Proceedings of the
2007 IEEE Intelligent Vehicles Symposium
Istanbul, Turkey, June 13-15, 2007

A disturbance response decoupling controller for emulating vertical dynamics of vehicles

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Abstract— **Vehicle emulation applications motivate the need for active suspension systems that not only isolate the cabin and its passengers from road disturbances, but also track desired roll dynamics. In this paper we present a novel control structure for achieving this objective and present an initial evaluation of its performance.**

I. Introduction

An outstanding problem in automotive control concerns the design of prototype vehicles that are able to emulate, in a programmable manner, a range of vehicle dynamics. Vehicle emulation has emerged as a promising solution to an outstanding challenge in the development of ride and handling characteristics for advanced passenger cars: the bridging of the gap between numerical simulations and experiments on a proof-of-concept prototype vehicle. An emulating vehicle would act as a generic prototype and would be equipped with advanced computer-controlled actuators enabling it to modify its ride and handling characteristics. Examples of such advanced actuators include four-wheel-steering, brake-by-wire and active suspensions. An integrated chassis controller is required to command these actuators to track reference signals while considering the interactions between the different actuators.

Roughly speaking, the design of such vehicles involves the design of integrated lateral and vertical controllers to control the vehicle emulator. While it has been shown by a number of authors that the design of controllers to track pre-specified lateral dynamics may be carried out in a relatively straightforward manner [1][2], the design of active suspensions for the task of vertical emulation, is more difficult. Whereas, traditional suspension design is concerned primarily with the trade off between comfort and vehicle handling, vertical emulators must not only accommodate these requirements, but also track pre-specified roll trajectories, while at the same time rejecting considerable inertial perturbations from the lateral motion of the vehicle. In this paper we present a first step in this direction. We consider an active suspension arrangement considered by Williams [3]. We propose a control strategy for this structure that not only achieves vertical tracking to a required degree of accuracy, but also achieves this in a manner that is consistent with traditional suspension design (namely by decoupling the tracking problem from the problem of specifying the response of the vehicle to road disturbances).

This research was supported by EU Project CEmACS 004175.

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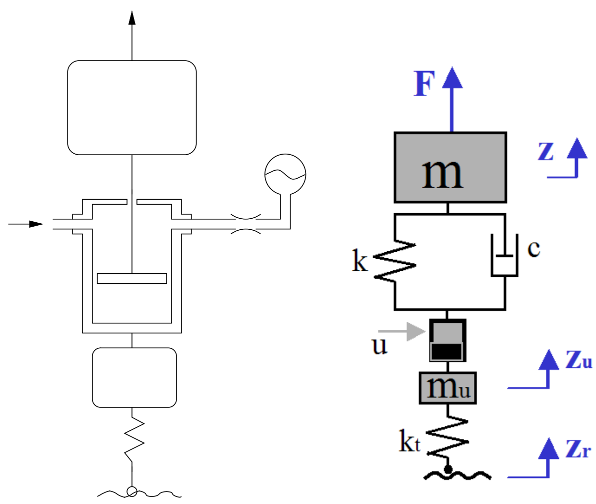
II. Related Work

The design of suspension control systems has been an active topic of research for more than four decades, and since the end of the 1980's active suspensions have been a feature of production automobiles. The design of controllers for active and semi-active suspensions has been approached using many different methods. An extensive survey of vehicle active suspension control, with particular focus on optimal control theory, is presented in [4]. Historically, initial attempts to design active suspension systems involved using two degree of freedom (2DoF) quarter car models. Such designs are not only of historical interest; they are also still used today. For example, many recent studies [5][6] have been based on the classical 2DoF model. Using this model it is possible to illustrate intuitively the skyhook and groundhook concepts used by many authors: see [7],[8],[9],[10] and [4] for more details about skyhook and groundhook.

Most of the recent work in the area has been based on controllers that are designed using more complex vehicle models. In particular, a half vehicle model with 4 degrees of freedom (4DoF) is used by a number of authors; see for example [11] and [12]. Such models are used in applications where it is desired to control vehicle pitch and heave. Full vehicle models with 7 degrees of freedom (7DoF) capture more vehicle behaviours and are used when not only heave and pitch are to be controlled but also vehicle roll and warp [11].

Previous work on vehicle emulation, to date, has been confined to the lateral dynamics. In [13], the vehicle emulator concept and its applications is analysed and this work is extended to lateral dynamics emulation in [14]. Lateral dynamics emulator design is also considered in [1] and [2]. Our focus in the present paper is the control design task for vehicle vertical dynamics emulation using a 7DoF vehicle model.



(a) (b)

Fig. 1. Suspension schematic diagram with (a) hydraulic elements and (b) equivalent mechanical arrangement.

III. Active Hydropneumatic Suspension Model

We develop our control strategy based on a quarter-car model and later extend it to regulate the roll angle of a full vehicle model. We consider a vehicle equipped with an active suspension of the hydropneumatic type[3]. For convenience, we use a mechanical equivalent obtained by linearizing the hydraulic elements around an operating point and neglecting leakages and fluid compressibility. The car is subject to an inertial vertical force F and road displacement z_r disturbance. The control input available is the actuator displacement u defined positive for expansion and negative for contraction. Thus the displacement of the passive elements at the strut z_{rel} is defined as: $z_{rel} = z - z_u - u$, where z is the sprung mass displacement and, z_u, the unsprung mass displacement. The sprung mass typically represents a quarter of the vehicle mass while the unsprung mass corresponds

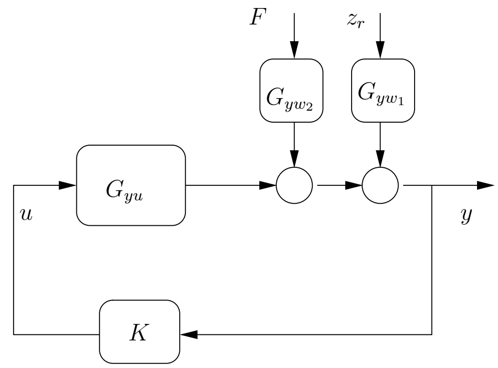


Fig. 2. General controller structure

mainly to the wheel mass. The vertical dynamics of the vehicle evolve according to:

$$m\ddot{z} = F - f_{strut}, \quad (1)$$
$$m_u\ddot{z}_u = f_{strut} + k_t(z_r - z_u), \quad (2)$$

where m denotes the sprung mass, m_u the unsprung mass, k_t the tyre stiffness constant, f_{strut} is the force at the suspension strut defined as

$$f_{strut} = c\dot{z}_{rel} + kz_{rel}, \quad (3)$$

where c is the damping of the shock absorber and k is the spring stiffness. The mass of the suspension strut and the mass of actuator are both neglected. Therefore, the force in the actuator is equal to the force at the suspension strut: $f_{act} = f_{strut}$.

Now consider that the measurements available are the vertical acceleration $\ddot{z}$, the strut displacement $z - z_u$ and the actuator force f_{strut}. The Laplace transfer function of the generalized plant is partioned as:

$$\begin{bmatrix} z \\ z_u \\ \hline \ddot{z} \\ z - z_u \\ f_{strut} \end{bmatrix} = \left[\begin{array}{cc|c} G_{xw_1} & G_{xw_2} & G_{xu} \\ G_{yw_1} & G_{yw_2} & G_{yu} \end{array}\right] \left[\begin{array}{c} z_r \\ F \\ \hline u \end{array}\right] \quad (4)$$

where the transfer function elements are derived from (1),(2) and (3).

IV. Vertical Trajectory Tracking with Disturbance Response Decoupling

A. Control Design

The emulation requirement is to track a desired trajectory z on the assumption of minimal road disturbances. The task of rejecting road disturbances will be left to the natural passive part of the suspension.

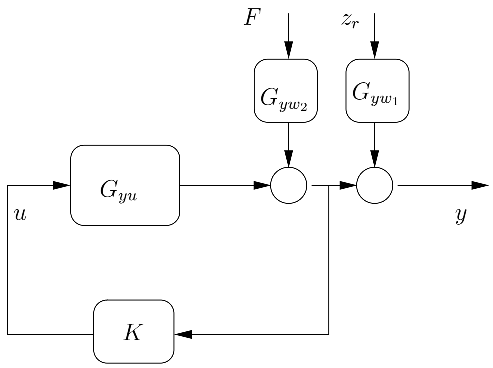


Fig. 3. General controller structure

Our basic idea is to exploit the active suspension cylinder force sensor in a direct manner to yield a decoupling controller. Specifically, based on the plant measured outputs for the quarter car model in (1) to (3) we select a control input composed of two parts: (a) to reject inertial force perturbations, (b) to track of the desired vertical motion. That is, the control input consists of two virtual inputs: $u = u_a + u_b$.

(a) Reject inertial perturbations

Consider the system in (4) and depicted in Figure 2. Normally, we could design a controller K robust to external perturbations based on the plant G_{yu} to regulate the displacement z. Nevertheless, in our case we just want to reject the inertial perturbations F while leaving the effect of the road disturbances z_r untouched. This requierement may be stated as choosing K such that the road distrubances z_r remain outside of the control loop as depicted in Figure 3.

Let us start defining K as:

$$K = Q_1 \Gamma, \tag{5}$$

where Q_1 is a non-zero scalar tranfer function and Γ is a non-zero vector with as many elements as system outputs. The elements of Γ should be proper and stable tranfer functions. Our requirement of the road disturbances z_r effect to remain outside the control loop is satisfied as long as the following equation is satisfied:

$$\Gamma G_{yw_1} = \begin{bmatrix} \gamma_1 & \gamma_2 & \gamma_3 \end{bmatrix} \begin{bmatrix} \frac{m}{d(s)} \\ \frac{m}{d(s)} \\ \frac{m}{d(s)} \end{bmatrix} = 0 \tag{6}$$

One simple solution to (6) is to make

$$\Gamma = \begin{bmatrix} m & 1 & 0 \end{bmatrix}. \tag{7}$$

Other possible solutions that satisfy (6) are

$$\Gamma = \begin{bmatrix} \frac{m}{ms^2+cs+k} & 0 & \frac{cs+k}{ms^2+cs+k} \end{bmatrix}, \tag{8}$$

$$\Gamma = \begin{bmatrix} \frac{m}{cs+k} & 0 & 1 \end{bmatrix} \tag{9}$$

used in [3] and [11], respectively.

Thus, in order to reject inertial perturbations, we choose a control input of the form:

$$c\dot{u}_a + ku_a = -Q_1(m\ddot{z} + f_{act}) \tag{10}$$

were $Q_1 = -\frac{k+k_t}{k_t(cs+k)}$ was chosen to achieve zero steady-state error of the sprung mass displacement z. Hence, the actuator input u_a is only a function of F. That is, we have

$$c\dot{u}_a + ku_a = -Q_1 F. \tag{11}$$

The control input structure for u_a is depicted in Figure 4(b) while the controller by Williams [3] is depicted in Figure 4(a).

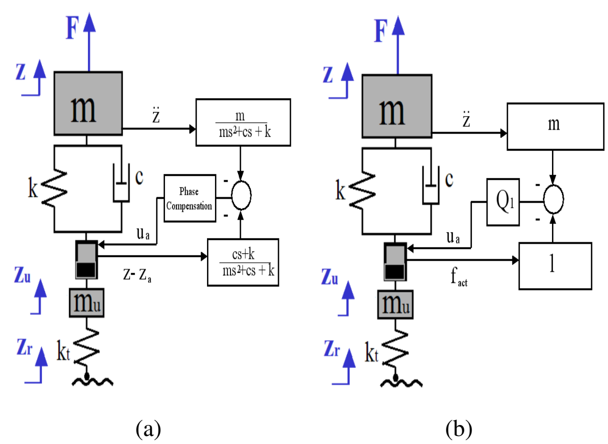


Fig. 4. (a) Control structure proposed by Williams[3] and (b) our approach to reject inertial perturbations.

(b) Tracking

Tracking control is achieved by selecting the virtual input u_b appropriately. Specifically, we consider:

$$u_b = (z - z_u) + \left(\frac{c^*}{c} e_0 + \left(k^* - k\frac{c^*}{c}\right) v_1\right) + v_2, \tag{12}$$

where: $c\dot{v}_1 = -kv_1 + e_0$, $c\dot{v}_2 = -kv_2 + m\ddot{z}^*$, $e_0 = (z^* - z_{uSS}) - (z - z_u)$, and z^* is the reference vertical motion trajectory that we are required to track, m^*, c^*, k^* specify the target closed-loop dynamics and z_{uSS} is the steady-state value of the unsprung mass (z_u) estimated as: $z_{uSS} = \frac{f_{act}}{k_t}$. Hence we notice that the first element on the right side of (12) cancels the damping and spring strut dynamics and the second element replaces them with the target tracking dynamics. The last element v_2 acts as a feedforward and completes the error dynamics for the sprung mass. As a remark, z_{uSS} is estimated using the measured force and reduces steady-state errors. Thus the sprung mass of the feedback system evolves as:

$$m\ddot{e}_1 + c^*\dot{e}_0 + k^* e_0 \quad = \quad 0, \tag{13}$$

where: $e_1 = z^* - z$. Nevertheless, by rewriting (12) in the frequency domain, and using appropriate values for c^* and k^*, decoupling can be achieved at certain chosen frequencies:

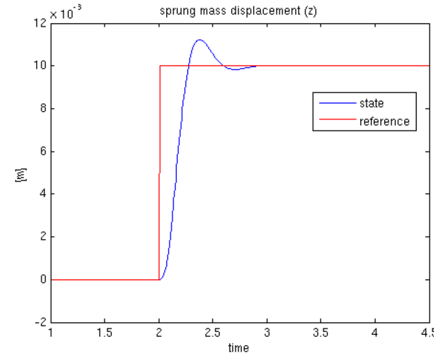
$$u_b = \left(1 - \frac{c^*s + k^*}{cs + k}\right) z_{rel} + \frac{m^*s^2z^* + (c^*s + k^*)z^*_{rel}}{cs + k}, \tag{14}$$

where $z^*_{rel} = z^* - z_{uSS}$. Notice that the virtual input u_b is a function of $z - z_u$ so it affects the original decoupling. Nevertheless, by choosing c^* and k^* as a lead-lag filter where $k^* = k$ and $\frac{c^*s+k^*}{cs+k}$ is close to 1 for low frequencies the road perturbations z_r remain practically outside the control loop below the selected frequency. Recalling that vibrations from 0 to 25 Hz are considered to affect perceived ride behaviour, and that road roughness effects start to increase after 1 Hz (rising to be one order of magnitude greater at 10 Hz) [15], we select c^* to leave chassis road isolation characteristics (z_r to z) practically unaffected after 4 Hz. The parameters m^*, c^*, k^* of the target closed-loop tracking dynamics are chosen to meet the following controller specifications: *10% overshoot, 0.3 s rise time, and settling time (at 5%) of 3s.*

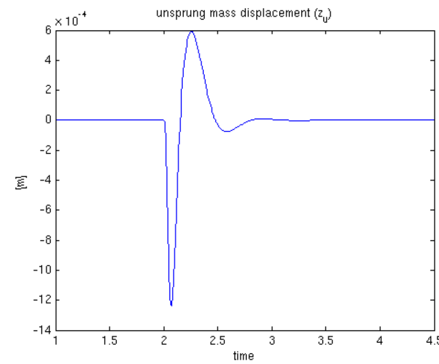
B. Performance

In Figure 5 the step response of the tuned tracking controller is presented. It can be seen that the tracking controller specifications are satisfied with overshoot of 12%, rise time of 0.28 s and settling time around 5% of final value of 2.85 s. We further illustrate the tracking performance of the proposed controller in Figure 6. A 100 N inertial force perturbation (F) is applied at the sprung mass at $t = 5s$.

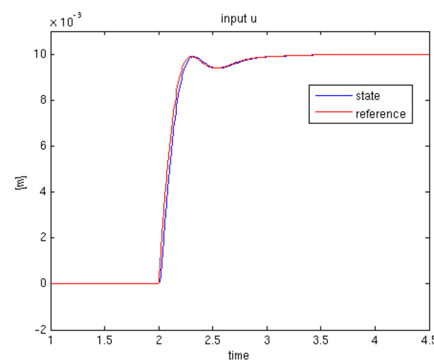
After testing the control structure using a suite of reference trajectories, we now consider the vehicle emulation application. Vehicle reference models are used to convert driver steering wheel angles into reference time-series trajectories for each of the vehicle states. The job of vertical and lateral controllers is to track these vehicle states. Notice that the vertical controller will be subject to the inertial perturbations caused by the lateral dynamics tracking.



(a)



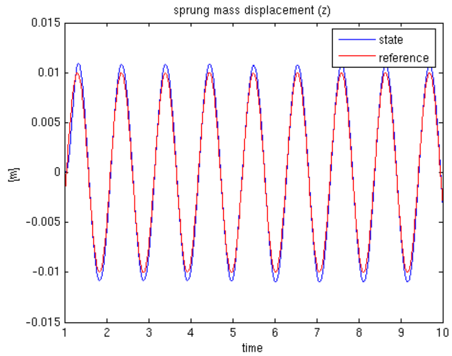
(b)



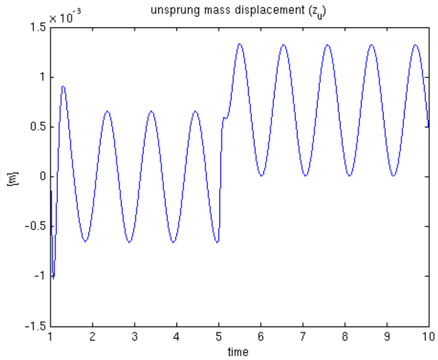
(c)

Fig. 5. (a) Sprung mass response, (b) unsprung mass response and (c) input to a step of magnitude 0.01m.

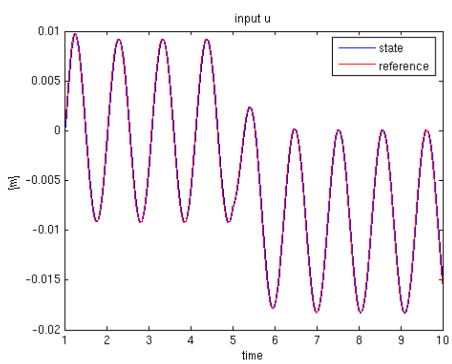
A full vehicle vertical dynamics simulation was used to evaluate the emulation response of a medium size test vehicle. The model has 7 degrees of freedom as the one in [11] but includes suspension geometry as well. It is described in [16]. The reference models used are from a large vehicle (emulation vehicle 1) and a mini-size car (emulation vehicle 2). The tests are performed at 72km/h with a J-turn maneuver at 1000 degrees/s reaching a lateral acceleration of around 4 m/s^2. The simulation is performed such that the inertial perturbation on the sprung mass corresponds



(a)



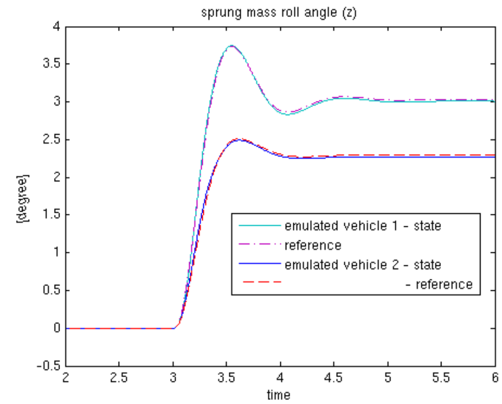
(b)



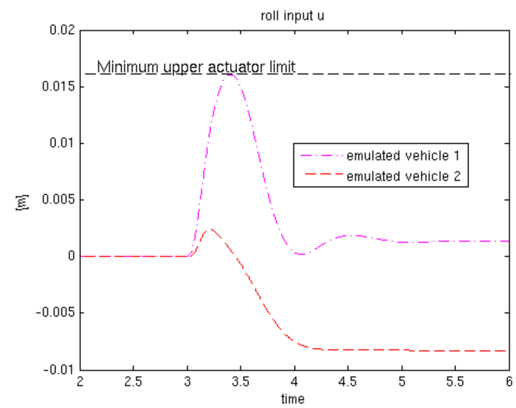
(c)

Fig. 6. (a) Sprung mass response, (b) unsprung mass response and (c) input for tracking a 1Hz sinusoidal trajectory with 100 N force step perturbation at t=5s.to a step of magnitude 0.01m.

to that induced by the lateral dynamics emulation. Roll angle time histories are shown in Figure 7(a) for both emulation vehicle tasks, together with the target emulation responses. The corresponding actuator displacements are shown in Fig. 7(b). It can be seen that the actuators displace more for vehicle 1 than vehicle 2, as vehicle 1 has a larger steady-state roll angle. The motion remains, however, within the actuator physical limits at all times. As a remark, we note that the steering maneuver selected is chosen to place extreme



(a)



(b)

Fig. 7. (a) Sprung mass and (c) input required for emulation of roll dynamics of two vehicles during a J-turn maneuver.

demands on the vertical controller.

V. Conclusions

In this paper we develop and evaluate a novel control algorithm for an active suspension system for emulation applications. It not only provides road isolation, but also tracks reference roll dynamics. While it has not been reported here, the robustness of the proposed strategy has been investigated for large parameter uncertainty. Future work will report this, extend the approach to other suspension arrangements, and include complete decoupling during emulation.

VI. Acknowledgements

This work was supported by EU STREP grant 004175.

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