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Simulation of an Active Suspension Control for Frequency Response from Road Vibrations Using H-Infinity Synthesis and Validation for Robust Operation Using M-Synthesis

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Abstract
In a general suspension system ride comfort and suspension performance are inversely related. Whenever the vibrations are damped in a periodic and slow manner, the ride comfort improves but the suspension performance degrades. This project presents a frequency-based controller for active suspension using the H-infinity algorithm. The implementation includes performing the analysis using a general quarter car model. This is done to reduce the complexity of the model and at the same time helps to carry out the analysis without any compromise. In the implementation, the quarter car model is converted into a state-space model to make further analysis easier. The next part includes the development of a robust controller so that the system response is acceptable even in the case of uncertain input. In the model developed, the output states to be monitored are taken as suspension deflection and body acceleration. The input state to the controller is taken as the actuator force input. Intent of the model development is to mitigate the general trade-off between ride comfort and suspension performance.

Keywords: Active suspension, quarter car, hydraulic actuator, state-space, weighting function

1. Introduction

Active suspension system represents the highest level of performance in the suspension types. There is a general trade-off between ride comfort and suspension performance in every suspension system. As the designer goes on improving the ride comfort, handling degrades. This happens because system performance is defined by the vibration damping characteristics of the system. The faster the damping the better the system performance. On the other hand, ride comfort is measured on the basis of how smoothly the system damps out the vibrations. In the project, the main objective kept in mind is to design such a suspension system which mitigates this problem to a considerable extent. Suspension deflection and body acceleration are the two main parameters to be analyzed.

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In the model developed, the output states to be monitored are taken as suspension deflection and bodyaccelerationwithinthelimits-0.315m/s2to+0.315m/s2forcomfortand-2m/s2to+2 m/s2 for handling as per ISO 2631 standard used by Fabian Leon Vargas et al [9]. The input state to the controller is taken as the actuator force input with a minimum force capacity of 2.08 kN for lifting the sprung mass. The road profile used for testing the system was developed by Mohammad M ElMadany et al [7].

1.1. Aim

To simúlatean H-infinity controller for a general active suspension system using frequency response to roadex citations to mitigate the general trade-off betweenridecomfort and performance and making the system robust using Mu synthesis.

1.2. Objectives

- To convert a quarter car suspension model into state-spacemodel
- To design a controller for the model using frequency analysis of roadexcitations
- To define and test the system for external disturbances (roadexcitations)
- To calculate suspension deflection and body acceleration as a function of frequency to gain insight into the performance of thesystem
- To make the system performance robust which means that the body acceleration values should lie within the range of -0.315 m/s^2 to +0.315 m/s^2 for the comfortmode -2 m/s^{2}to +2 m/s^{2}for the handling mode and the system be stable in the frequency rangeevenwhen the input (the roadexcitationsmayvarytill 100% from the givenamplitudelimit) byemploying the μ synthesis
- To ensure a proper balance between handling where it refers to damping the vibrations withstability

2. Literature review

2.1.Quarter car model

Quarter car model is the reduced scalere presentation of the whole car. It isused because it reduces the complexity confronted while using a whole car model as well as reduces the computation time and the results can be easilyscaledto a full car model. In the model, a quarter car model employed by WiechaoSun et al [11] isused to carryout further analysis. Newton’s second law is applied on the model to represent equations of motion.

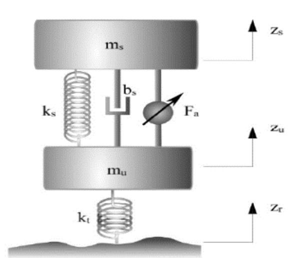

Figure 1: Quarter car mode representation

- Parameters:

1. Sprung mass (ms = 320kg)

2. Un-sprung mass (mu = 40kg)
3. Spring constants of the suspension (ks = 18000N/m)
4. Spring constants of the tire (kt = 200000N/m)
5. Damping coefficient of the damper (bs = 1000Ns/m)

The abovemodel can be brokendownaccordingtoNewton'slawsofmotion as follows:

$$m_u\ddot{z}_u + c_s(\dot{z}_u - \dot{z}_s) + k_t(z_u - z_q) + k_s(z_u - z_s) + c_t(\dot{z}_u - \dot{z}_q) = 0$$

$$m_s\ddot{z}_s + c_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) = 0$$

2.2.Roadprofile

2.2.1. Assumptions

i) The road profile is assumed to be sinusoidal with frequency fixed at 0.314 rad/s
ii) The maximum amplitude of the profile is fixed at 0.0025m

In the model, the road profile takenis sinusoidal and based on power spectraldensity. Powerspectraldensity (PSD) is a measureof the signal'spowercontent versus frequency and isused tocharacterize broadbandsignals. PSD shows the variation in the amplitudes over the frequency range as done byVerros, G. Papadimtrou et al [8]

$$x_g(t) = \sum_{n=1}^{N} s_n \sin(n\omega_0 t + \varphi_n). \qquad (2)$$

where xg is the value of the amplitude at a giveninstantof time and $s_n$is the maximum amplitudeof the roadprofilegiven as:

$$s_n = \sqrt{2S_g(n\Delta\Omega)\Delta\Omega} \qquad (3)$$
$$\Delta\Omega = 2\pi / L \qquad (4)$$

where L (1000 m) is the lengthof the road and Ω is the spatial frequency and Sg is the roughnessof the road and the value of Ω comes outto be 0.00628 m^{-1}

The value of sn comes outto be 0.0025 m forbadroads and 0.00063 m forgoodroads. Forgoodroads: $Sg=16\times10^{-6} m^2 cycle^{-1} m^{-1}$(takenfromISO2631standard)

Forbadroads: $Sg=256\times10^{-6} m^2 cycle^{-1} m^{-1}$

2.3. H-infinitymethod

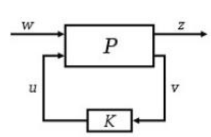

Figure 2: H-infinityparameters

As can be seen in fig 2, an H-infinity controller consistsof the mainsystemorplant (P) and the feedback controller (K) as suggested byMiaomiao MA et al [14]. K mapsmeasuredquantity ‘y’ onto control input ‘u’ as u=Ky, where y are the measured output quantities (in this case body acceleration and suspensióndeflection). P istheplantmatrix.

$$\begin{bmatrix} z \\ v \end{bmatrix} = \mathbf{P}(s)\begin{bmatrix} w \\ u \end{bmatrix} = \begin{bmatrix} P_{11}(s) & P_{12}(s) \\ P_{21}(s) & P_{22}(s) \end{bmatrix}\begin{bmatrix} w \\ u \end{bmatrix} \qquad (5)$$

w is the input (roadexcitations in this case) and u is the controlled input (actuatorforce), z is the error signalwhichneedsto be minimized, v represents the measuredvariables.
The basicobjectivelies in minimizing the norm:

$$\overline{z}=(s)\omega=[D+C(sI-A)]B \qquad (6)$$

2.4. μ-synthesis

Robust systemimplies a system which can giveresults withinlimitsofpermissibilityeven in the case ofuncertain input, for the given case the robustness criteria should be satisfied with the input varying as 30 % t ‘μ’. The expresión ofμisgivenas:

$$\mu = \frac{1}{det(I-\Delta M)} \qquad (7)$$

M is obtained as follows:

$$M=P+(I-PK)^{-1}P (8)$$

where P is the plantfunctiongivenby:

$$P = D+(sI-A)B (9)$$

3. Method of Analysis

1) Design a basic quarter car model representing a suspensionmodel
2) Setup the values for road disturbance to test the systemresponse
3) Finding the values of undesirable frequencies – tire hop and rattle spacefrequency
4) Design of a force actuator model (representing the actuator of the activesuspension)
5) Allocate weights to the control effort, suspension deflection, body acceleration so as to minimize the H-infinity norm from disturbanceinputs
6) Design of a feedback controller (nominal H-infinitycontroller)
7) Simulate suspension deflection and body acceleration as a function of time first using the regular H-infinity controller so as to see whether the parameter response is within the specifiedlimits
8) Test the system for robustness usingμ-synthesis

4. Results and discussion

4.1 Rattle-space and tire hop frequencies

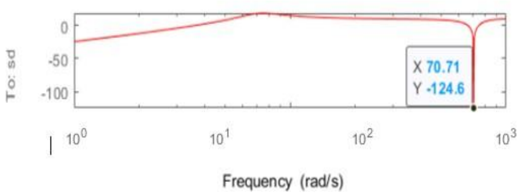

Figure 3: Magnitude plot of the transmissibility transmissibility of road excitation to suspension

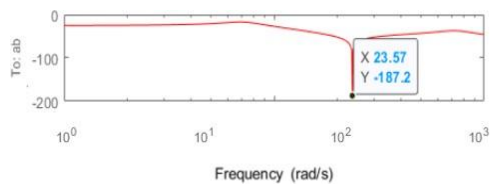

Figure 4: Magnitude plot of theof road excitation totire

Twosuddenundesirablechanges can be seencorrespondingtorattlespace (70.71 rad/s) and tire hop frequency (23.57 rad/s) respectively**.**

4.2 Results for regular H-infinitymodel

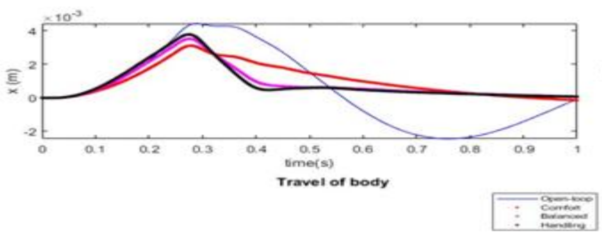

Figure 5: Suspension travel (m) vstime(s)

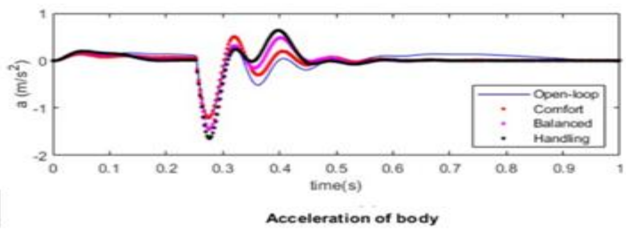

Figure 6: Body acceleration (m/s2) vs time(s)

From fig 5 and 6, it can be inferred that the graph for the handling modesettles the fastest with a maximum body travel of 0.0037 m and within the acceleration range (0.64 m/s^2 to-1.63 m/s^2) whichis the highest acceleration range. As opposite tothis case, the graph for the comfortmode case follows the acceleration range (0.30 $m/s^2$to -1.1 m/s^2), for the balanced mode the acceleration liesbetween -1.5 $m/s^2$to 0.42 m/s^2.

4.3 Validation with standard results

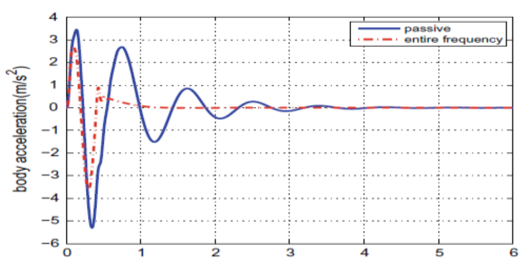

Figure 7: Acceleration(m/s2) vs time(s) graph for entire frequency H-infinity control as obtained in the research

The resultcomparisonbetween the two shows a good agreement with a maximum differenceof 7.1%.

4.4 Results using robustmodel

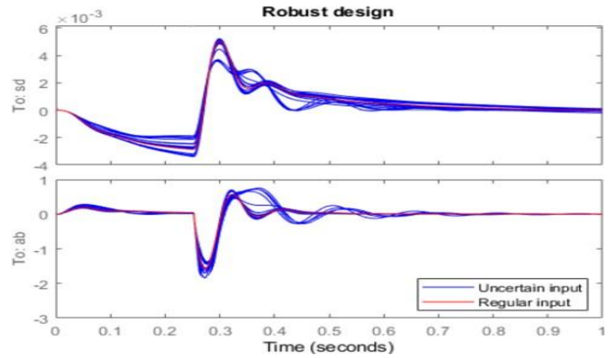

Figure 8: Output response of the robust controller for normal and uncertain input

The number of samples for the uncertain input are kept the same as 1000 in number. The peak μ value is obtained as 0.85.

5. Conclusion

In reality the suspension controller modelledwouldwork in the normal frequency rangewith the exceptions ofhigher frequencies as can be inferred from Bode plots. The controller modes can be implemented in drive control as per the requirement of the driver. If passenger confort is the requirement, confort mode will be selected since itis linked with the least body acceleration and in badroad conditions handling modewill be preferred. The lower the norm, the betteritwill be from an economical pointofviewsinceitwill lower the actuatorforce requirement. So, in short, to reduce the costof the system the confort mode would be the best choice. Increasing the uncertainty range response of the system will have a direct effecton the actuator cost and thus on the overall systemcost. Again, the controller implemented is a hydraulic onewhich the designer canchange.

6. References

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