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Optimisation of an Active Suspension Force Controller using Genetic Algorithm for Random Input

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ABSTRACT

A novel control scheme for the active suspension in a 4-DOFs half-car model is presented. A force cancellation control scheme is used to isolate the sprung and the unsprung masses. Skyhook damper and virtual damper concepts are employed to stabilise the sprung and unsprung masses respectively. Road-following springs are applied for the sprung mass to follow the trend of the road surface condition and to maintain the suspension stroke within a reasonable range. For efficiency, genetic algorithm is employed to search for the parameters like damping ratio and spring constant to achieve an optimum trade off among ride comfort, handling quality, and suspension stroke simultaneously for random input. Computer simulations are performed using MATLAB software to verify the proposed control scheme and effectiveness of the applied genetic algorithm.

Keywords: Active suspension, force cancellation control scheme, genetic algorithm, dampers

NOMENCLATURE

a	Road roughness constant
b_{cf}, b_d, b_s, b_t	Damping constants for compensation front end dynamics, skyhook, suspension, virtual tyre
DOFs	Degrees of freedom, 1-bounce of sprung mass, 2-pitch of sprung mass, 3-bounce of front wheel, 4-bounce of rear wheel
f_a, f_p	Active and passive force
F_b	Balance force
λ	Wave number or spatial frequency
F_1, F_3	Road following spring forces at front- and rear-ends
F_2, F_4	Skyhook damping force at front- and rear-ends
F_5, F_6	Virtual tyre damping force at front- and rear-ends
FH	Front wheel handling quality index
FS	Front suspension travel index
I_p	Pitch inertia of spring mass
k_{cf}	Spring constant for compensating front-end dynamics
k_r,k_s,k_t	Road following, suspension spring constants, and tyre stiffness respectively
l_f, l_r	Distance from front- and rear-ends to mass centre

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m_s, m_u	Sprung and unsprung masses
OBJ	Objective function
PA	Pitch motion index
RH	Rear wheel handling quality index
RQ	Vertical ride quality index
RS	Rear suspension travel index
$S_x(\lambda)$	Wave number spectrum of road elevation
u,t,g,x	Weighting factors in objective function
v	Vehicle forward speed
α	Slope constant
z_p	Vertical displacement of sprung mass centre
z_r	Road input
z_s	Vertical displacement of sprung mass
z_u	Vertical displacement of unsprung mass
θ_p	Pitch displacement

Suffix

f	front
r	rear

1. INTRODUCTION

The development of active suspension systems has recently gained a significant momentum in vehicle design activities. Compared with the passive system, generally constructed with the springs and dampers, the active suspension offers greater capability to control the system behaviour by detecting the appropriate state variables in a specified control scheme[1]. With rapid advances in electronic technology[2], it becomes increasingly feasible to realise an active suspension system. Consequently, ride comfort and handling qualities can be greatly improved by applying suitable control schemes without excessively degrading the compactness of suspension working space.

By the late 60's, random vibrations became a research topic and the advent of random vibration analysis focused on finding the best set of spring and damping rates for specific problems by often minimising the mean square responses. Vehicle vibrations were measured assuming the road to have the concept of isotropic road roughness[3,4] and optimisations were carried out in frequency and time domains assuming road roughness and their cross relations. The design of optimum vehicle suspensions and isolators subjected to random excitation has been the subject of research by Bender[5], Karnopp, Trikha[6] and Dahlberg[7]. Stochastical optimal control theories were also extensively used, e.g., Hac[8], Narayan and Raju[9] and Elmadany[10].

In short, the key objective of an active suspension design mentioned above is to minimise the objective function that contains the sprung mass displacement, tyre deflection, and sprung mass acceleration for improving performance of ride comfort, handling quality, and vehicle compactness, respectively. Since the model of the suspension system is nonlinear or the objective function includes nonlinear terms, specific control methods need to be developed to handle the nonlinearity associated with the model.

Moran and Nagai[11] used a neural network for the identification and control of a half-car model with nonlinear suspensions. Yeh and Tsao[12] proposed a fuzzy preview control scheme with a quarter-car model which employed force cancellation and virtual damper concepts and generated a reference curve via fuzzy interference to improve the ride and handling qualities for vehicles over rough terrains. Moreover, based on the principle of popular genetics, genetic algorithms have become an effective search and optimisation procedure for engineering application. In a genetic algorithm study, Davis[13] and Goldberg[14] provided comprehensive introduction and review of the technology. Tsao and Chen[15,16] applied genetic algorithm to propose a force control scheme in an active suspension design with both quarter-car and half-car models for input deterministic for an effective search for specific control parameters.

2. SYSTEM DESCRIPTION

The vehicle is modeled as a dynamic system made of masses, interconnected by linear springs and dampers for both passive and the active system incorporating the control scheme. The 4-DOF half-car passive model and the active model are shown in Figs 1 and 2 respectively. For the active model spectral densities of acceleration of all 4-DOFs are calculated and plotted against temporal frequency using method of frequency response function.

Equating the net forces to zero, the following equations are obtained:

(a) Sprung mass

$$m_s \ddot{z}_p = f_{pf} + f_{af} + f_{pr} + f_{ar}$$

where

$$f_{pf} = -k_{sf}(z_{sf} - z_{uf}) - b_{sf}(\dot{z}_{sf} - \dot{z}_{uf})$$

$$f_{pr} = -k_{sr}(z_{sr} - z_{ur}) - b_{sr}(\dot{z}_{sr} - \dot{z}_{ur})$$

f_{af} is the front active force, and f_{ar} is the rear active force.

(b) Unsprung mass front

$$m_{uf}\ddot{z}_{uf} = -f_{pf} - f_{af} + k_{tf}(z_{rf} - z_{uf})$$

(c) Unsprung mass rear

$$m_{ur}\ddot{z}_{ur} = -f_{pr} - f_{ar} + k_{tr}(z_{rr} - z_{ur})$$

(d) Equation of pitch

Now equating sum of moments to zero, one gets the equation of pitch as

$$I_p\ddot{\theta}_p = (f_{pf} + f_{af})l_f - (f_{pr} + f_{ar})l_r$$

It is also assumed that the pitch angle is small, so

$$z_{sf} = z_p + l_f\theta_p$$
$$z_{sr} = z_p - l_r\theta_p$$

The following parameters are used for controller design[6]:

m_s	= 730 kg	b_{sf}	= 1290 N s/m
I_p	= 1230 kgm^2	b_{sr}	= 1620 N s/m
m_{uf}	= 40 kg	k_{tf}	= 175, 500 N/m
m_{ur}	= 35.5 kg	k_{tr}	= 175, 500 N/m
k_{sf}	= 19, 960 N/m	l_f	= 1.011 m
k_{sr}	= 17, 500 N/m	l_r	= 1.803 m

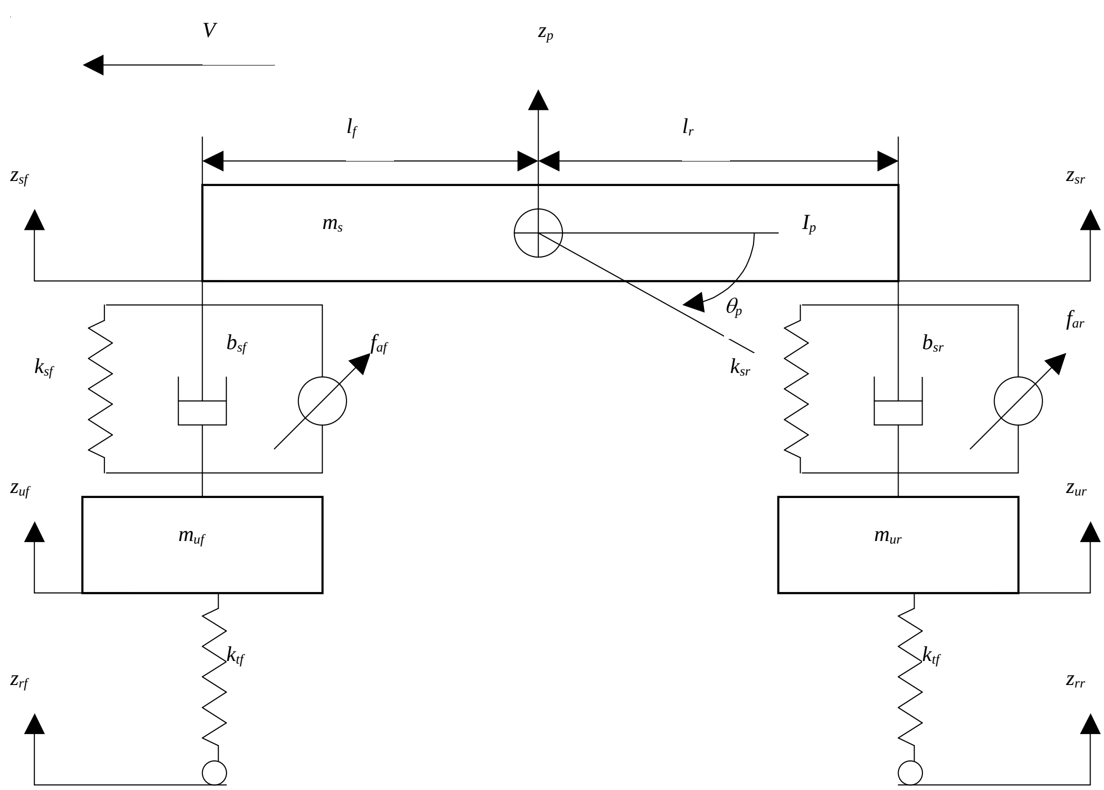

Figure 1. Half-car model.

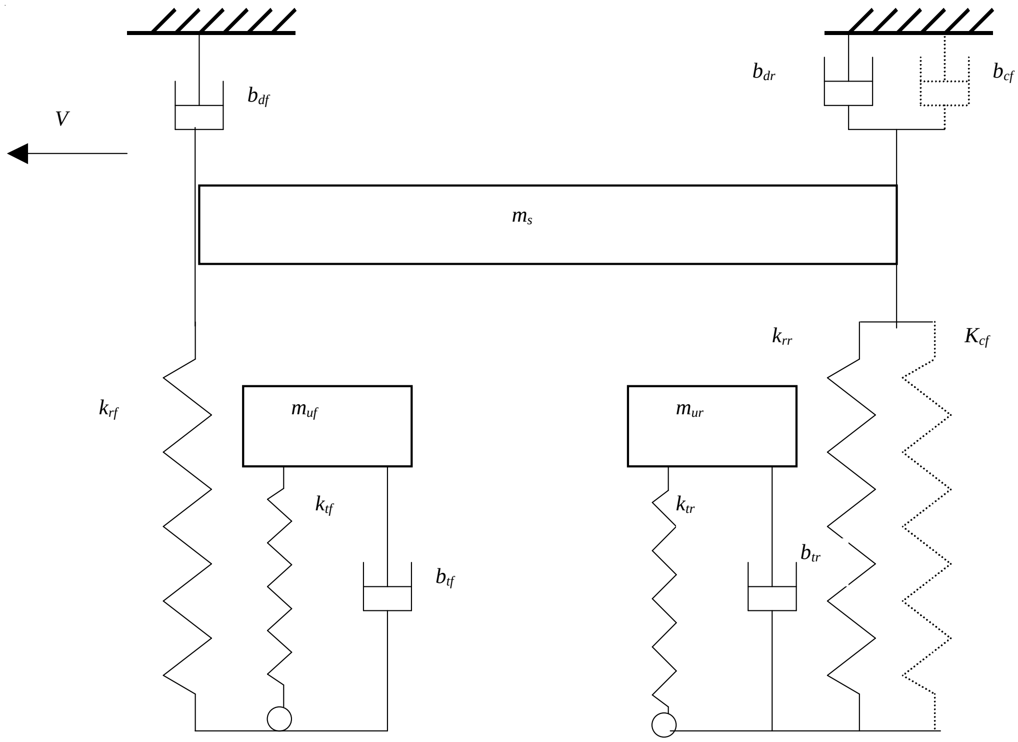

Figure 2. Control configuration.

3. DESIGN CONCEPTS OF FORCE CONTROL

In this study, force cancellation, skyhook damper, virtual damper and road-following spring concepts have been used to design the force controller in active suspension. This is called a force cancellation control scheme and is intended to improve ride comfort.

3.1 Force Cancellation

The dynamics of sprung mass indicates that the vehicle body will not fluctuate if there are no external forces acting on the sprung mass. To achieve ride comfort, it is necessary to cancel the forces caused by the passive springs and dampers and which act on the sprung mass, resulting in heave and pitch accelerations. In this design, the active forces are generated with equal magnitude but in opposite direction to the passive force, such that sprung mass is completely isolated from the unsprung mass and the acceleration of sprung mass is ideally zero. This scheme is intended to improve ride comfort.

3.2 Skyhook Damper, Virtual Damper, and Road-following Spring

According to the force cancellation concept, the sprung and unsprung masses are separated into three subsystems, i.e., sprung mass, front wheels and rear wheels. By observing the dynamic equations after force cancellation, the behaviour of sprung mass is like a double integrator without damping and is unstable. To stabilise the vehicle motion, it is necessary to feedback the velocity signals of the sprung mass wrt the inertial frame. This is known as skyhook damper. Further, the front and rear wheels become two mass spring subsystems without damping, which are also unstable according to the force cancellation control scheme. To stabilise the tyre road subsystems, the relative velocities between tyres and road are fedback to increase the damping of two unsprung masses, called the virtual damping concept.

To optimise the suspension working space, the distance between the sprung mass and road surface should be fed back to maintain reasonable ranges among the sprung mass, unsprung mass and road surface and to make the vehicle body follow the tendency of the road surface. This is called road-following spring.

Although the passive spring and damper forces are eliminated by force cancellation, the forces generated by skyhook damper, virtual damper and road-following spring will result in acceleration acting on the sprung mass. Hence, it is important to determine the parameters of the skyhook damper, virtual damper and road – following spring simultaneously so that these effects can be significantly reduced.

3.3 Force Control Scheme of a Half-car Model

To achieve better ride quality, the force cancellation control scheme (Fig. 2) is applied to balance the passive force. The front actuator will generate the balance force F_{bf} to balance passive spring force (caused by k_{sf}) and damping force (caused by b_{sf}). Similarly, F_{br} generated by the rear actuator cancels the rear passive spring and damper forces. As force cancellation causes instability, the skyhook damping concept and road-following spring concept are employed to ensure stability. F_1 and F_2 are used to produce the conceptual road following spring (k_{rf}) and skyhook damping (b_{df}) forces in the front end of the vehicle. Similarly, F_3 and F_4 are used to generate the rear-end forces[21] of k_{rr} and b_{dr} respectively.

When the road input is near the natural frequency of the tyre-road subsystem, the tyre deflection will be enlarged owing to the force cancellation. So, the virtual damping concept is utilised to prevent it. The active forces F_5 and F_6 are introduced to provide enough tyre damping in front (b_{tf}) and rear (b_{tr}) tyres to improve handling quality. In addition, signals of z_{rf}-z_{sf} and z_{sf} are fed back to rear actuator to compensate the effect transferred from the front- end to the rear-end of sprung mass[21]. The active control forces mentioned above are summarised as

$$F_{bf} = k_{sf}(z_{sf} - z_{uf}) + b_{sf}(\dot{z}_{sf} - \dot{z}_{uf}) = -f_{pf}$$

$$F_{br} = k_{sr}(z_{sr} - z_{ur}) + b_{sr}(\dot{z}_{sr} - \dot{z}_{ur}) = -f_{pr}$$

$$F_1 = k_{rf}(z_{rf} - z_{sf})$$
$$F_2 = b_{df}\dot{z}_{sf}$$
$$F_3 = k_{rr}(z_{rr} - z_{sr})$$
$$F_4 = b_{dr}\dot{z}_{sr}$$
$$F_5 = b_{tf}(\dot{z}_{rf} - \dot{z}_{uf})$$
$$F_6 = b_{tr}(\dot{z}_{rr} - \dot{z}_{ur})$$

Hence, the front actuator force (f_{af}) and rear actuator force (f_{ar}) can be expressed as

$$f_{af} = F_{bf} + F_1 + F_2 + F_5$$
$$f_{ar} = F_{br} + F_3 + F_4 + F_6 + k_{cf}(z_{rf} - z_{sf}) + b_{cf}\dot{z}_{sf}$$

where the eight parameters k_{rf}, b_{df}, k_{rr}, b_{dr}, b_{tf}, b_{tr}, k_{cf}, and b_{cf} should be determined simultaneously.

4. MODELLING OF ROAD ROUGHNESS

The spectral characteristics of road roughness have been measured extensively for various types of roads, from high quality highways to poorly maintained secondary roads[17-19]. The measurements are analysed in terms of wave-number spectra, or power spectral density where the independent variable is distance rather than time. Most of the road roughness data follow an equation, whereas a first order of approximation, the wave number spectral density of road elevation is proportional to a negative power of the wave number as follows:

$$S_x(\lambda) = a * \lambda^{-\alpha}$$

where, α is a dimensionless constant while the value of a varies with the value of α. The values of α and a for different road surfaces are shown in Table 1. For vehicle vibration analysis, it is more convenient to express the spectral density of surface profiles in terms of temporal frequency in Hz than in terms of spatial frequency since vehicle vibration is a function of time. The relationship is

$$f(\text{Hz}) = v * \lambda \quad \text{and} \quad S(f) = S(\lambda)/v$$

Wave number spectra provide a convenient, vehicle and speed-independent measure of roughness of roads. One has

Table 1. Slope constant and road ruffness constant for different road surfaces

Road surface description	α	***a*** **(m^2)**
Smooth runway	3.8	$4.3 * 10^{-11}$
Rough runway	2.1	$8.1 * 10^{-6}$
Smooth highway	2.1	$4.8 * 10^{-7}$
Highway with gravel	2.1	$4.4 * 10^{-6}$
Pasture	1.6	$3.0 * 10^{-4}$
Plowed field	1.6	$6.5 * 10^{-4}$

$$f = v * \lambda$$

$$S_x(f) = S_x(\lambda)/v$$

$$S_x(\lambda) = a * \lambda^{-\alpha}$$

$$S_x(f) = \frac{a}{v} * \left(\frac{f}{v}\right)^{-\alpha}$$

$$S_x(f) = a * f^{-\alpha} v^{\alpha - 1}$$

5. FREQUENCY RESPONSE FUNCTION

For a linear system, a direct relationship exists between the input and the output. This relationship also holds for random functions. The vehicle system characterised by its transfer function modifies the input representing surface irregularities to the output representing the vibration of the vehicle. The transfer function or frequency response function is defined as the ratio of the output to input under steady-state condition.

Spectral density of the input and output of the system are related by

$$S_{output}(f) = \left| H(f)^2 \right| * S_{input}(f)$$

5.1 Complex Frequency Function and Output Spectral Densities for a Pitch Plane Model

The computation of complex frequency function can be done by giving a unit harmonic input for the appropriate input r and keeping it zero for all other inputs. The force equation is given as

$[D]H_r(\omega) = F_r(\omega)$, here r = 1.2 for 4-DOFs

where D is a 4×4 complex dynamic stiffness matrix given by

$$[D] = \omega^2 [M] + i\omega[C] + [K]$$

where $[M]$, $[K]$, $[C]$, $[F]$ are mass, stiffness, damping, and force matrix, respectively and H_r is 4×1 complex frequency response function vector for r^{th} ground input.

For a linear system having n input, the expression for spectral density of the output process is given by

$$S_y(\omega) = \sum_{r=1}^{n}\sum_{s=1}^{n} H_r^*(\omega) H_s(\omega) S_{x_r x_s}(\omega)$$

In matrix form, it is represented by

$$S_y(\omega) = H_r *^T (\omega)[S_{x_r x_s}(\omega)] H_s(\omega)$$

where $S_x(\omega) = S_{x_r x_s}(\omega)$ is the 2×2 input spectral density matrix and is given by

$$S_x(\omega) = \begin{bmatrix} S_{x1} & S_{x1x2} \\ S_{x2x1} & S_{x2} \end{bmatrix}$$

where the input spectral density function for a given type of road, $S_d = S_{x1} = S_{x2}$

$$S_{x1x2} = S_d * e^{-i\omega l/v}, S_{x2x1} = S_d * e^{i\omega l/v}$$

l is the wheel base. Hence,

$$S_{yi}(w) = [H_{1i}^* \quad H_{2i}^*][S_x(w)] \begin{bmatrix} H_{1i} \\ H_{2i} \end{bmatrix}$$

i is the required DOF.

The plots of spectral density for displacement and acceleration corresponding to each DOF against frequency are plotted for passive system. The area under these curves gives the mean square value of the acceleration for each DOF, which in turn would decide the comfort criteria. Forms of input are assumed for three road conditions and the response calculated.

6. DESIGN CRITERIA

It is assumed that (i) All springs and damping rates are assumed to be constant, (ii) vehicle structure is non deformable, (iii) vehicle forward speed is constant, (iv) suspension system friction is negligible in comparison to shock absorber damping, (v) viscous damping is rubber and is neglected, (vi) suspension deflections are small, and (vii) road is considered to be rigid

The key objective of the active suspension design is to minimise the performance index that contains the sprung mass acceleration, tyre deflection and suspension displacement for improving the performance of ride comfort, handling quality (road holding) and vehicle compactness, respectively. In the present design, full advantage is taken care of suspension working space. The following responses were obtained in the form of mean square values for both passive and active models:

(a) Vehicle bounce acceleration indicating discomfort

(b) Tyre deflection indicating road holding

(c) Relative displacement between wheel and vehicle body indicating working space of suspension.

The evaluation functions are chosen as follows:

$RQ = E[(\ddot{z}_p)_n]^2$ is the mean square value of acceleration of sprung mass

$PA = E[(\ddot{\theta}_p)_n]^2$ is the mean square value of pitch acceleration of sprung mass

$FH = E[(z_{rf} - z_{uf})_n]^2$ is the mean square value of relative displacement between the road and the forward wheel.

$FS = RMS[(z_{sf} - z_{uf})]$ is the root mean square value of relative suspension displacement of forward end

$RH = E[(z_{rr} - z_{ur})_n]^2$ is the mean square value of relative displacement between the road and the rear wheel

$RS = RMS[(z_{sr} - z_{ur})]$ is the root mean square value of relative suspension displacement of rear-end.

where, *RQ* is vertical ride comfort index; *PA* is pitch motion index; *FH* is front wheel handling quality index; *FS* is front suspension travel index; *RH* is rear wheel handling quality index; and *RS* is rear suspension travel index.

6.1 Genetic Algorithm Calculation for Parameter Searching

Genetic algorithm manipulates the population of potential solution to an optimisation or search problem. Specifically they operate on encoding representation of the solution, equivalent to the genetic material of the individuals in nature. Essential to the simple genetic algorithms working is a population of binary strings. Each string of 0s and 1s is the encoded version of a solution to the optimisation problem. The initial population is chosen randomly and simple operations are carried out on the initial populations to generate successive populations that improve over time. A simple genetic algorithm that yields good results in any practical problem is composed of following three operations:

- *Reproduction*: A process in which individual strings are copied according to their objective function values.
- *Crossover*: Here, pairs of strings are picked at random from the existing population to be subjected to crossover.
- *Mutation*: After crossover, strings are subjected to mutation. Mutation is applied to each child individually.

The flow chart for design procedure of active suspension force controller is shown in Fig. 3. The programming for design procedure has being implemented using MATLAB. In the design procedure of genetic algorithm, 100 individuals are selected to be the population size. The range of encoded values is 0 to 4095(=2^{12}–1). The extraordinary condition is that the ranges of last two parts, i.e., k_{cf} and b_{cf} are chosen as from –2048 to 2047 to

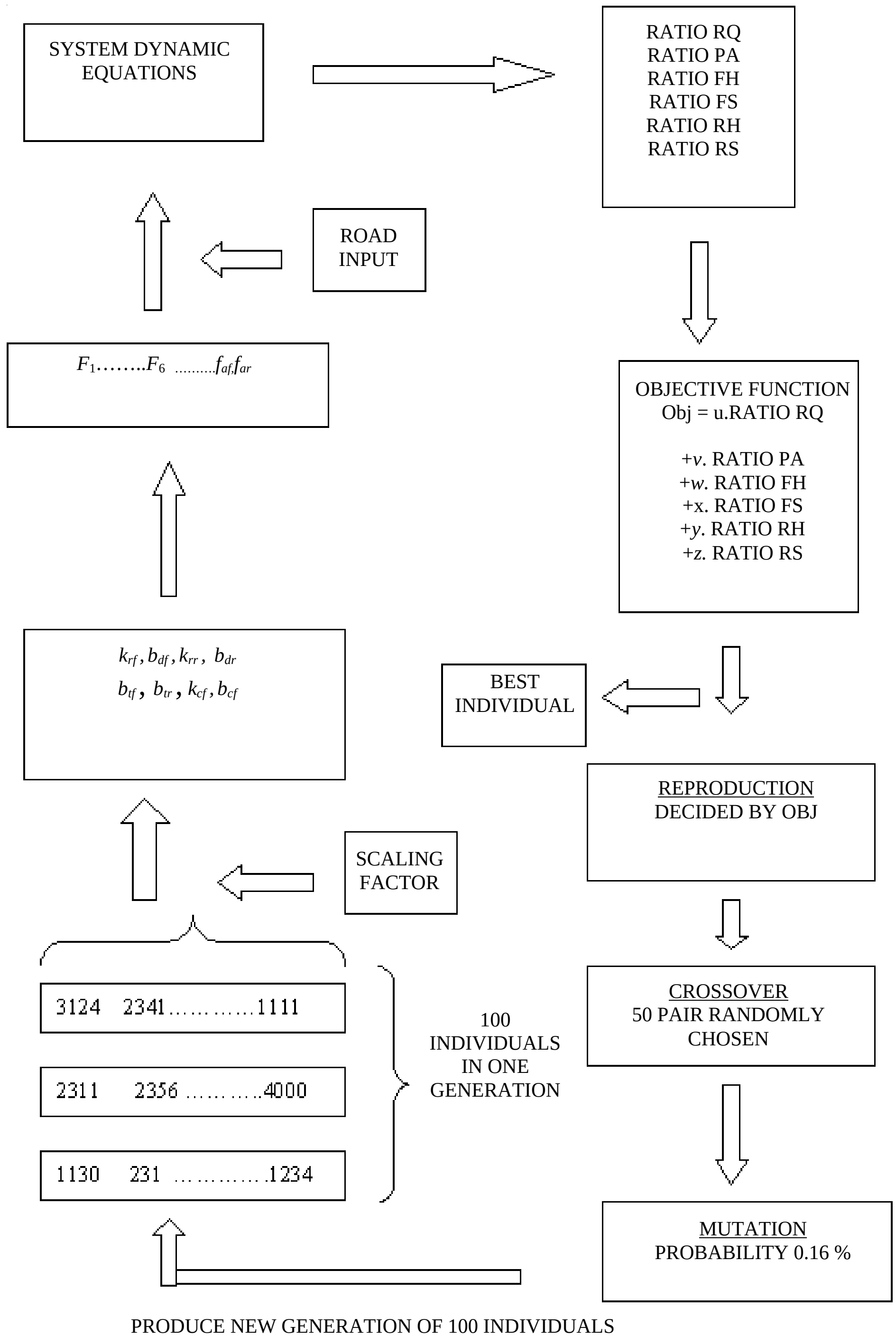

Figure 3. Flow chart of design porcedure.

satisfy all possibilities as the sign of last two parts is dependent on the dynamic behaviour of the front-end of the sprung mass.

The objective function (*OBJ*) used to evaluate the fitness of the individuals is:

$$OBJ = u * \text{ratio } RQ + t * \text{ratio } PA + g * \text{ratio } FH + x * \text{ratio } FS + e * \text{ratio } RH + z * \text{ratio } RS$$

where ratio $RQ = RQ/PRQ$ (PRQ is RQ in the passive system), ratio $PA = PA/PPA$ (PPA is PA in the passive system), and so forth. The coefficients *u*, *t*, *g*, *x*, *e*, and *z* are the weighting factors as per the importance given to the output. The reproduction operator selects the fit individuals from the current population and places them in mating pool. Highly fit individuals get more copies in the mating pool whereas the less fit ones get fewer copies. After carrying out crossover, mutation (0.16 %) is applied to the new offspring to ensure potential candidates in the genetic algorithms mechanism. At the end of the mutation one gets a new generation of 100 individuals. The same procedure is repeated again and again till the objective function converges.

7. RESULTS AND DISCUSSION

The computer programs for implementation of design procedure based on genetic algorithm were run for the vehicles with random road input corresponding to highway with gravel. These were measured road input of highway with gravel. The vehicle is assumed to be traveling at a constant velocity of 17 m/s. Both approximate method and roulette-wheel selection of reproduction have been used[14]. HG1, HG2 pertains to approximate selection of reproduction schemes and HG3, HG4 pertains to roulette-wheel selection of reproduction scheme as referred in Table 2. In approximate selection of reproduction method, the objective function is sorted based on fitness value and most fit individuals have more representation in the next generation.

7.1 Driving Condition for Highway with Gravel

Computer simulations are performed for four cases (HG1 to HG4). Table 2 lists the various weighting factors, the scaling factors and the final values of the eight parameters in each case. In the first case (HG1) and in the first run, all weighting factors are chosen to be 1, i.e., equal importance is being given to each design criterion. The results show (Fig. 4) that all performance indices are <1 except for the front suspension stroke (ratio FS = 1.9). This indicates that the maximum suspension displacement of the front-end is larger than that of the passive system by 90 per cent. However, the ride comfort is greatly improved for both heave and pitch accelerations.

In the second case (HG2), the weighting factor of the front suspension is chosen as 10 to emphasise the limitation of the front stroke. Also, the heave acceleration index is chosen as 10 to emphasise its importance for comfort levels. Consequently after optimisation the front-travel index becomes 1.07 (7 % higher than passive) being effectively improved. Although bounce acceleration of the sprung mass, the heave acceleration, and handling qualities were deteriorated compared to the first case, these were still superior to those of the passive system.

In the third case (HG3), all weighting factors are 1; the results show that all performance indices are < 1 except for the front suspension stroke (ratio FS = 1.89). This indicates that the maximum suspension displacement of the front-end is larger than that of the passive system by about 89 per cent. However, the ride comfort is greatly improved for both heave and pitch accelerations.

In the fourth case (HG4), the weighting factor of the front suspension is chosen as 10 to emphasise

Table 2. Weighting factors, scaling factors, and final parameter values for highway with gravel[21]

Weighing factor	HG1	HG2	HG3	HG4	Scaling factor
u	1	10	1	10	
t	1	1	1	1	
g	1	1	1	1	
x	1	10	1	10	
e	1	1	1	1	
z	1	1	1	1	
krf	6500	65000	47500	200900	100
bdf	383900	13000	254500	357600	100
btf	6.9	247	306.2	22.1	100
krr	0	0	74300	115400	100
bdr	273000	403000	286800	307000	100
btr	13	0	209.6	172.4	100
kcf	-6600	-200800	–122000	-88200	200
bcf	-3300	-3300	–71900	113600	100

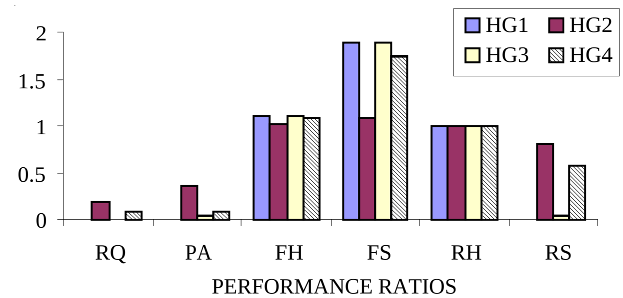

Figure 4. Performance comparison for highway with gravel.

the limitation of the front stroke. Also, the heave acceleration index is chosen as 10 to emphasise its importance for comfort levels. Consequently after optimisation the front travel index becomes 1.73, being effectively improved. Although vertical acceleration of the sprung mass, the heave acceleration, and handling qualities were deteriorated compared to the first case, these were still superior to those of the passive system.

Figures 5(a) - 5(d), and 6(a) - 6(b) show the spectral density graphs of cases HG2 and HG4 with weighting factors. Figures 5(a) to 5(d) show output spectral densities in terms of displacement whereas Figs 6(a) to 6(d) show acceleration spectral densities as a function of temporal frequency and give comfort levels directly. Figure 6(a) shows the acceleration spectral density graphs for sprung mass for both active and passive systems. The area under the graphs gives the mean square value of the acceleration of sprung mass. It can be clearly seen that the mean square value acceleration values of active system have reduced. This is prominent both in the bounce frequency peaks and the wheel hop frequency peaks. Figure 6(c) and 6(d) also show that the acceleration levels of the unsprung mass (rear) and pitch acceleration have reduced considerably. Figures 7(a) - 7(d), and 8(a) - 8(d) show the spectral density graphs using roulette selection. Acceleration spectral density graphs show considerable reduction for active system except for the forward suspension unsprung mass bounce. However suitable weighting factors would reduce that too.

Figure 4 shows the relative comparison of performance indices of all four cases and Figs 9 and 10 show the objective function transition and convergence of cases HG2 and HG4. The noisy nature of convergence using roulette can be made out easily because of population selection process is truly random. Table 2 gives parameter values, weighting factors, and scaling factors for the four cases to show the effect of these.

It was also considered that the vertical mode natural frequency obtained in all cases is limited to 1 Hz, which ensures adequate rattle space and a good design. It can also be seen that the wheel hop frequencies obtained lie outside the frequency range of vibration to which the human body is most sensitive, i.e., < 8 Hz. Also, the pitch and the bounce frequencies lie close so that the pitch motion does not create a bounce motion at a location away from the CG.

8. CONCLUSION

The study gives a control scheme for force control in an active vehicle suspension design using genetic algorithm with a half-car model for random input. A force cancellation control scheme is used to isolate the sprung and unsprung masses. Skyhook damper and virtual damper concepts are employed to stabilise the sprung mass and the unsprung mass. In addition, road-following springs are applied for the sprung mass to follow the trend of road surface condition and to maintain the suspension stroke within a reasonable range. The ideas mentioned above are then integrated to develop the control scheme that improves the ride comfort and handling quality, while the suspension stroke is restricted to be less than or equal to that in the passive system.

Using input spectral densities and frequency response function, mean square acceleration as a measure of comfort is calculated from the output spectral density by subjecting the model to random excitations. Further in the design of systems subjected to random excitations, the constraints on frequency can be specified from knowledge of the PSD function. It was observed from the above study that if the natural frequencies of the system lie in regions where the PSD of the excitations has small values, the response levels are reduced leading to a better design. In addition if the natural frequencies are in the region where the PSD of the excitation is

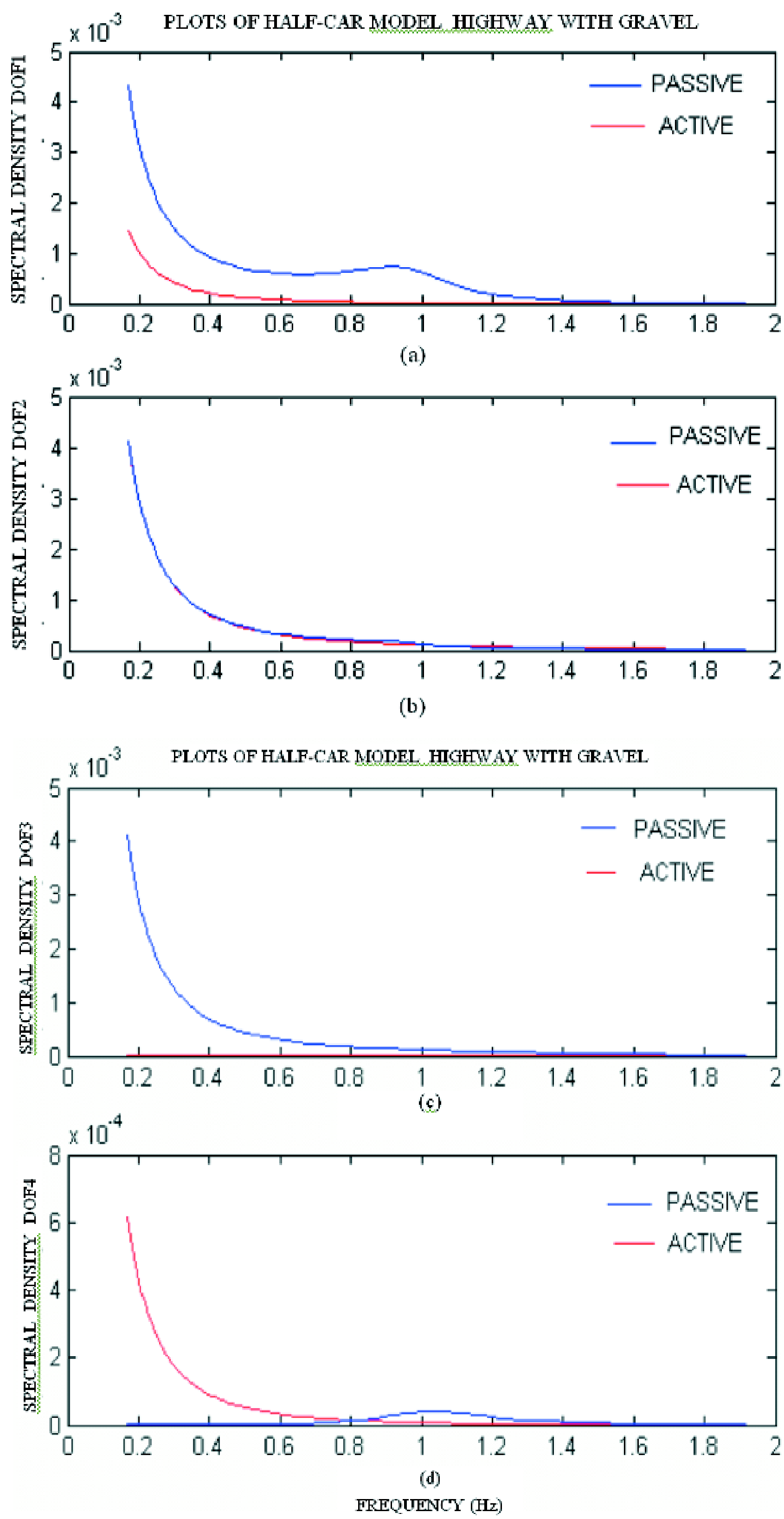

Figure 5. Output spectral density-(a) DOF1, (b) DOF2, (c) DOF3, and (d) DOF4 in terms of displacement.

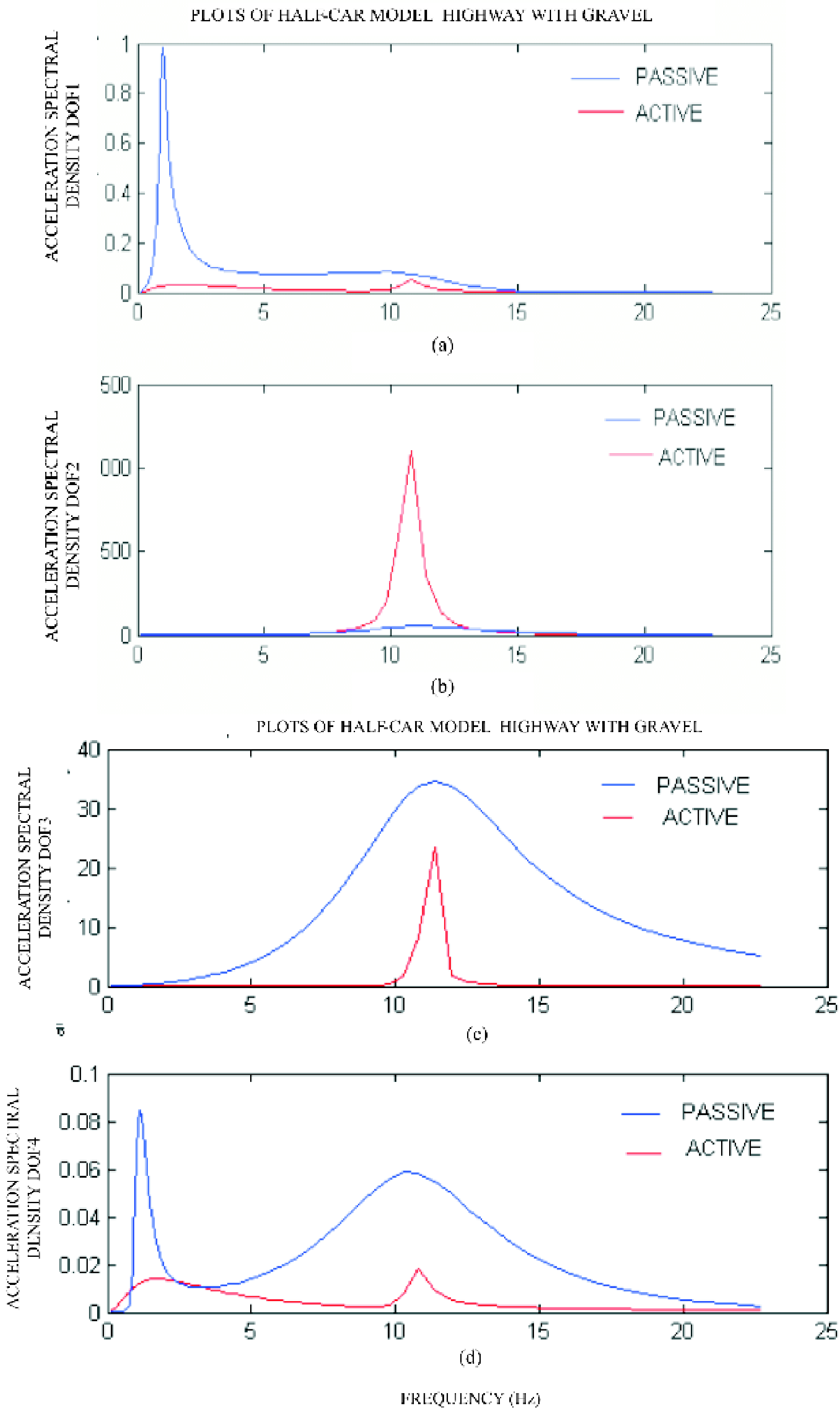

Figure 6. Acceleration spectral density-(a) DOF1, (b) DOF2, (c) DOF3, and (d) DOF4 as a function of frequency.

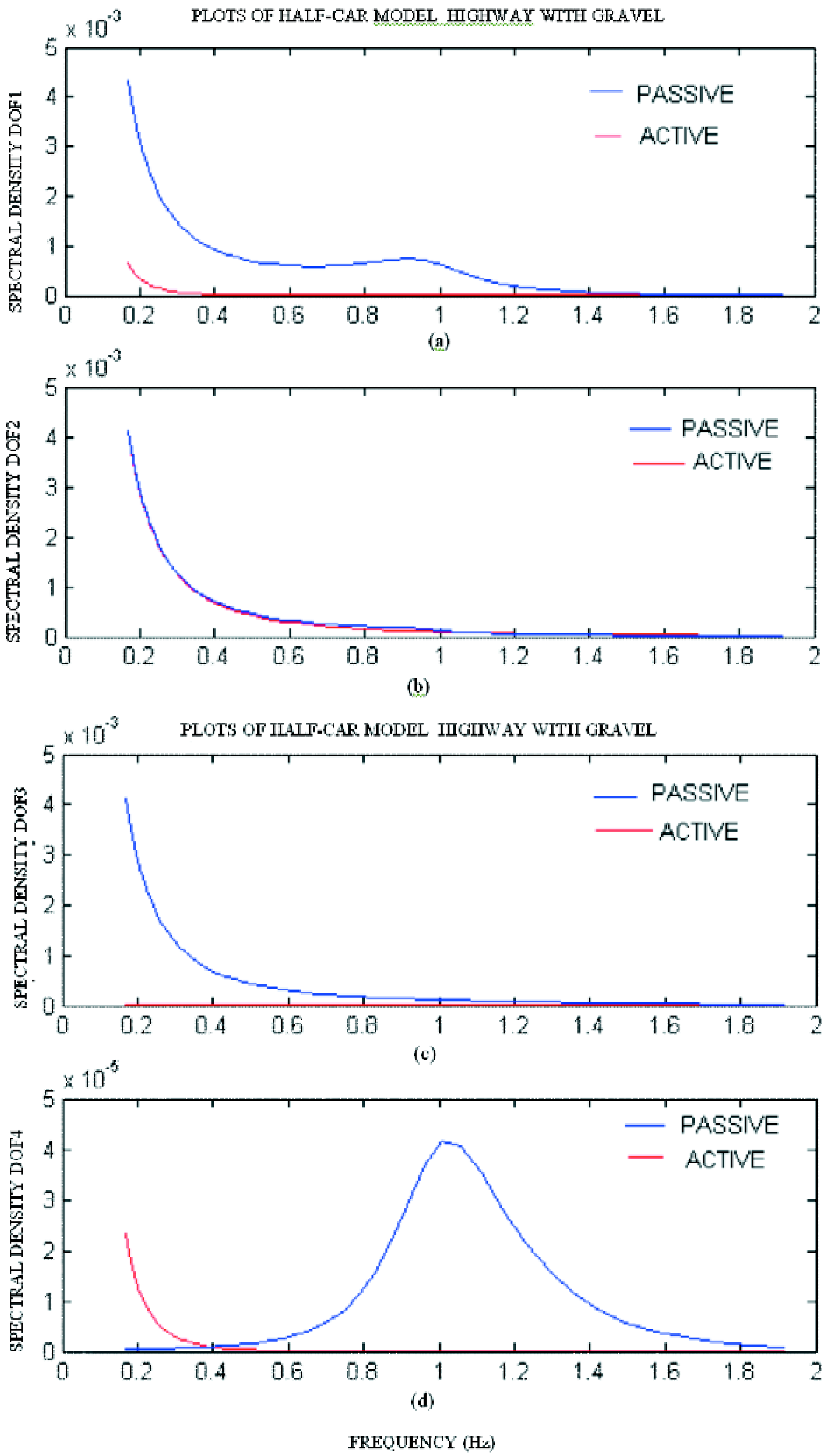

Figure 7. Spectral density-(a) DOF1, (b) DOF2, (c) DOF3, and (d) DOF4 using roulette selection.

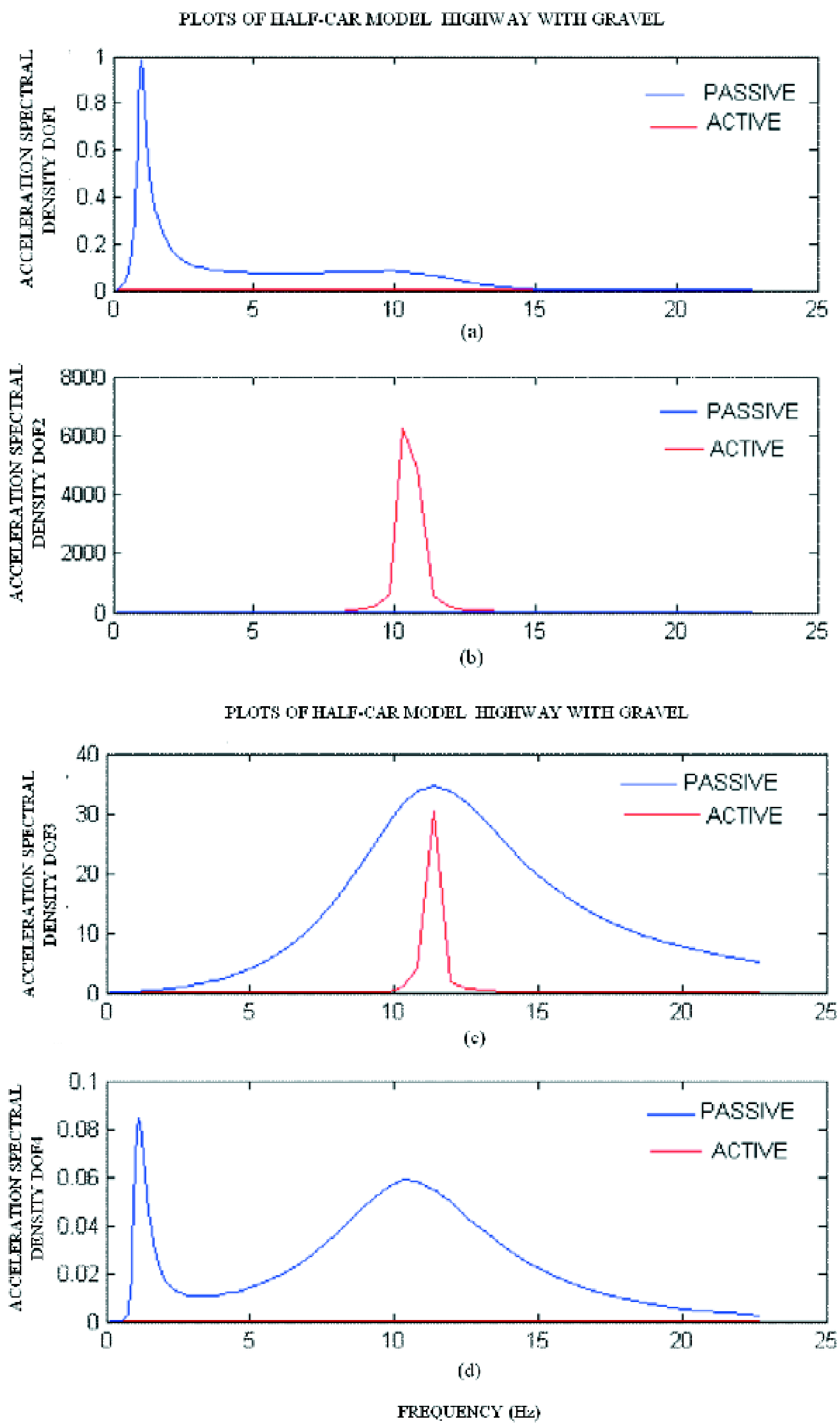

Figure 8. Acceleration spectral density-(a) DOF1, (b) DOF2, (c) DOF3, and (d) DOF4 using roulette selection.

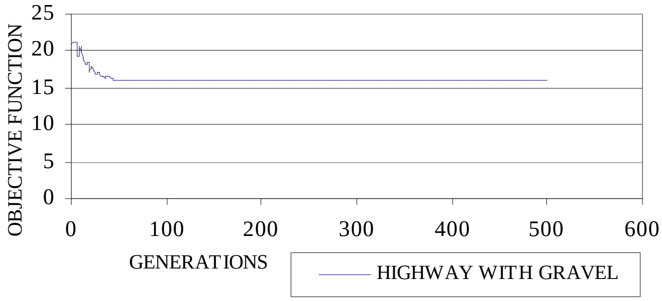

Figure 9. Objective function transition and convergence of case HG2.

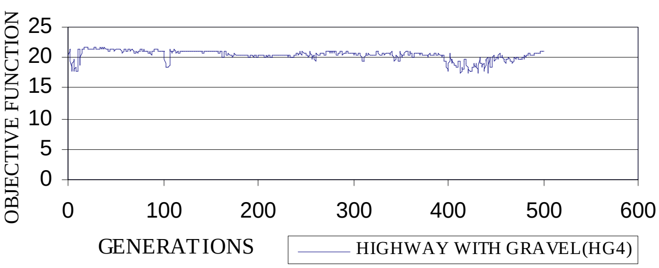

Figure 10. Objective function transition and convergence of case HG4.

more or less uniform, then the response calculations pertinent to design can be considerably simplified.

Genetic algorithms are employed in the design to obtain an effective search for optimum control parameters. Although the genital algorithm method may not result in the optimal OBJ value in the entire search space, its efficiency has been demonstrated in this work since it only uses 50,000 individuals (500 generations * 100 individuals per generation) among 296 possible candidates in the search space. The efficiency of using genetic algorithms is also shown in the rapid convergence of the objective functions using only a few candidates in the search space. Moreover, satisfactory results are obtained from the control scheme simulation, which give a necessary force required for an actuator to provide active control of the car suspension. The flexibility of the control algorithm is shown by the change of weighting factors, which can be used to achieve the required objective function. Thus, the proposed control scheme and the use of genetic algorithm are far more advantageous than the traditional search methods since its search is for global optimum as against local optimum in traditional search methods.

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