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Control of Active Suspension System using Robust H∞ control with Genetic Algorithm

Mohammed Kaleemullah[1], Waleed F. Faris[2*], FariedHasbullah[2],Nouby M. Ghazaly[3]
[1]*Department of Electronics and Communication Engineering, C. Abdul Hakeem College of Engineering and Technology, Vellore, Tamil Nadu, India*
[2]*Faculty of Engineering, International Islamic University Malaysia, Kuala Lumpur, Malaysia*
[3]*Mechanical Engineering Dept., Faculty of Engineering, South Valley University, Qena-83521, Egypt*
**Corresponding Author: waleed@iium.edu.my, mkalim@gmail.com*

Abstract

Better ride comfort and controllability of vehicles are pursued by automotive industries by considering the use of suspension system which plays a very important role in handling and ride comfort characteristics. Comprehensive comparison on half car model was conducted to analyze the effect of active suspension system, using Robust H-infinity on the model. Passive suspension system is also compared with active suspension technique for the purpose of benchmarking. Parametric uncertainties are used to model the non-linearities associated in the system. Genetic Algorithm is used to develop weighting function for robust control design purpose. Comparison of all models also shows that in spite of adding uncertainties in the system, the designed Robust H-infinity controller achieved better settling time than the traditional passive suspension system.

1. Introduction

Much has been said about reducing the polluting emission from automobiles which affects the environment. Perhaps, toxic gases are not the sole type of pollution from vehicles. Noise is another form of pollution which has immense affects on passenger/s in the vehicle. There are many sources that can generate noise, such as, aerodynamic forces, engine, air ducts, and the vibrations transmitted to the car frame due to uneven road profile. In order to reduce engine noise level, work such as, sound proofing the engine compartment and the soft mounting the engine has been done earlier [1][2].

To improve ride comfort of the passengers and to reduce fatigue damage to the various vehicle components, the suspension system is equipped to the vehicle that acts like a cushion. The main objective of the suspension system is to provide superior handling performance, proper ride quality (passenger comfort) and increased road holding ability of the tires. A suspension system thus absorbs the energy exerted by the spring from road profile to the vehicle and dissipates it.

The road disturbance are broadly classified into two types, they are shock and vibration. Shocks are generally termed as suddenly applied inputs such as those from potholes. Potholes are described as discrete events with short duration but high power and vibrations are consistent excitation [3].

The function of a suspension system is to provide sufficient force between the road and tires. Different types of forces such as tractive, cornering, and braking forces are induced by the tires while on motion depending on the wheel position and its motion. All of these forces are related to the vertical force acting between the road and the tires. When the

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suspension is under vibration, the magnitude of the vertical force varies. The road holding ability is assessed by the vibration of tires and it consequently affects the handling performance of the vehicle. The magnitude of the dynamic vertical force should not exceed that of the static load to avoid losing contact with the ground [4].

Handling performance refers to vehicle steering command response to road profile input. Suspension is used to improve the feeling of ride by absorbing vibration of vehicle under uneven state of road surface [5]. Vehicle suspension is also necessary to keep tire contact with the ground, and to keep wheels in appropriate position on road surface [6]. This objective can be achieved by minimizing vertical car body acceleration using suspension system.

Asuspension is normally divided into the following categories depending upon the operating principle: the passive suspension consists of springs and dampers, the semi-active suspension using a variable damper, the active suspension using hydraulic, air, or electric force actuator. Passive suspension is the simplest to design and economically advantage. The main drawback of passive suspension is its limit of suppressing the vibration occurring due to irregular road surface [6]. Semi active suspension gives freedom to vary the damper characteristics along with the road. Anactive suspension has the additional advantage of negative damping and larger range of force can be generated at low velocities [7]. Another benefit of active suspension is that they offer dynamic compensation as compared to passive suspension and various techniques can be used to design control algorithm [8].

[9]developed robust tracking control scheme to improve ride comfort at any specified location by designing two ideal vehicles so that ride comfort becomes best at each different location. [10]designed the whole system linearly except actuator which was designed nonlinearly. Robust H_∞ control method was used to design linear part to achieve robustness and non-linear adaptive based on back-stepping control method was used to design the actuator. [10]considered the error between the desired acting force from H_∞controller and force generated by actuator as the disturbance to the linear system. [11]used pragmatic approach to select the uncertainty and performance weight. [12]considered vehicle inertial properties to design robust controller. Ride comfort is optimized by minimizing H_∞norm, while, road holding of the vehicle and suspension deflection are carried out by constraining the generalized H_2 norm. [13]presented robust control method with MR damper and evaluated the performance in a Hardware-in-the-loop HiL platform to evaluate the performance using Robust H_∞ control. [14]used an input delay approach to change the sampling measurement to continuous time signal with delay in the state. A polytopic parameter uncertainty was used to characterize the uncertain situation and lyapunov functional approach was employed to achieve the H_∞ performance. [15]designedH_∞ control with a three dimensional kinematic model of Macpherson suspension system to study the influence of the control force variation in wheel motion such as wheel performance, toe angle, camber angle and track width were simulated. However, in this study, the passenger ride comfort and car handling were not carried out and were ignored. [16] designed Robust H_∞ controller with ER suspension system subject to parametric uncertainty. Dynamic bandwidths of cylindrical ER damper operating with two different fluids (fast response and slow response characteristics) were identified. [17]used sampled data for H_∞ control to ensure asymptotical stability of the closed loop system with a given level of disturbance attenuation and to satisfy desirable output constraint performance, input delay approach with sampled data H_∞ control were introduced. [18]proposed robust H_∞state feedback quadratic controller from the solution of convex optimization problem. [19]investigated the issue of robust quantized H_∞ control by considering the vehicle load variation. Then required performance of the vehicle suspension

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such as ride comfort, car handling and suspension deflection was transformed with sampling and quantization measurements into a continuous time system with input delay and sector bound uncertainties by using input delay method.

During the last decade of the twentieth century, many researchers explored non-linearities of suspension system which requires the use of non-linear model and non-linear control scheme [20-21]. These non-linear models make active suspension control system too complex and challenging to employ in practice. In application, engineers often have to deal with complex systems having multiple parameter models with possible non-linear coupling. The linear way of modeling such a system for predicting its behavior based on analytical techniques can be proved to be inadequate even at the initial stages of mathematical modeling. It is evident that one has to deal with high degree of uncertainty in real time systems [22]. Thanks to robust control in dealing with system uncertainties which include unmodelled lags (time delay), parasitic coupling, hysteresis and other non-linearities. Such perturbation represents variation of particular system parameters over some possible range [23]. In this paper, we have designed Robust H_∞synthesis with system uncertainty to study the performance of a half car system using Genetic Algorithm. These active suspension techniques were further compared with passive suspension system.

2. Vehicle Modeling

A model generally represents an approximation of the actual physical system which can be modelled in different ways. However, a good system model must include all the important dynamic characteristics of the system so that the response achieved by the model could satisfactorily match the behaviour of the actual system.

Modelling in vibration can be categorized into two types – lumped parameter system (discrete system) and distributed parameter system (continuous system). In this work, discrete modelling of system is considered which describes vehicle as lumped mass or finite degree of freedom system.

To study ride quality, vehicle mass is usually separated into two – sprung mass (vehicle body) and unsprung mass (vehicle wheel). Unsprung mass includes mass of tire, brakes, suspension linkages and other mass associated with the wheel. This part of the vehicle is on the roadside and thus reacts to irregular road profile with no damping.

The two axle four degree-of-freedom model used in this study is shown in the Figure 1**Error! Reference source not found.**. The four DOFs are, sprung mass vertical displacement (x_s), sprung mass pitch (θ), front unsprung mass displacement (x_{uf}) and rear unsprung mass displacement (x_{ur}). One of the inputs to the considered linear model is the displacement which represents a typical road profile. This input from the road surface excites the unsprung mass which corresponds to suspension components, wheel and tire. The unsprung mass is connected to the sprung mass which represents the body of vehicle through spring, damper and actuator. The equation of motion of the half car system can be derived as

$$m_s\ddot{x}_s + k_{sf}(Z_f) + k_{sr}(Z_r) + c_{sf}(\dot{Z}_f) + c_{sr}(\dot{Z}_r) - f_f - f_r = 0 \quad (1)$$

$$I_{yy}\ddot{\theta} - k_{sf}(Z_f)a + k_{sr}(Z_r)b - c_{sf}(\dot{Z}_f)a + c_{sr}(\dot{Z}_r)b + f_f a - f_r b = 0 \quad (2)$$

$$m_{uf}\ddot{x}_{uf} - k_{sf}(Z_f) - c_{sf}(\dot{Z}_f) + k_{tf}(x_{uf} - x_{rf}) + f_f = 0$$

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$$m_{ur}x_{ur} - k_{sr}(Z_r) - c_{sr}(\dot{Z}_r) + k_{tr}(x_{ur} - x_{rr}) + f_r = 0 \tag{4}$$

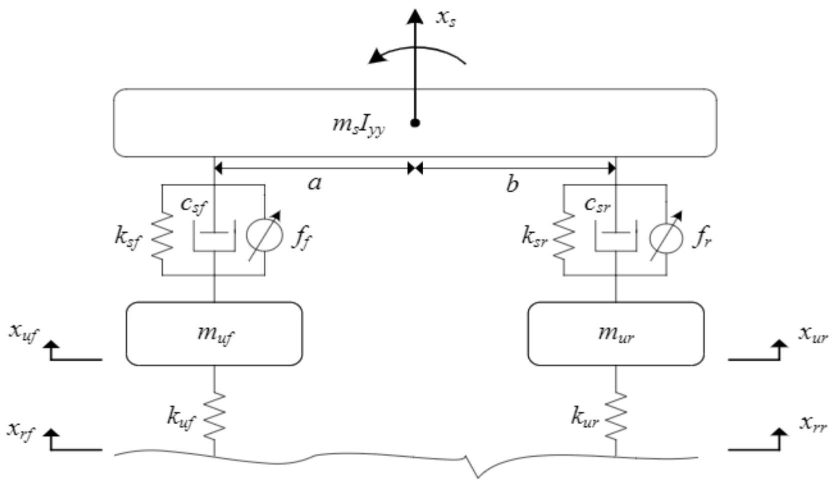

Figure 1. FourDoF half car model

where

$Z_f = x_s - a\theta - x_{uf}$

$Z_r = x_s + b\theta - x_{ur}$

These equations of motion can be described in state space form as

$$\dot{X} = Ax + Bu + Gw \tag{5}$$

where state vectors x represents the controlled input, u represents the actuator input and w represents the road disturbance,

$x = [x_s \quad \theta \quad x_{uf} \quad x_{ur} \quad \dot{x}_s \quad \dot{\theta} \quad \dot{x}_{uf} \quad \dot{x}_{ur}]^T$

$u = \begin{bmatrix} f_f \\ f_r \end{bmatrix}$

$w = \begin{bmatrix} x_{rf} \\ x_{rr} \end{bmatrix}$

The interested output state variable are sprung mass vertical acceleration $(\ddot{x}_s)$, sprung mass pitch angular acceleration $(\ddot{\theta})$, front suspension deflection $(x_s - a\ddot{\theta} - x_{uf})$, rear suspension

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deflection $(x_s + b\ddot{\theta} - x_{ur})$, front tire deflection $(x_{uf} - x_{rf})$ and rear tire deflection $(x_{ur} - x_{rr})$.

The output of the system is written as

$$y = Cx + Du + Hw \tag{6}$$

The state variable descriptions and vehicle nominal model parameters values are provided in
Table 1 and Table 2 respectively.

Table 1: State variables and input description for half car

Symbol	Parameters
x_s	m_s displacement (m)
θ	m_s pitch (rad)
x_{uf}	m_{uf} displacement (m)
x_{ur}	m_{ur} displacement (m)
x_{rf}	Front input (m)
x_{rr}	Rear input (m)

Table 2. Model parameters of half car

Symbol	**Description**	**Nominal Value**	**unit**
m_s	Sprung mass	730	kg
I_{yy}	Pitch moment of inertia	2460	kgm^2
m_{uf}	Front unsprung mass	40	kg
m_{ur}	Rear unsprung mass	40	kg
a	Distance from ms CG to front	1.011	m
b	Distance from ms CG to rear	1.803	m
k_{sf}	Front suspension stiffness	19960	N/m
k_{sr}	Rear suspension stiffness	17500	N/m
k_{tf}	Front tire stiffness	175500	N/m
k_{tr}	Rear tire stiffness	175500	N/m
c_{sf}	Front suspension damping coefficient	1290	N s/m
c_{sr}	Rear suspension damping coefficient	1620	N s/m

3. Disturbance Modeling

As discussed earlier, vehicle ride and handling are influenced mainly by two sets of disturbance. One is caused by different forces that originate due to braking, turning and wind

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gusts for instance and the other is due to road roughness. The more significant of the two types is the input disturbance from road surface.

The input from road surface can be broadly classified as shock and vibration. Shocks are nothing but discreet event of relatively short duration but high intensity, such as the disturbance caused by a bump or pothole on a smooth road profile. However, vibrations are characterized by prolonged and consistent disturbance that are felt on rough road [24]. These geometric irregularities of the road play a major role in causing vehicle vibration, which directly influence vehicle wear, ride comfort and safety. In order to understand the seriousness of uneven surface, road profile and its roughness have to be measured and classified before the analysis. The designers develop vehicles for varied application by referring to the condition of the road [25].

3.1. Random Disturbance

Road roughness is an important factor as an indicator of road condition in terms of road pavement performance and as a determinant of road user cost. Road profiles fit in the category of "broad-band random signals" and thus can be described by both as a profile or its statistical properties. Power Spectral Density (PSD) is highly used to represent random road disturbance. Nevertheless, other properties of random signals such as stationary and ergodic are also used to describe road profile.

It is found that the relation between the power spectral density and the spatial frequency for the road profile can be approximated by

$$S_g(\Omega) = C_{sp}\Omega^{-N} \tag{1}$$

where, $S_g(\Omega)$ is the PSD of the elevation of the surface profile, and C_{sp} and N are constants. A typical measured spectral densities of various terrains by [3] using the above Equation (1) is shown in the Figure 2, with its corresponding values in Table 3.

Over the years, various organizations have made attempts to classify the road irregularities. The international Organization for Standardization (ISO) has proposed road irregularities classification based on power spectral density (PSD) as shown in Table 4 and Figure shows the classification of roads proposed by ISO [26].

Table 3: Values of C_{sp} and N for PSD for various surfaces [3]

Description	***N***	C_{sp}
Smooth runway	3.8	4.3×10^{-11}
Rough runway	2.1	8.1×10^{-6}
Smooth highway	2.1	4.8×10^{-11}
Highway with gravel	2.1	4.4×10^{-11}
Pasture	1.6	3×10^{-4}
Plowed field	1.6	6.5×10^{-4}

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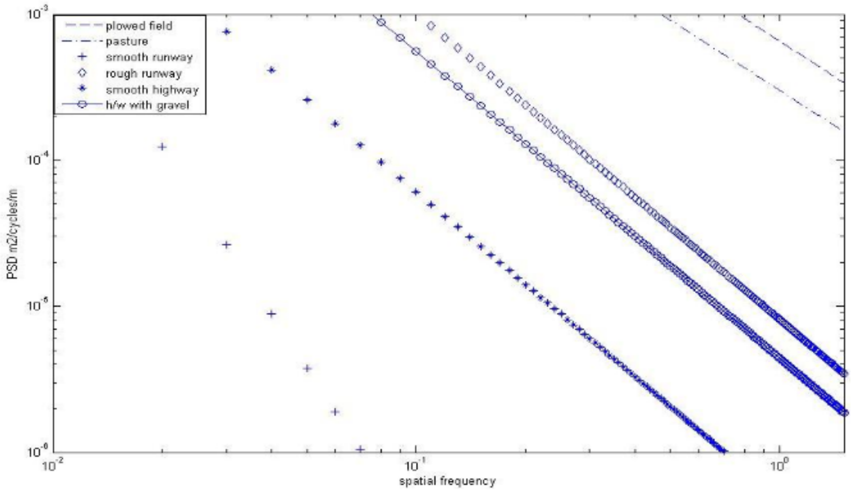

Figure 2: Power Spectral Density for various types of road and runways

The relationship between PSD $S_g(\Omega)$ and the spatial frequency Ω for different road roughness can be approximated by

For $\Omega \leq \Omega_o = 1/2\pi$ cycles/m,

$$S_g(\Omega) = S_g(\Omega_o)\,(\Omega/\Omega_o)^{-N1} \tag{2}$$

For $\Omega > \Omega_o = 1/2\pi$ cycles/m,

$$S_g(\Omega) = S_g(\Omega_o)\,(\Omega/\Omega_o)^{-N2} \tag{3}$$

Table 4, shows the range of values of $S_g(\Omega_o)$ at a spatial frequency $\Omega_o = 1/2\pi$ cycles/m for different classes of road. The values of N_1 and N_2 are 2 and 1.5 respectively.

Table 4: ISO classification of road roughness

	Degree of roughness $S_g(\Omega_o)$, 10^{-6} m^2/cycles/m	
Road Class	Range	Geometric Mean
A (very good)	<8	4
B (good)	8 - 32	16
C(average)	32 - 128	64
D (poor)	128 - 512	256
E (very poor)	512 - 2048	1024
F	2048 - 8192	4096
G	8192 - 32,768	12288

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H	>32,768	16384

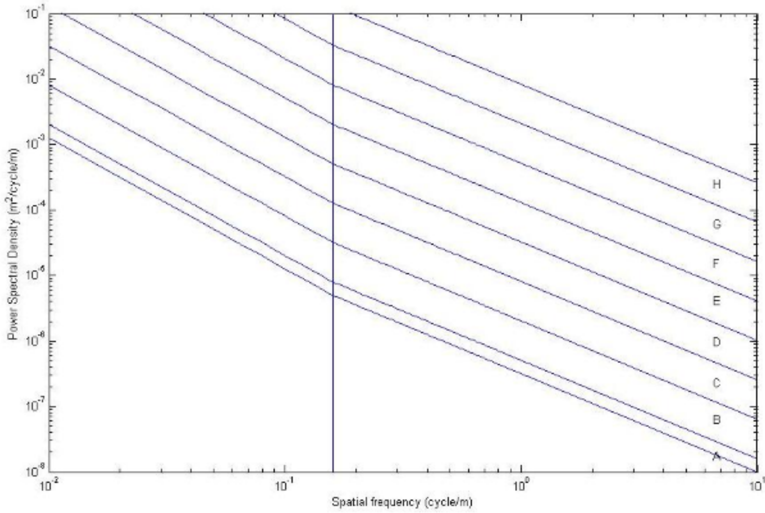

Figure 3: classification of road surface roughness by ISO

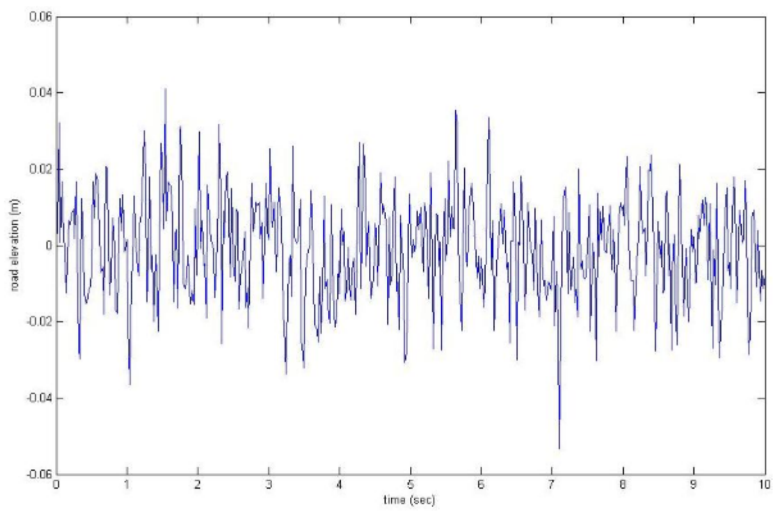

Figure 2: Random surface profile

Excitation from ground surface as shown in Figure 2 can be described using random disturbance more realistically. Equation (10), is used to describe random profile [27],

$$\dot{z}_r(t) + w_o z_r(t) = \sqrt{S_g(\Omega_o)u}.\, w(t) \tag{4}$$

Where $z_r(t)$ is the road profile, $w(t)$ is a white noise, w_o is equal to $0.2\pi u$, with u as the velocity of the vehicle and $S_g(\Omega_o)$ is the road roughness.

3.2. Pothole Input

A further study is performed on a more common road disturbance like potholes. Usually, this kind of disturbance is commonly faced by passenger on roads. The pothole disturbance used in this study is based on the pothole track at the Gerotek test facilities in South Africa which is 80 mm deep in the road[28]. It is assumed that the vehicle is travelling

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at a constant velocity of 80 km/h. In the Figure 3, front disturbance is represented using line, while the dashed lined represents rear input after certain time delay.

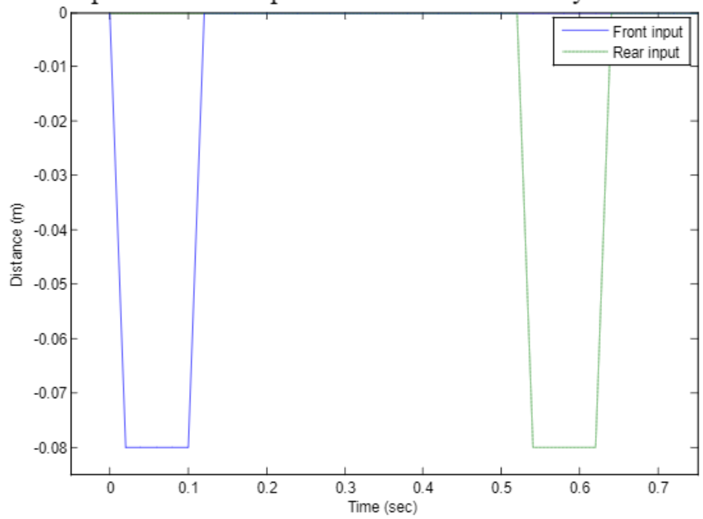

Figure 3: Pothole disturbance input

4. Uncertainty Modeling

Any mathematical representation of a physical system needs approximation which directly leads to model uncertainties. Due to the presence of uncertainties, the nominal controller may not stabilize or achieve the required performance of the actual system. Hence, these uncertainties are necessary when designing the controllers.

Usually, an uncertain model is described by a set of perturbed nominal model to describe the inherent uncertainty of the system under consideration. The uncertainty can be classified as structured (parametric) and unstructured (additive, multiplicative, inversed multiplicative, inverse additive etc) [29]. In this research, parametric and multiplicative uncertainties are considered for the plant and actuator respectively.

4.1. Model Uncertainty

In realistic system, where the system to be controlled is not completely known, it is required to design a controller that can achieve some given performance specifications. Usually, in classical approach, an approximate mathematical model of the system is developed based on linear time-invariant and finite dimensional models. In this case, their mathematical model is set of linear differential equations that are accurate for some degree of certainty. The main disadvantage of this kind of model is that the achieved performance of the system may not meet the design requirement and may lead to closed-loop instability.

The two main reason to include uncertainties in the model are [30]

- The behavior of the real systems cannot be represented by the class of model used.
- Always there exists uncertainty in the numerical values of various parameters of the model.

The sources of the uncertainties in a system can be summarized as [31]

1. The linear model may vary due to the presence of nonlinearities like friction, eccentricity, saturation and hysteresis or changes in operating conditions.

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2. Some parameters in the linear model are only known approximately or contain some errors which lead to inaccurate model.
3. Errors due to noise in measuring devices.
4. The models are unknown at high frequencies. The uncertainty will exceed 100% at certain frequency.
5. It is appropriate to work with simpler nominal model and represent the neglected dynamics as uncertainties.

For instance, mass of the body varies significantly with and without passenger, which is very uncertain in nature. The uncertainty of the wheel mass, spring stiffness and damping coefficient should also be considered. However, it is assumed that their values are within certain known intervals.

$$m_s = \bar{m}_s (1 + p_{ms}\,\delta_{ms}) \tag{15}$$

$$m_{uf} = \bar{m}_{uf} (1 + p_{muf}\,\delta_{muf}) \tag{16}$$

$$m_{ur} = \bar{m}_{ur} (1 + p_{mur}\,\delta_{mur}) \tag{17}$$

$$c_{sf} = \bar{c}_{sf} (1 + p_{csf}\,\delta_{csf}) \tag{18}$$

$$c_{sr} = \bar{c}_{sr} (1 + p_{csr}\,\delta_{csr}) \tag{19}$$

$$k_{sf} = \bar{k}_{sf} (1 + p_{ksf}\,\delta_{ksf}) \tag{110}$$

$$k_{sr} = \bar{k}_{sr} (1 + p_{ksr}\,\delta_{ksr}) \tag{111}$$

$$k_{uf} = \bar{k}_{uf} (1 + p_{kuf}\,\delta_{kuf}) \tag{112}$$

$$k_{ur} = \bar{k}_{ur} (1 + p_{kur}\,\delta_{kur}) \tag{113}$$

where, $m_s, m_{uf}, m_{ur}, c_{sf}, c_{sr}, k_{sf}, k_{sr}, k_{uf}, k_{ur}$ are uncertain parameters and $\bar{m}_s, \bar{m}_{uf}, \bar{m}_{ur}, \bar{c}_{sf}, \bar{c}_{sr}, \bar{k}_{sf}, \bar{k}_{sr}, \bar{k}_{uf}, \bar{k}_{ur}$ are its corresponding nominal values. p_{ms}, p_{muf}, p_{mur}, p_{csf}, p_{csr}, p_{ksf}, p_{ksr}, p_{kuf}, p_{kur}, and δ_{ms}, δ_{muf}, δ_{mur}, δ_{csf}, δ_{csr}, δ_{ksf}, δ_{ksr}, δ_{kuf}, δ_{kur} represents possible perturbation on the parameters listed in Table 5. In this study, $\delta_{ms}, \delta_{muf}, \delta_{mur}, \delta_{csf}, \delta_{csr}, \delta_{ksf}, \delta_{ksr}, \delta_{kuf}, \delta_{kur} \leq 1$.

Values considered to represent uncertainty in terms of percentage to individual parameters are listed in
Table 5.

Table 5: Perturbation uncertainty values

Perturbation	Uncertainty (%)
p_{ms}	30
p_{muf}	5

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p_{mur}	5
p_{csf}	20
p_{csr}	20
p_{ksf}	30
p_{ksr}	30
p_{kuf}	30
p_{kur}	30

5. Controller Design

5.1. H-infinity Control Theory

Robust H_∞technique with nominal model and modelling uncertainty is considered due changing system parameters owing.

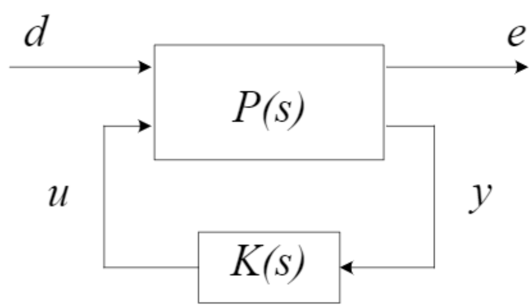

Figure 6: Robust controller K(s)

In Figure6*P(s)* is the generalized plant, *d* denotes all external disturbance, output signal is taken as error *e* which is to be minimized, *y* is the input to controller *K* and *u* is the vector of controlled signals which is also to be reduced to avoid the saturation of the actuator.

State space equation in the form as shown in Eq. (20) is used to represent the system shown in Figure6.

$$\begin{aligned} \dot{x} &= Ax + B_1 d + B_2 u \\ e &= C_1 x + D_{11} d + D_{12} u \\ y &= C_2 x + D_{21} d + D_{22} u \end{aligned} \tag{20}$$

The above state space representation can be written in matrix form as,

$$P(s) = \begin{bmatrix} A & B_1 & B_2 \\ C_1 & 0 & D_{12} \\ C_2 & D_{21} & 0 \end{bmatrix} \tag{21}$$

By taking lower LFT for the plant matrix *P(s),*

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$$F_L(P,K) = P_{11} + P_{12}K(I - P_{22}K)^{-1}P_{21} \tag{22}$$

The goal of H-infinity controller is to find a stabilizing controller *K(s)* to minimize H-infinity norm of the closed loop transfer function is less than a given positive number, i.e.

$$\|T_{ed}\|_\infty = \|F_L(P,K)\|_\infty \tag{23}$$

We may obtain an optimal solution by successively reducing the value of γ from a large number [32]. It should be taken care that reducing γ more than the nominal range makes the solution unreliable [33]. Given an acceptable value of γ, we can compute a controller K such that,

$$\|F_L(P,K)\|_\infty < \gamma \tag{24}$$

The H_∞ controller can be found using two Ricatti equations by solving it iteratively [34].

$$A^T X_\infty + X_\infty a + C_1^T C_1 - \gamma^{-2} X_\infty B_1 B_1^T X_\infty - X_\infty B_2 B_2^T X_\infty = 0 \tag{25}$$

$$AY_\infty + Y_\infty A^T + B_1 B_1^T - \gamma^{-2} Y_\infty C_1^T C_1 Y_\infty - Y_\infty C_2^T C_2 Y_\infty = 0 \tag{26}$$

whereX_∞ and Y_∞ are the optimal solution of the Ricatti equations. In the Figure 7, $\Delta(s)$ represents the uncertain parameters of the system.

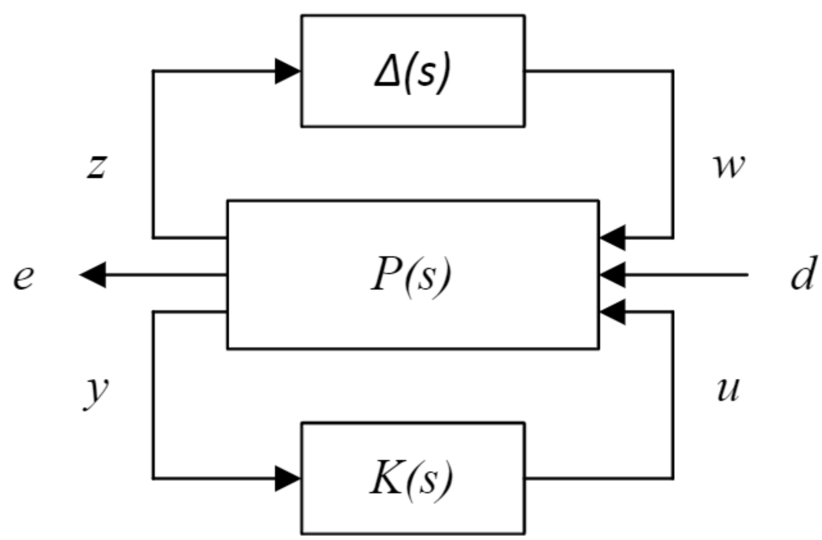

Figure 7: System with uncertainty

5.2. Weighting Function

Selecting an appropriate weighting function is one of the most important steps in the robust controller design. The weighting function W_d, W_n, W_p, W_a are included for the following reasons [35].

- To avoid saturation of the actuator W_a by constraining the magnitude of the input signal.
- To ensure good closed loop performance specification W_p

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- To signify the frequency content of external disturbances and noises such as W_d and W_n respectively.

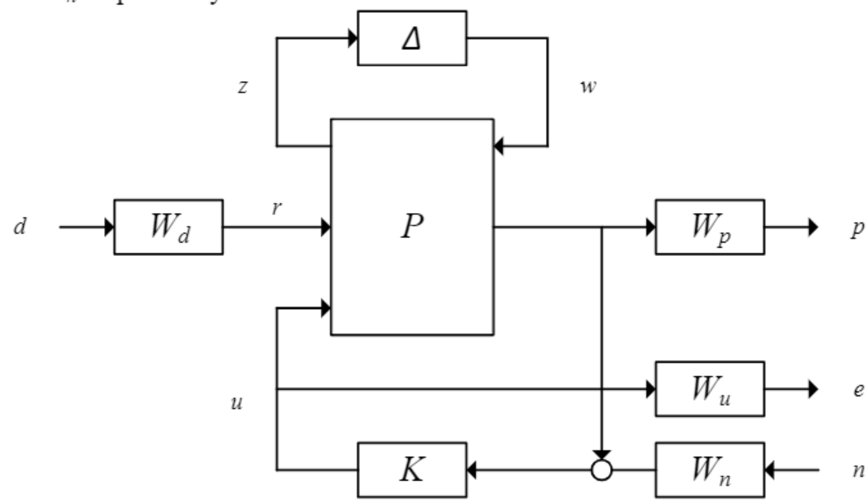

Figure 8. Closed loop system with uncertainty

It should be noted that the amplitude of the disturbance signal does not correspond to the actual external disturbance affecting the system. For instance, in figure (8), the norm of $\widetilde{d}$ must be less than or equal to 1 and the norm of d is less or equal to the maximum amplitude of the disturbance which may occur from improper road surface.

Sensor noise Wn= 0.02 is chosen for sprung mass vertical acceleration and pitch acceleration thus representing that the sensor noise is 0.02m/s^2. It is assumed that the sensor noise for the suspension deflection is 0.001 m/s^2. The front and rear weighting function for road disturbance are chosen as 0.05. Closed loop model of the system after adding uncertainty and weighting function is shown in figure (8).

6. Genetic Algorithm

Genetic Algorithm (GA) toolbox was used to find the actuator weighting function W_{af}, W_{ar} for front and rear actuator respectively and performance weighting function W_{p1} to W_{p4} for body vertical acceleration, pitch acceleration, front suspension deflection and rear suspension deflection respectively.

The idea behind this is that the GA creates series of new population of individuals or chromosomes in the existing generation that are used to produce the next population in subsequent steps:

a. Scores all individual member of the existing population by computing its fitness value and scales the raw scores to convert them into a more practical range of values.
b. Selects parents based on their fitness values.
c. Perform elitist selection, through which some of the better member of the current population are allowed to carry over to the next population unchanged.
d. Children are produced either by mutation or crossover.
e. Replaces the current population with that of children to form the next generation.

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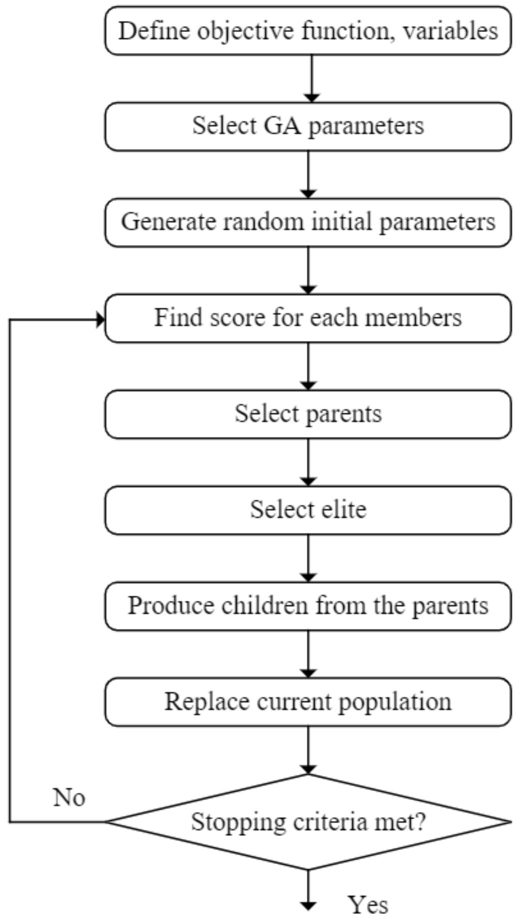
Figure 9. Flow chart of the algorithm



The algorithm stops when one of the criteria is met. The performance and actuator weighting functions for robust control are chosen as the individual to be optimized. Figure 9, shows flow chart of the algorithm.

6.1. Optimization Problem Formulation

To improve performance of the vehicle, the objective function was set to reduce sprung mass vertical acceleration, $\ddot{x}_s$ of the vehicle.

$$y_{ptp} = \max\big(y(t)\big) - \min\big(y(t)\big) \tag{27}$$

As the performance of the active suspension system is evaluated with passive system, the following constraints are considered

$$g_i = \frac{y_{a,i}}{y_{p,i}} \leq 1, i = 1,2,3, \ldots, n \tag{28}$$

where
y_a = output of active suspension system
y_p = output of passive suspension system
n = number of outputs

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By using the penalty approach, the constrained optimization problem is converted into unconstrained optimization problem. The modified objective for this approach is

$$f(1) = \ddot{x}_s + \rho \tag{29}$$

whereρ is the penalty and is given by

$$\rho = \begin{cases} \sum_{i=2}^{n} g_i, & if\, g_i > 1 \\ 0, & otherwise \end{cases} \tag{30}$$

7. Results and Discussion

7.1. Pothole Disturbance

The result of vehicle body vertical acceleration in figure (10) shows that Robust H-infinity control improves the settling time performance by 50.4% when compared to passive system. Similarly, result of vehicle pitch acceleration in figure (11) shows that;robust controller improves PTP and settling time performance by 20.3% and 56.4% when compared with passive system respectively. Result from sprung mass vertical acceleration and pitch acceleration shows that the robust control improves ride comfort of the passenger.

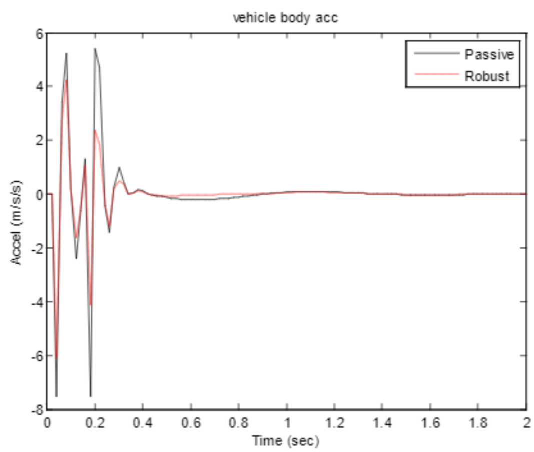

Figure 10. Sprung mass vertical acceleration

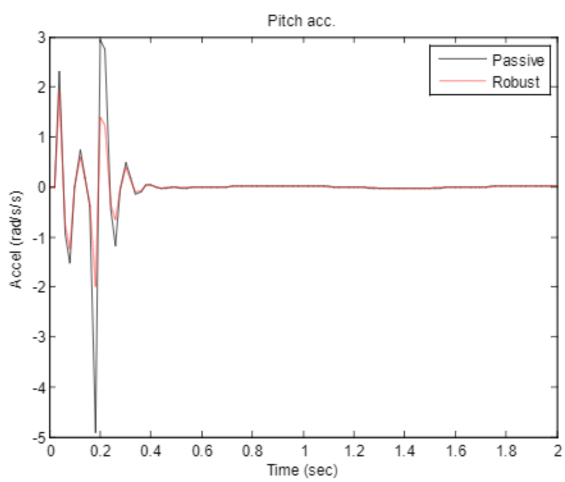

Figure 11. Sprung mass pitch acceleration

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PTP and settling time of the front suspension deflection show that robust controller performs 22.8% and 70.4% better than passive system in terms of PTP and settling time respectively. Similarly, PTP value and settling time of rear suspension deflection performance show that robust control performance is better than passive system. Overall performance of suspension deflection shows passive system has the worst result. It evident from this that, active controllers improves road holding of the car compared to passive system.

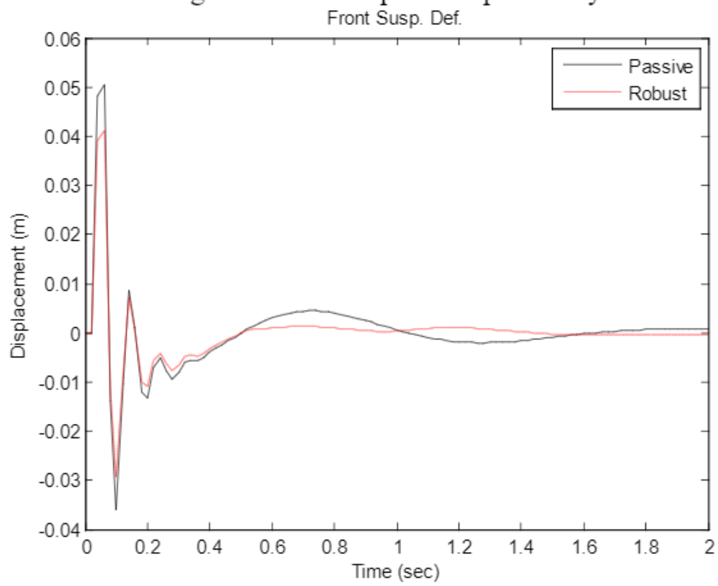

Figure 12. Front suspension deflection

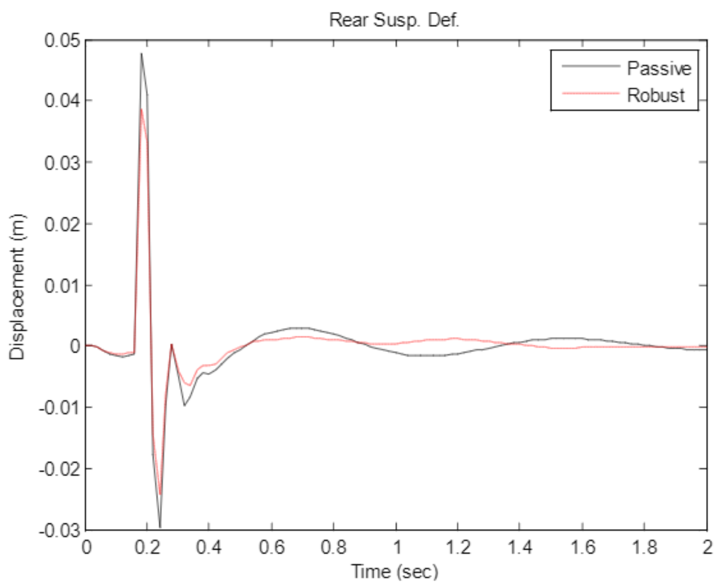

Figure 13. Rear suspension deflection

Front and rear tire deflection is shown in fig (14) and fig(15). The result proves that even in this case robust control outperforms passive system thus improving road holding of the car.

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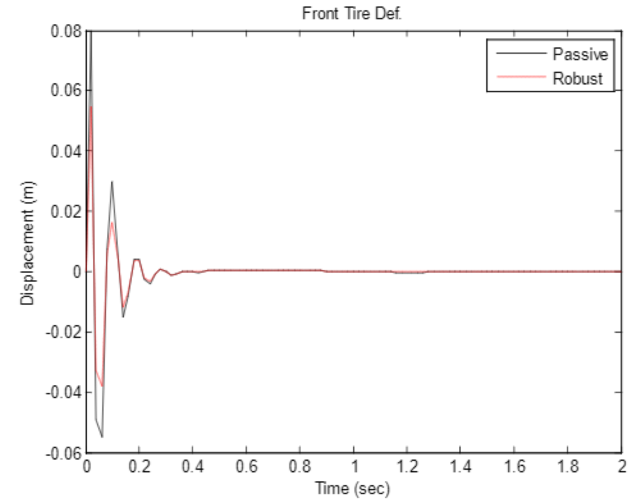

Figure 14. Front tire deflection

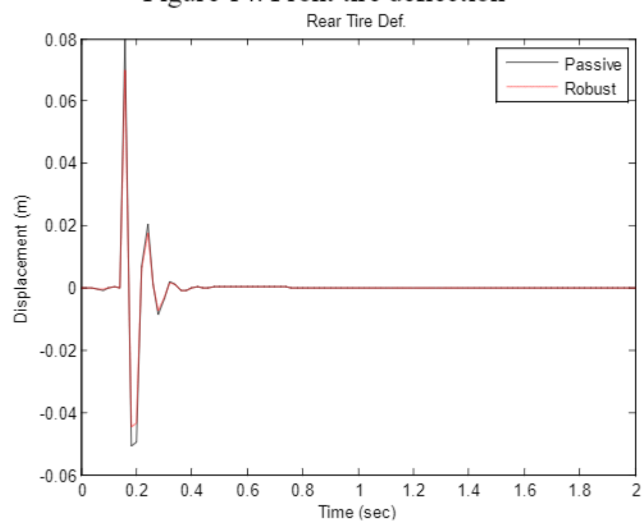

Figure 15. Rear tire deflection

7.2. Random Disturbance

The RMS value for sprung mass acceleration to random input is shown in fig (16) and fig (17). The results shows that robust control gives best response for both vertical acceleration and pitch acceleration with 60.7% and 53.3% increase respectively over passive.

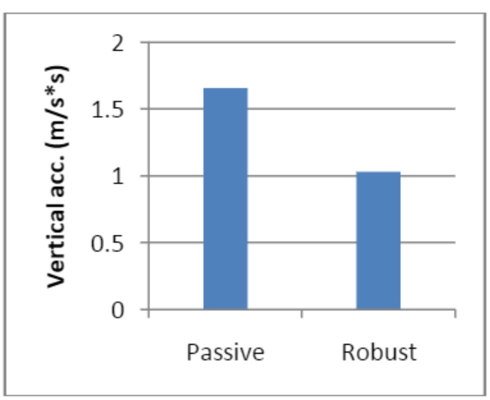

Figure 16. Sprung mass vertical acceleration

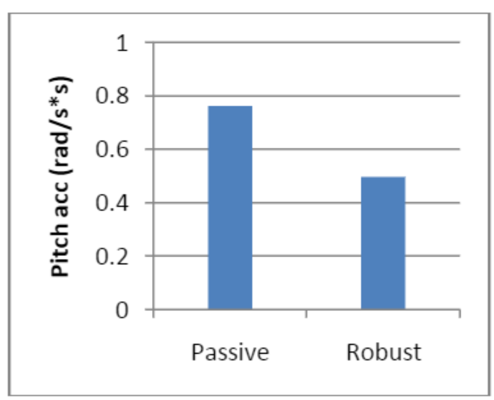

Figure 17. Sprung mass pitch acceleration

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Fig (18) and fig (19) shows the RMS value of front and rear suspension deflection respectively. In front suspension deflection, robust shows 7.7% improvement over passive system. In rear suspension deflection, performance improvement achieved by robust over passive system is 8.3%. The summary of RMS value of front and rear tire deflection subject to random input is shown in fig (20) and fig (21). Again, robust control shows significant performance improvement over passive system.

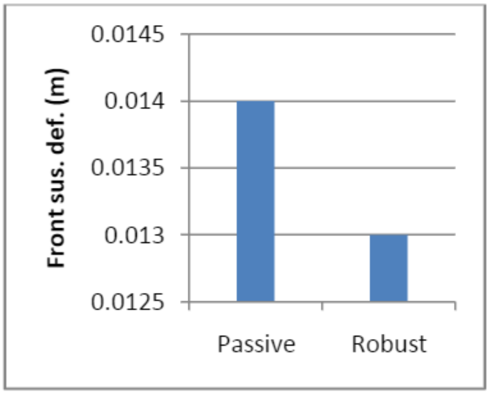

Figure 18. Front suspension deflection

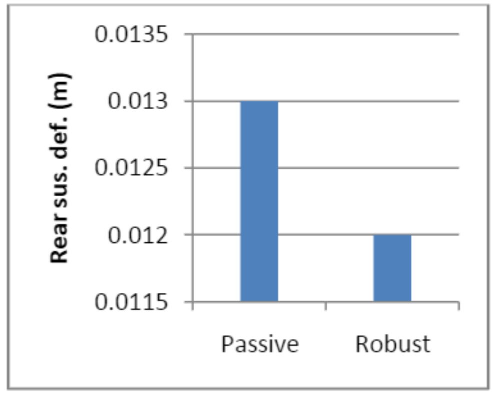

Figure 19. Rear suspension deflection

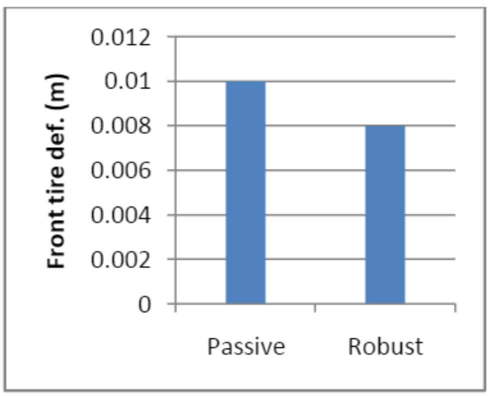

Figure 20. Front tire deflection

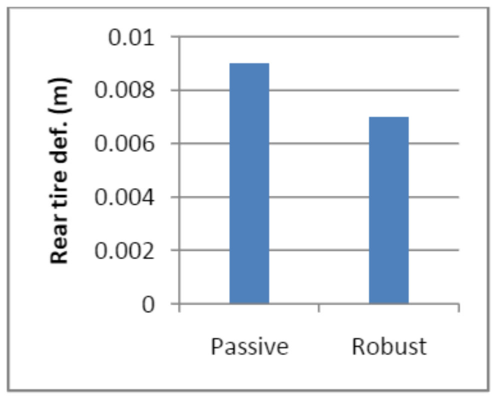

Figure 21. Rear tire deflection

8. Conclusion

Comparison on half car model was conducted to analyze the effect of active suspension system, using Robust H-infinity on the model. Passive suspension system is also compared with active suspension technique for the purpose of benchmarking. Parametric uncertainties are usedto model the non-linearities associated in the system. Genetic Algorithm is used to develop weighting function for robust control design purpose. Comparison of all models also shows that in spite of adding uncertainties in the system, the designed Robust H-infinity controller achieved better settling time than the traditional passive suspension system.

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