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Optimal Design of Robust Controller for Active Car Suspension System Using Bee's Algorithm

Abdoljalil Addeh[a*], Abolfazl Ebrahimi[b]

[a] *Faculty of Electrical and Computer Engineering, Babol University of Technology, Babol, Iran*
[b] *Department of Electrical Engineering, Sirjan Branch Islamic Azad University, Sirjan, Iran*

Keywords

Suspension,
H-infinity,
Quarter-car Model,
Optimization.

Abstract

Vehicle suspension plays a critical role in improving car's ride comfort and handling ability. Active suspension can potentially improve the driving maneuverability and passenger's comfort simultaneously, thanks to the advanced control algorithm and improved actuators. In this paper an intelligent control system based on H-infinity is proposed for active car suspension system. The proposed system includes two main modules: the controller module and the optimization module. The H-infinity control method is one approach to robust control that has been receiving much attention recently in the controls community. In controller module, we used H-infinity controller. To effectively use the H-infinity, one must be able to choose reasonable performance and uncertainty weights. There is no systematic way to select these transfer functions. Therefore in the optimization module of proposed method, we used bee's algorithm (BA) to find the best values of the transfer function coefficients. To evaluate the performance of proposed method, we used quarter-car model and the obtained results show that the proposed method has excellent performance.

1. Introduction

The suspension system is the main tool to achieve ride comfort and drive safety for a vehicle [1- 3]. Passive suspension systems have been designed to obtain a good compromise between these objectives, but intrinsic limitations prevent them from obtaining the best performances for both goals [4, 5]. Compared with passive suspension systems, active and semi-active suspension systems can achieve a better compromise during various driving conditions [6]. Passive shock absorbers have a fixed damping characteristic determined by their design. Depending on the road excitation, however, it is desirable to adjust this characteristic to increase performance. Semi-active and active suspension systems offer the possibility to vary the dampercharacteristics along with the road profile e.g. by changing the restriction of one or two current-controlled valves or by changing the viscosity of a magneto-rheological fluid [7]. An active shock absorber has the additional advantages that negative damping can be provided and that a larger range of forces can be generated at low velocities, thereby potentially improving system performance. Semi-active suspensions on the other hand are less complex, more reliable and commercially available. They do not require an external power source (e.g. hydraulic pump) and are more robust because they can only dissipate energy and therefore cannot render the system unstable [8, 9].

Literature describes several linear and non-linear techniques to control the active or semi-active suspension of a car. Butsuen [10], Karnopp and Crosby [11] and Esmailzadeh and Taghirad [12] apply linear control strategies based on linear physical car models consisting of lumped masses, linear springs and dampers and a shock absorber modeled as an ideal force source. Real car dynamics are much more complex and active shock absorbers are not ideal force sources but have a complex non-linear dynamic behavior. These unrealistic assumptions make these linear control approaches less appropriate for practical applications.

Non-linear control strategies such as linear parameter-varying gain scheduling [13, 14], back-stepping [15] and adaptive control [16, 17] have been applied to active suspension systems. These controllers are based on a nonlinear physical car and damper model which have a large number of parameters. The experimental identification of these model parameters is a complex problem. In addition, the design and tuning of the above-mentioned non-linear controller is not straightforward. Basically, the use of non-

* Corresponding Author:
E-mail address: jalil_addeh@yahoo.com – Tel, (+98) 9112761472 – Fax, (+98) 1733296234

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linear models and controllers lead to time-consuming designs, since no standard techniques or software tools are available. Lauwerys et al. [18] present a practical, experimental approach using linear identification and robust control techniques on an active suspension of a quarter car test-rig.

A linear robustly performant controller is obtained using m-synthesis based on an experimentally identified linear model of both the active suspension and the quarter car dynamics. The relatively simple construction of the test-rig and the linearity of the active suspension make it possible to successfully apply linear identification and control design techniques. The dynamics of a real car are however much more complex and a semi-active suspension behaves quite differently than an active suspension: it becomes uncontrollable when the rattle velocity1 goes to zero. Because of these two reasons, the techniques developed in [18] are not extendable to this application.

Robust control theory is one of the control theories and the robustness especially handles the uncertainty during the design of the controller [19]. The study began in late 1970s and early 1980s and rapidly developed a technology of processing bounded system un-certainty method. “Robustness”, which means “strength”, can be explained that the control system maintains the ability of certain performance characteristics such as stability robustness and performance robustness under the parameter perturbation. Robust control theory is a control theory and method which maintain the function of the system when working under the model perturbation and random disturbance from the outside or inside of the model [20, 21].

The main robust control theories are as follows: (1) Interval theory of Kharitonov [22]; (2) H-infinity control theory [23]; (3) Structured singular value theory (or µ theory) [24]. Among the three theories, the H-infinity control theory is the most successfully adopted control theory and its theoretical system is almost perfect [25]. In this paper an intelligent robust control system based on H-infinity is proposed for active car suspension system.

2. Optimization Method

So far many controllers have been developed to control the car suspension system, of which the H-infinity controller is found to guarantee robustness and performance. The synthesis procedure of H-infinity controller can be done only by selecting proper weight functions. The selection purely depends on the plant model. There are no hard and fast rules for selecting the performance weight function and the robustness weighting functions. In the proposed method, an automatic weight selection algorithm is proposed to design robust H-infinity controller automatically for active car suspension system. As mentioned in the proposed method, we used quarter car model and H-infinity robust controller. The main structure of proposed method is shown in Figure 1.

The design of linear suspension controllers that emphasize either passenger comfort or suspension deflection. As is standard in the H-infinity framework, the performance objectives are achieved via minimizing weighted transfer function norms.

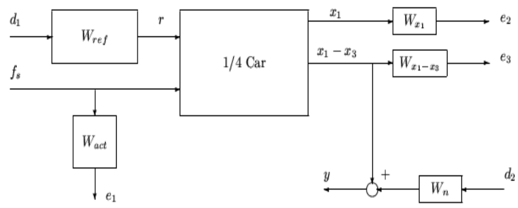

Figure 1. The main structure of proposed method

Weighting functions serve two purposes in the H-infinity framework: They allow the direct comparison of different performance objectives with the same norm, and they allow for frequency information to be incorporated into the analysis. The measured output or feedback signal y is the suspension deflection $(x_1 - x_3)$. The controller acts on this signal to produce the control input, the hydraulic actuator force f_s. The block w_n serves to model sensor noise in the measurement channel. w_n is set to a sensor noise value of 0.01 m. In a more realistic design, w_n would be frequency dependent and would serve to model the noise associated with the displacement sensor. The weight W_{ref} is used to scale the magnitude of the road disturbances. Assume that the maximum road disturbance is 7 cm and hence choose $W_{ref} = 0.07$. The magnitude and frequency content of the control force f_s are limited by the weighting function W_{act}. The main structure of W_{act} in the proposed method is as follows

$$W_{act} = A\frac{s+B}{s+C} \tag{1}$$

The purpose of the weighting functions W_{x1} and W_{x1-x3} is to keep the car deflection and the suspension deflection small over the desired frequency ranges. In the first design of proposed method, we are designing the controller for passenger comfort, and hence the car body deflection x_1 is penalized. The main structure of W_{x1} in the proposed method is as

$$W_{x1} = \frac{D}{s+E} \tag{2}$$

The suspension deflection weight W_{x1-x3} is not included in this control problem formulation. More details regarding car model can be found in [26, 27]. In the second design, we are designing the controller to keep the suspension deflection transfer function small. Hence the road disturbance to suspension deflection $x_1 - x_3$ is penalized via the weighting function W_{x1-x3}. The W_{x1-x3} weight magnitude rolls off above $10\ \mathrm{rad/s}$ to roll off before an open-loop zero $(23.3\ \mathrm{rad/s})$ in the design. It is defined as

$$W_{x1-x3} = \frac{25}{10s+1} \tag{3}$$

In the proposed method, the coefficients of weight transfer functions found by bee’s algorithm [28]. Figure 2 shows the sample bee in the proposed method.

$$Sample\ bee = [A \quad B \quad C \quad D \quad E]$$

Figure 2. Sample bee in the proposed method

3. Simulation Results

In this section the computer simulation results are presented. The computational experiments for this section were done on Intel core 2 Duo with 4 GB RAM using ASUS computer. The computer program was performed on MATLAB (version 2015) environment. In this section, we have done several experiments for evaluating of the proposed method. All the analysis till now has been in the frequency domain. Time-domain performance characteristics are critical to the success of the active suspension system on the car. Time response plots of the two H controllers are shown in following figures. All responses correspond to the road disturbance $r(t)$ that described by Eq. (4) as

$$r(t) = \begin{cases} a(1 - \cos 8\pi t), & 0 \leq t \leq 0.25 \\ 0 & \text{otherwise} \end{cases} \quad (4)$$

where $a = 0.025$ corresponds to a road bump of peak magnitude 5 cm.

3.1. The Performance of Controller Without Optimization

In this experiment, the performance of controller without optimization is investigated. In this experiment, the weight functions are selected by loop shaping method. The obtained results are as follows

$$W_{act} = \frac{100}{13}\frac{s+50}{s+500} \quad (5)$$

$$W_{x1} = \frac{251.3}{s+31.42} \quad (6)$$

The performance of formed robust controller are shown by Figures 3 to 5. It can be seen that the difference between the desired body travel and road disturbance is great. Also body acceleration has high value. From Figure 5, it can be seen that high control force is required to control the car body.

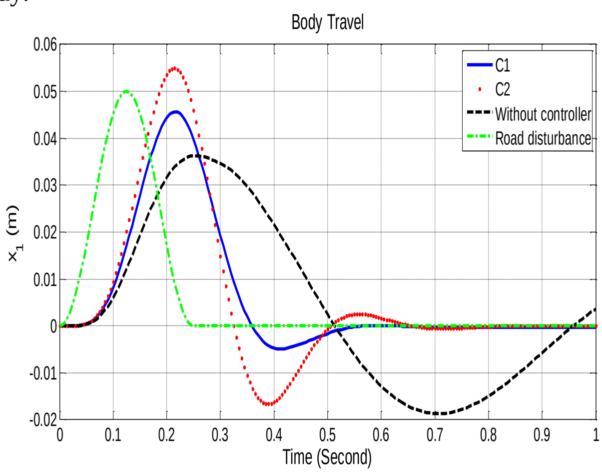

Figure 3. Body travel (without optimization)

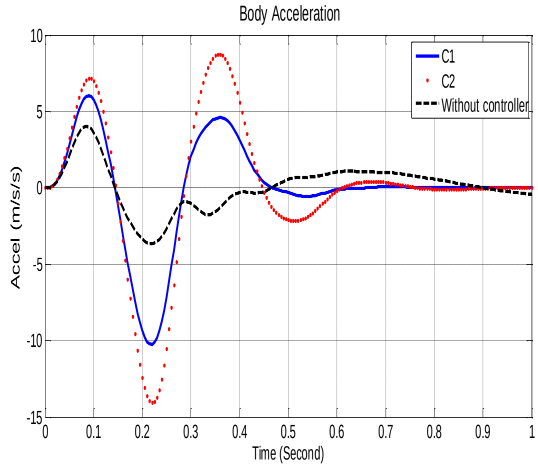

Figure 4. Body acceleration (without optimization)

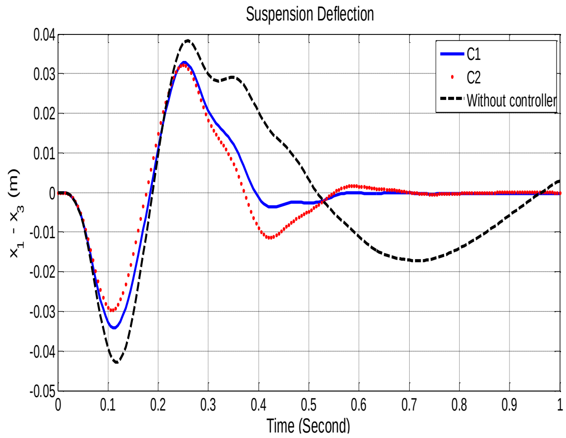

Figure 4. Suspension deflection (without optimization)

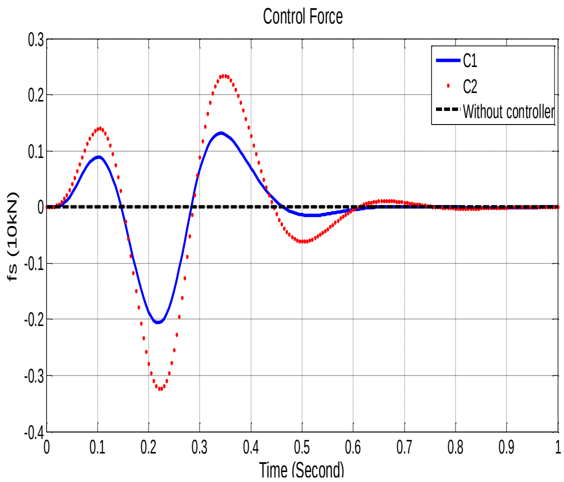

Figure 5. Control force (without optimization)

3.2. The Performance of Controller Wit Optimization

In this experiment, the performance of optimal robust controller is investigated. In this method, the coefficients of weight transfer functions are selected by bee's algorithm. The BA parameters are listed in Table 2. The performance of formed robust controller after optimization are shown by

Figures 6 to 9. It can be seen that the performance of robust controller is enhanced significantly after optimization.

Table 2. The BA parameters

Parameter	Value
Scout bee's	10
Elite bee's	3
Onlooker bee's	3
Maximum iteration	100

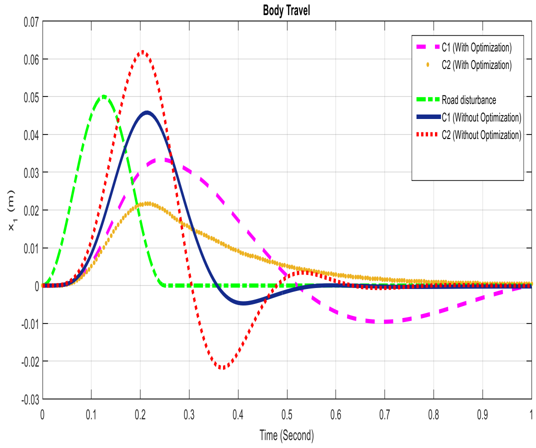

Figure 6. Body travel (with optimization)

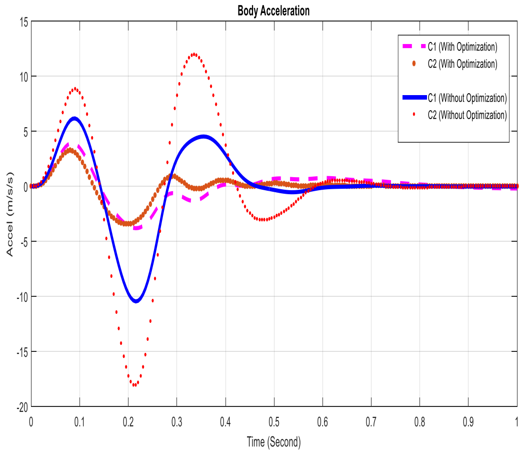

Figure 7. Body acceleration (with optimization)

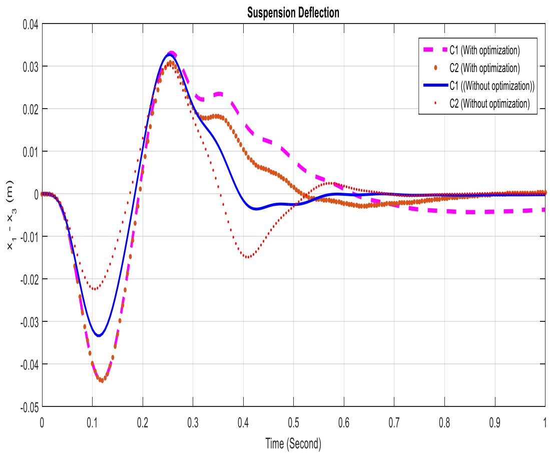

Figure 8. Suspension deflection (with optimization)

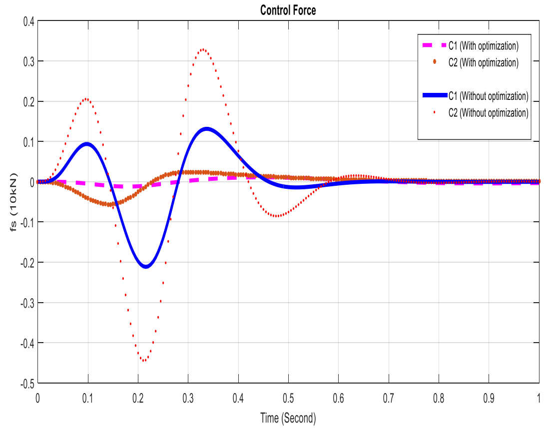

Figure 9. Control force (with optimization)

4. Conclusions

The road irregularities cause discomfort riding whendriving the vehicle. Vehicle suspension systems have acrucial role in reducing the sprung mass acceleration,providing comfortable and safe riding and suitablesuspension deflection to minimize the road damages.Comfort and road handling of a passenger car can be improved by replacing its passive suspension by a controlled activesuspension. The selection of an appropriate control structure is crucial, since it determines the complexity of thecontrol design and parameter tuning process.The purpose of this paper was to present a robust control scheme for a quarter-car suspension system under aroad disturbance profile. Here an intelligenthybrid model is presented in order to design aH-infinity controller that allows avoid the induced road variations over the car body. Novelty of this paper is given bythe bee's algorithm used to find the weighttransfer functions. The obtained results showed that the proposed method has excellent performance.

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