including velocities and deflections, which add to the complexity and cost of such systems.

The control (active damping) force in the present study is obtained as

$$F_{aij} = g_{ij}\,\dot{z}_{sij} \tag{1}$$

where g_{ij} is the active damping constant and $\dot{z}_{sij}$ is the relative vertical velocity between sprung and unsprung masses.

Semi active damping provides an approximate damping control law without necessitating the addition of large amounts of power to the suspension. Any occurrence of a damping force requiring additional power is set to zero because the controlled actuator is only allowed to dissipate power. In the present study the following semi active control, given by Redfield (1991) is used

$$F_{aij} = \begin{cases} g_{ij}\dot{z}_{sij}, & \dot{z}_{sij}V_{rij} > 0 \\ 0, & \dot{z}_{sij}V_{rij} \le 0 \end{cases} \tag{2}$$

where $V_{rij} = \dot{z}_{sij} - \dot{z}_{ij}$ is the relative velocity in the suspension *ij*.

VEHICLE MODEL

A full car model with 8 DOF (Figure 1) is considered for the present study. The degrees of freedom considered are: bounce motions of the four unsprung masses, bounce, pitch and roll of the sprung mass and bounce of seat. The vehicle's yaw motion is not considered.

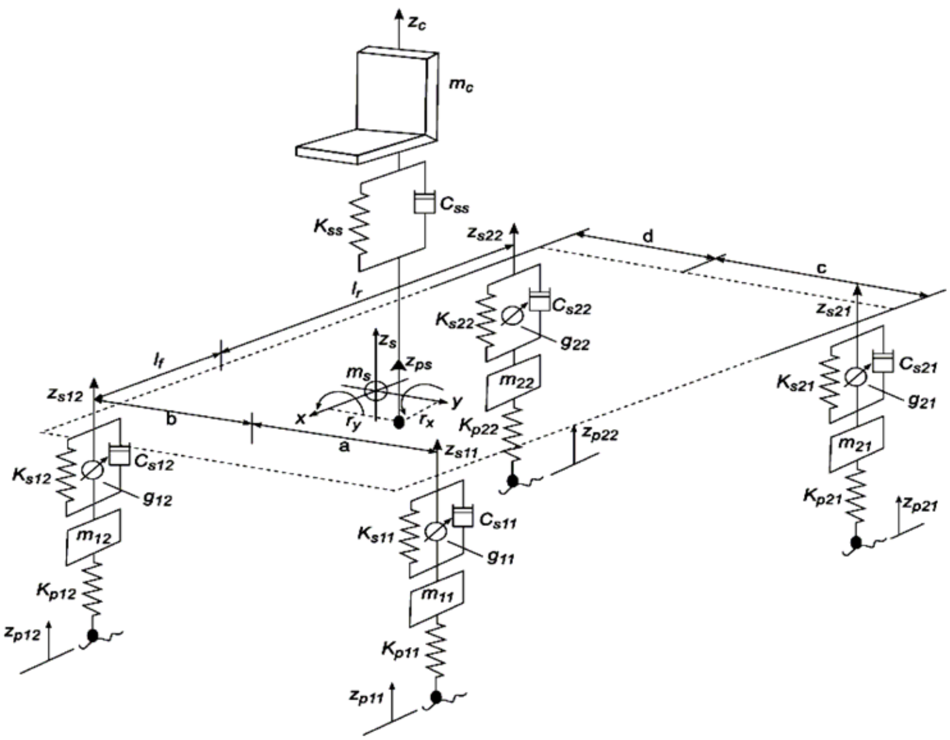

Figure 1 -8 DOF vehicle model (Mohamed Bouazara and Richard (2001))

The system is made up of the following parameters: m_c, m_s, m_{ij} (i, j =1, 2) being the masses of the driver, sprung body and wheel axles ij respectively. C_{ss}, C_{sij} (i, j =1, 2) are damping coefficients and K_{ss}, K_{sij} (i, j =1, 2) are the stiffness coefficients of the seat and vehicle suspension. K_{pij} (i, j =1, 2) are the tire stiffness coefficients and lastly g_{ij} (i, j =1, 2) are the active damping coefficients. r_x and r_y

are the longitudinal and lateral distances of the seat from the centre of gravity of the vehicle. The equations of motion of the system obtained using Newton-Euler equations are as shown below. The force F_{aij} becomes zero in the case of active suspension system.

$$\left.\begin{aligned}
&m_c\ddot{Z}_c = -F_{ss}\\
&m_s\ddot{Z}_s = -F_{s11} - F_{s12} - F_{s22} - F_{s21} - F_{a11} - F_{a12} - F_{a22} - F_{a21} + F_{ss}\\
&I_{sy}\ddot{\theta} = l_f F_{s11} + l_f F_{s12} - l_r F_{s22} - l_r F_{s21} + l_f F_{a11} + l_f F_{a12} - l_r F_{a22} - l_r F_{a21} - r_x F_{ss}\\
&I_{sx}\ddot{\theta} = -aF_{s11} + bF_{s12} - cF_{s22} + dF_{s21} - aF_{a11} + bF_{a12} - cF_{a22} - dF_{a21} - r_y F_{ss}\\
&m_{11}\ddot{Z}_{11} = F_{s11} + F_{a11} - K_{p11}(Z_{11} - Z_{p11}), \qquad m_{12}\ddot{Z}_{12} = F_{s12} + F_{a12} - K_{p12}(Z_{12} - Z_{p12})\\
&m_{22}\ddot{Z}_{22} = F_{s22} + F_{a22} - K_{p22}(Z_{22} - Z_{p22}), \qquad m_{21}\ddot{Z}_{21} = F_{s21} + F_{a21} - K_{p21}(Z_{21} - Z_{p21})
\end{aligned}\right\} \quad (3)$$

where

$$\left.\begin{aligned}
&Z_{ps} = Z_s - r_x\theta + r_y\phi \qquad && Z_{s11} = Z_s - l_f\theta + a\phi\\
&Z_{s12} = Z_s - l_f\theta - b\phi && Z_{s21} = Z_s + l_r\theta + c\phi\\
&Z_{s22} = Z_s + l_r\theta - d\phi && F_{ss} = K_{ss}(Z_c - Z_{ps}) + C_{ss}(\dot{Z}_c - \dot{Z}_{ps})\\
&F_{s11} = K_{s11}(Z_{s11} - Z_{11}) + C_{s11}(\dot{Z}_{s11} - \dot{Z}_{11}) &&\\
&F_{s12} = K_{s12}(Z_{s12} - Z_{12}) + C_{s11}(\dot{Z}_{s12} - \dot{Z}_{12}) &&\\
&F_{s21} = K_{s21}(Z_{s21} - Z_{21}) + C_{s21}(\dot{Z}_{s21} - \dot{Z}_{21}) &&\\
&F_{s22} = K_{s22}(Z_{s22} - Z_{22}) + C_{s22}(\dot{Z}_{s22} - \dot{Z}_{22}) &&\\
&F_{a11} = g_{11}\dot{Z}_{s11},\ F_{a12} = g_{12}\dot{Z}_{s12},\ F_{a21} = g_{21}\dot{Z}_{s21},\ F_{a22} = g_{22}\dot{Z}_{s22} &&
\end{aligned}\right\} \quad (4)$$

PROBLEM FORMULATION

The objective of the problem is to find the optimum values of the vehicle suspension parameters that ensure good ride comfort and handling ability. The objective functions considered are minimization of (i) maximum acceleration of driver's seat and (ii) maximum roll angle. The following relations express these two objectives:

$$\min(\max(\ddot{z}_c)) \qquad \textit{and} \qquad \min(\max(\phi)) \tag{5}$$

The constraints represent good road holding ability and working space. For good road holding, the maximum tire deflection is assumed to be 0.05 m (Baumal, 1998). At least 0.127 m of stroke must be available in order to absorb a bump acceleration of one-half "g" without hitting suspension stops (Gillespie, 1992). The following relations represent the constraints:

$$\left| z_{ij} - z_{pij} \right| - 0.05 \le 0, \qquad \left| z_{sij} - z_{ij} \right| - 0.127 \le 0, \tag{6}$$

A double bump road profile as shown in Figure 2 and as suggested by Mohamed Bouazara and Richard (2001) is used as the road input. The purpose of the double bump is to excite pitching and rolling motions. In the present study, a constant vehicle velocity of 20 m/s is assumed. Table 1 shows the vehicle parameters used and Table 2 the design variables and their bounds.

Table 1- Vehicle parameters

Vehicle parameter	Value	Vehicle parameter	Value
m_c	75 kg	K_{pij} (i, j = 1, 2)	175.5 kN/m
m_s	730 kg	l_f	1.011 m
m_{ij} (i, j = 1, 2)	40 kg	l_r	1.803 m
I_{sy}	1230 kg m^2	a, b	0.761 m
I_{sx}	1230 kg m^2	c, d	0.755 m

Table 2 - Design variables and bounds for 8-DOF model

Design variable	Lower - Upper	Design variable	Lower - Upper
K_{11}, K_{12} (N/m)	13,000-30,000	K_{ss} (N/m)	50,000 - 1,50,000
C_{11}, C_{12} (Ns/m)	500 - 2,000	C_{ss} (Ns/m)	500 - 4,000
K_{21}, K_{22} (N/m)	20,000 - 30000	r_x (m)	0.0 - 0.7
C_{21}, C_{22} (Ns/m)	500 - 2,000	r_y (m)	0.2 - 0.7

The optimization problem is solved using Elitist Non-Dominated Sorting Genetic Algorithm (NSGA-II) (Deb, 2001). This is different from a single objective GA in the way the fitness is assigned to individuals. First, the random population is created and then sorted based on the non-domination level. Second, these solutions are assigned fitness values based on their non-domination, where the first front is allotted the highest fitness value. In the first generation, tournament selection, recombination, and mutation are used to create children having the same population as the parents. For other generations, elitism is applied by combining the children (Q_t) of size N and parents (P_t) of size N together. Non-dominated solution sorting is applied to the combined population (R_t) of size $2N$. However, this time, the crowding distance method is added for the next fitness assignment step for the purpose of preserving diversity. The new parents (P_{t+1}) of size N are then chosen from this combined population based on rank and crowding distances. This new parent population (P_{t+1}) is used for selection, crossover and mutation to create new children (Q_{t+1}). This process continues until it reaches the stopping criteria.

RESULTS AND DISCUSSION

The optimization problem is solved in MATLAB. Since it is a multi objective optimization, a number of optimal solutions have been obtained and they are shown in the following optimal fronts in which every solution is non dominated by other solutions. These are shown in Figures 3, 4 and 5 for passive, active and semi active suspension systems respectively.

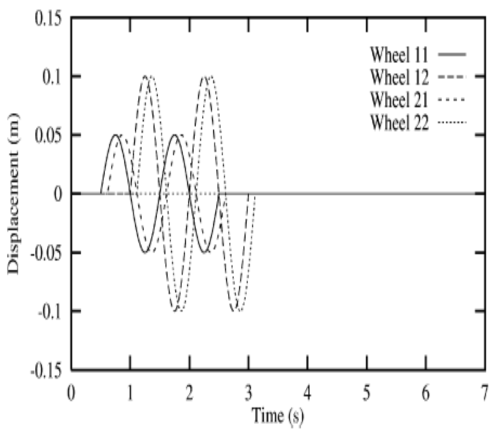

Figure 2 - Road excitation - double bumps

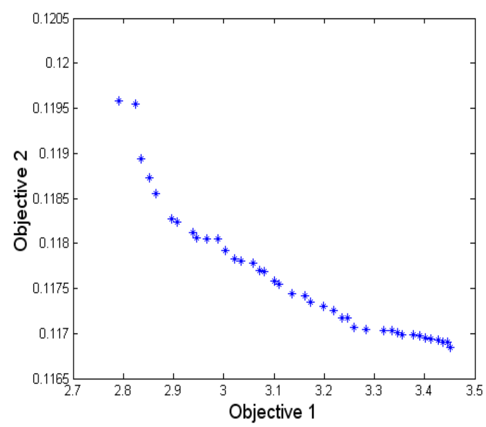

Figure 3 – Optimal front - Passive

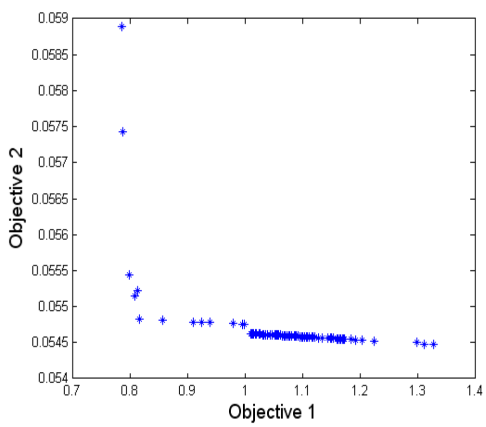

Figure 4- Optimal front – Active

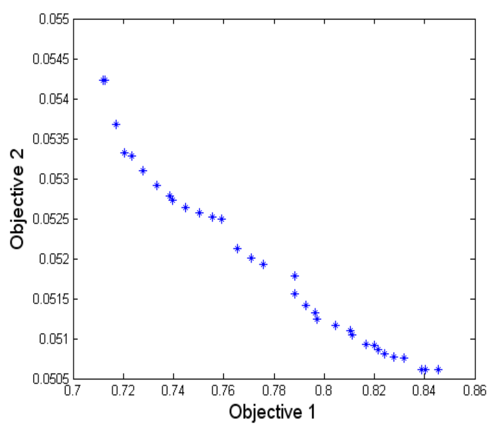

Figure 5 – Optimal front - Semi active

Figures 6 to 11 show a comparison of suspension performance for the three systems. The maximum vertical accelerations at the driver's seat are 3.06, 1.08 and 1.2 m/s^2 for passive, active and semi active suspension systems respectively and the maximum roll angles are 0.13, 0.06 and 0.05 rad respectively. From Figure 6, it is observed that the vertical acceleration at driver's seat is reduced by 65% and 61% by providing active and semi active suspensions respectively. Also the time taken to reach steady state is lower in both cases. It is clear from Figure 7 that the roll angle is decreased by 52% and 55% for active and semi active suspension systems respectively.

From Figure 8, the reduction achieved in the vertical acceleration of sprung mass is 57% and 43% by providing active and semi active suspension systems respectively. Pitch acceleration as shown in Figure 9 is also reduced considerably in active suspension system (35 %). The semi active damper also reduces the pitch acceleration, but the peak response exceeds that of the passive suspension system by 11%.

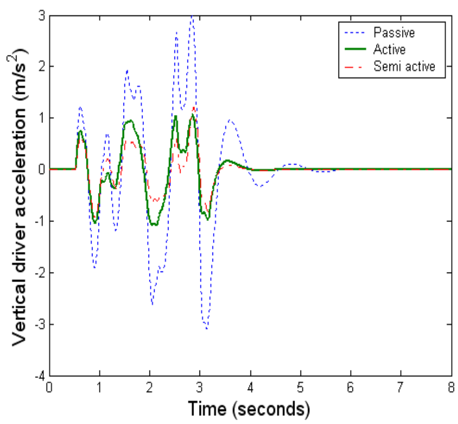

Figure 6 - Vertical acceleration of driver

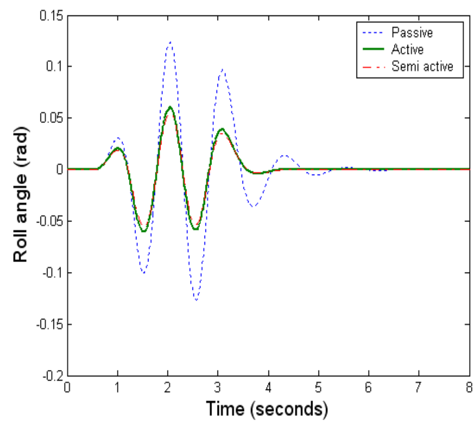

Figure 7 - Roll angle

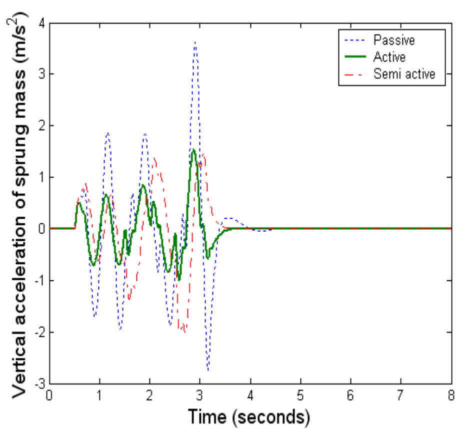

Figure 8 – Sprung mass vertical acceleration

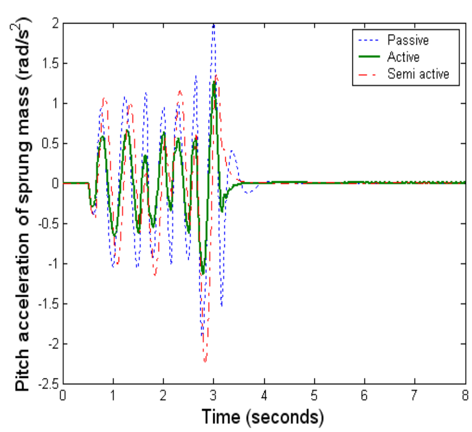

Figure 9 - Pitch acceleration

The road holding capability criteria can be expressed in terms of suspension and tire deflections. It is observed from Figure 10 that the reduction in suspension deflection is 36% for the active and semi active systems as compared to the passive one. The suspension deflection for the right wheel of the rear axle is shown in Figure 11 with a reduction of 15% for active and semi active systems.

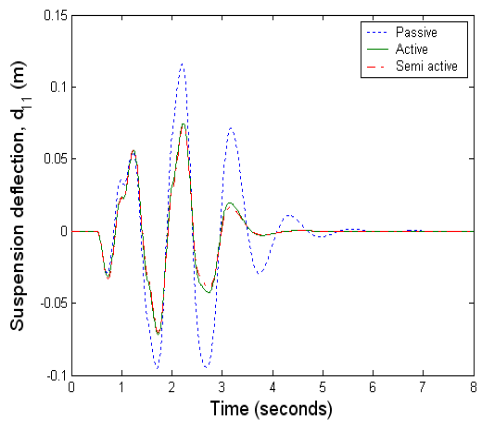

Figure 10 - Suspension deflection, d_{11}

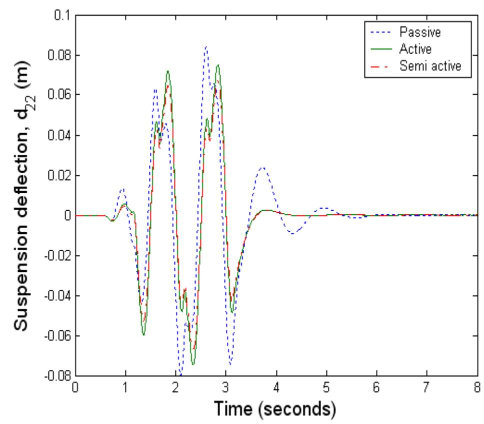

Figure 11 - Suspension deflection, d_{22}

CONCLUSIONS

The present study shows the advantages of active and semi active suspension systems over a passive one in terms of ride comfort and road holding capability. Besides, the semi active suspension is better than the active system from the point of view of road holding. The applicability of multi objective optimization using NSGA-II is proven to be promising. Good pareto optimal fronts are obtained in all the three cases.

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