# Combined stiffness/damping handling oriented control of a multichamber suspension

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***Abstract*—This paper proposes a combined handling-oriented stiffness/damping control for a multichamber suspension. It exploits the spring stiffness modulation capabilities to reduce the steady-state chassis angles of up to $49\%$ in roll and $36\%$ in pitch, with respect to the uncontrolled benchmark. The damper regulation prevents any overshoots or oscillations in the transient phase of harsh maneuvers, with an improvement of up to $44\%$ in pitch rate and $29\%$ in roll rate. Overall, the combined controller enhances the vehicle stability while ensuring satisfactory comfort performance on single-events obstacles and irregular roads.**

***Index Terms*—Suspension Control, MC suspension, Semi-Active Suspension**

## I. Introduction

The suspension system is among the components of a road vehicle that influence the most its entire behavior. It acts as a filtering element on irregular roads and, at the same time, during dynamic maneuvers it is of paramount importance for the road handling performance, intended as regulation of the chassis attitude angles and angular velocities. In the first case, to attain high comfort the suspension needs a soft calibration, conversely in the handling scenario a hard setting is required [1]. The need for compromise is partially overcome by the use of electronically controllable suspensions.

When it comes to road handling, most of the research deals with active suspensions [2] [3]: they have the advantage of introducing active energy in the system but the evident drawback of higher power consumption and potential safety issues. At the same time, semi-active suspensions, despite their potential, have been mainly studied with focus on comfort oriented control (*e.g.* [4] and citation within). The literature on handling control is scarce, however it is possible to find some works that apply LQ control [5] and LPV control [6] to increase vehicle stability. In [7] the authors propose a hierarchical approach that schedules (based on the driver's input) the parameters of a linear SkyHook formulation to increase the damping action during handling maneuvers, thus reducing angular oscillations.

In recent years a new technology capable of high frequency stiffness regulation, the multi-chamber suspension, has emerged on the market. Some high-end production vehicles exploit this solution by allowing the user to manually set different levels of stiffness according to the desired drive style [8]. To the best of the autors' knowledge, the only study in literature that proposes a closed loop handling-oriented control for a multichamber suspension is [9], where the potentiality of the actuator are fully exploited to reach up to $12\%$ in steady-state angle reduction with respect to the best passive benchmark. However, the study is developed under the assumption of linear passive damper.

This work analyses the potentialities of a handling-oriented combined damper and spring control, with three main objectives:

- develop a modular damping controller aimed at limiting the vehicle angular rates during steering, braking or acceleration maneuvers. The logic uses inertial measurement of the vehicle body to separately target its roll, squat and dive movements, with a simple formulation that facilitates deployment and calibration, and ensures a reduced computational load.
- define a priority based scheduling to effectively combine the handling controller modules, with the possibility to integrate additional functions (*e.g.*, comfort-oriented algorithm, end-of-stroke management) without any modification of the controller structure itself.
- combine the handling-oriented semi-active control with the state-of-the-art stiffness controller proposed in [9], ensuring enhanced vehicle stability thanks to the benefits of stiffness and damping regulation.

## II. Simulation model

In this Section the vehicle and suspension model used for control development and simulation are presented.

### A. Suspension model

The multichamber suspension in analysis is characterized by the parallel of a semi-active damper and a variable stiffness air spring, as visible in Fig. 1. The total suspension force is therefore given by:

$$F_{MC} = F_k + F_{damper}(\Delta \dot{z}) - Mg \tag{1}$$

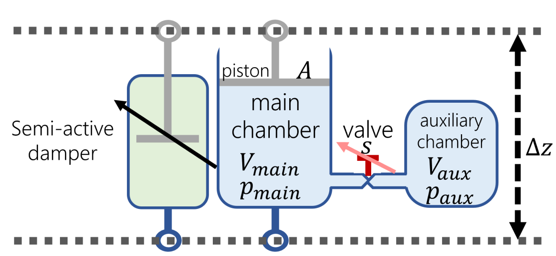


Fig. 1. Multichamber suspension scheme

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where $F_k$ is the spring elastic force, $F_{damper}$ is the dissipative force provided by the damper and $Mg$ is the vehicle static load.

The elastic element is a pneumatic actuator composed of a main chamber, which is subject to volume variations depending on the piston movement, and a secondary chamber, labeled *auxiliary*, with fixed volume. The two chambers are connected through an on-off controllable valve that permits to regulate the total volume of air that undergoes compression during the ride. By switching the valve position, it is possible to modify the spring equivalent stiffness, since it is proven that the stiffness coefficient $k$ is inversely proportional to the total air volume $V$ [1], as reported in the following :

$$k = \frac{\gamma \bar{p} A^2}{V}, \tag{2}$$

with $\gamma$ being the air polytropic coefficient, $\bar{p}$ the chambers pressure under static conditions and $A$ the piston area.

The spring model used in this work was first proposed in [10]. It is a thermodynamical model of the multichamber spring, based on the assumptions of adiabatic transformation and cylindrical main chamber. The valve position $s \in \{0 - closed; 1 - open\}$ is the actuator's discrete control variable: the open valve state correspond to a soft spring configuration, conversely closed valve implies a hard configuration. In Fig. 2 (up) the air spring is described by means of its elastic maps, obtained using the cited model, with the stroke being positive in elongation. The spring parameters are those of a commercially available multichamber suspension.

The dissipative element is an electro-hydraulic commercial damper, described by its static curves reported in Fig. 2 (down), where the stroke speed is intended positive in elongation. The damping force can be modulated continuously in the gray area enclosed between the minimum and maximum damping curve, by means of the control variable $c_{ref} \in [0, 1]$. Indeed, $c_{ref} = 0$ corresponds to the minimum damping curve, $c_{ref} = 1$ is the maximum damping curve; any intermediate value corresponds to a linearly scaled curve in between the two limits.

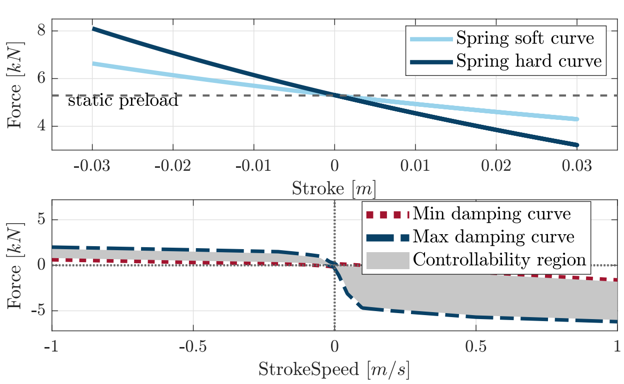


Fig. 2. Elastic and damping characteristic curves.

## B. Vehicle model

This work exploits the multi-body simulator VI-Grade to reproduce the vehicle dynamics: the selected model is a sedan car, equipped with four multichamber suspensions, as previously described. The main vehicle parameters used in simulation, along with the suspension parameters, are reported in Table I.

| Parameter | Symbol | Value |
|---|---|---|
| Vehicle sprung mass | $M$ $[kg]$ | 2100 |
| Wheelbase | $L$ $[m]$ | 3.3 |
| Track | $T$ $[m]$ | 1.6 |
| COG height | $H$ $[m]$ | 0.56 |
| Main chamber nominal volume | $V_{main,0}$ $[L]$ | 1.4 |
| Auxiliary chamber volume | $V_{aux}$ $[L]$ | 1.53 |
| Piston area | $A$ $[cm^2]$ | 133 |
| Air polytropic coefficient | $\gamma$ $[-]$ | 1.4 |

TABLE I
MODEL PARAMETERS

# III. STIFFNESS CONTROL

This work makes use of the stiffness controller proposed in [9]. Under the handling point of view, the aim of stiffness control is to minimize the steady-state roll and pitch angles when the vehicle is performing maneuvers that are strongly exciting for its longitudinal and lateral dynamics, thus enlisting all the cases when the driver is steering, braking or in traction. The principle behind handling-oriented stiffness control is to switch to hard spring during the maneuver, so to reduce the chassis angular variations, and back to soft spring after it has finished. Indeed, a soft spring layout is generally preferred for guaranteeing reasonably high comfort performance [1].

The control strategy is based on the idea that the chassis pitch and roll angles can be reduced by limiting the four suspension strokes independently from each other. This *decentralized* approach is implemented using an estimate of the load transfer at the four corners as input signal: it has the advantage of anticipating the stroke dynamics, thus predicting the chassis movement. The switching strategy has a core function that is based on a binary logic: whenever the force at the corner surpasses a predetermined threshold (detection of the start of the maneuver) the valve is closed to obtain a hard configuration. Conversely, when a second threshold is reached (detection of the end of the maneuver) the valves are switched back to soft configuration.

An additional feature that is worth mentioning targets the case when the load transfer of a corner rapidly inverts its sign, indicating an inversion of the maneuver. When the inversion is detected, a fast opening and closing of the valve is exploited to move the suspension equilibrium position in a manner that further reduces the stroke.

For a detailed explanation of these principles, the reader is again referred to [9].

# IV. SEMI-ACTIVE CONTROL

The damper main purpose in handling maneuvers (intended as any combination of steering, thrust or braking maneuvers)

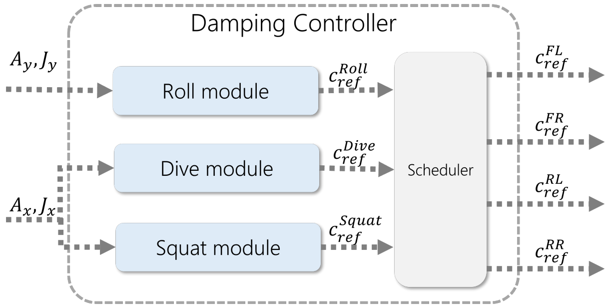


Fig. 3. Damper control scheme

is to smooth the vehicle angular oscillations by reducing the chassis angular rates. The damper mostly acts in the transient phase, characterized by an abrupt variation of lateral (longitudinal) acceleration: a strong damping action is desirable to maximize the angular rate reduction. However, the damper also has a non negligible influence on the ride feeling in the steady-state part of the maneuver, where the vehicle settles to a dynamic equilibrium with constant lateral (longitudinal) acceleration. In this second phase, a higher damping provides a *sport-like* ride feeling, with some detriment on the comfort performance. Indeed, as it is well known in literature, a higher damping configuration has negative effects on the suspension high frequency filtering capabilities [1].

Building upon the work proposed in [7], the aim of the semi-active controller is to promptly increase the damping action in the transient part of the handling maneuvers to improve the vehicle stability, while adding the possibility to modulate the force in the steady-state phase to achieve the desired drive feeling. Moreover, the vehicle comfort performances should be preserved by ensuring that the handling control action is present only when a handling maneuver occurs.

The general controller structure is depicted in Figure 3. Following a *modular* approach, separate *ad hoc* modules target each rotational movement of the vehicle body (excluding the yaw motion); each module provides a separate reference damping, whose intensity can be calibrated independently. The presence of two separate modules for dive (positive pitch angles) and squat (negative pitch angles) movements is given by the fact that the magnitude of the longitudinal acceleration (and longitudinal jerk) reached in braking is in general higher than the one reached in traction, thus requiring a different intensity in the control action. The two modules also allow for a calibration that takes into account any asymmetry in the front/rear load distribution, which greatly affects the overall vehicle dynamic behavior.

### A. Handling modules

The three handling modules are detailed in the following:

1) *Roll module*: it reduces the chassis roll rate in the event of maneuvers exciting the vehicle lateral dynamics. It reacts to variations in the chassis lateral acceleration $a_y$ and its derivative, the lateral jerk $j_y$. The output reference damping is indicated as $c_{\text{ref}}^{\text{Roll}}$.
2) *Dive module*: it targets dive motions, that generally occur in the first part of a braking maneuver. The control action is computed exploiting the negative part of the longitudinal accelerations $a_x^{neg}$ and the negative part of its derivative, the longitudinal jerk $j_x^{neg}$. The output reference damping is indicated as $c_{\text{ref}}^{\text{Dive}}$.
3) *Squat module*: it targets squat motions, that generally occur in traction or at the end of a braking maneuver, when the pedal is released or the vehicle comes to a full stop. The control action is computed exploiting the positive part of the longitudinal accelerations $a_x^{pos}$ and the positive part of its derivative, the longitudinal jerk $j_x^{pos}$. The output reference damping is indicated as $c_{\text{ref}}^{\text{Squat}}$.

It is necessary to remark that all the input signals are processed by means of suitable pass-band filters to remove undesired offsets and measurement noise. The jerk signals are obtained through a non-ideal derivative to prevent the amplification of high frequency oscillations in the measurements.

### B. Module structure

The three damping sub-modules have the same internal structure: each module presents a transient control, whose purpose in to promptly react when a handling maneuver is detected, and a steady-state control, which provides the possibility to modulate the damping force in the steady-state portion of the maneuver.

The transient module exploits the jerk signal as input to maximize the damping effect on the chassis oscillations. The equation describing the transient control is reported in the following:

$$c_{TRA}^{\text{Mod}} = \begin{cases} 0, & |j_{\text{in}}| \leq \bar{j}^{\text{Mod}} \\ sat_{[0,\bar{c}_{TRA}^{\text{Mod}}]} \left( k_{TRA}^{\text{Mod}} \left| j_{\text{in}} - \bar{j}^{\text{Mod}} \right| \right), & |j_{\text{in}}| > \bar{j}^{\text{Mod}} \end{cases} \tag{3}$$

where $c_{TRA}^{\text{Mod}}$ is the control action. The controller gain $k_{TRA}^{\text{Mod}}$ and the maximum damping value reached by the transient controller, $\bar{c}_{TRA}^{\text{Mod}}$, determine the intensity of the control action. The value $\bar{j}^{\text{Module}}$ is a dead-zone on the jerk signal: its purpose is to decouple the chassis movements in the vertical direction with respect to the ones in the longitudinal and lateral direction; it also prevents chattering due to residual measurement noise.

In a similar manner, the steady-state control reacts to the accelerations signal, according to:

$$c_{SS}^{\text{Mod}} = \begin{cases} 0, & |a_{\text{in}}| \leq \bar{a}^{\text{Mod}} \\ sat_{[0,\bar{c}_{SS}^{\text{Mod}}]} \left( k_{SS}^{\text{Mod}} \left| a_{\text{in}} - \bar{a}^{\text{Mod}} \right| \right), & |a_{\text{in}}| > \bar{a}^{\text{Mod}} \end{cases} \tag{4}$$

where $c_{SS}^{\text{Mod}}$ the control action. The controller gain $k_{SS}^{\text{Mod}}$ and the maximum damping value reached by the transient controller, $\bar{c}_{SS}^{\text{Mod}}$, determine the intensity of the control action. The value $\bar{a}^{\text{Module}}$ is a dead-zone on the acceleration signal, with the same working principle as the one above.

An envelope filter with tunable bandwidth ensures a smooth transition between the transient control and the steady-state control.

### C. Scheduling function

The *Scheduler* is in charge of computing the total output reference damping and distributing it to the four vehicle corners according to the scheduling function:

$$c_{\text{ref}}^{XX} = \max\left(c_{\text{ref}}^{\text{Roll}}, c_{\text{ref}}^{\text{Dive}}, c_{\text{ref}}^{\text{Squat}}\right) \tag{5}$$

where the apex $XX$ indicates the corner ($F/R$ for front/rear and $R/L$ for left/right) at which the reference damping is applied. Taking the maximum among the modules outputs ensures that priority is given to the module requesting the strongest control action, which in turn corresponds to the vehicle dynamics subject to the highest excitation. This priority scheduling guarantees the stability of the chassis for all handling maneuvers, including combined maneuvers that involve both the longitudinal and the lateral dynamics. Furthermore, exploiting this scheduling principle it is straightforward to *plug-in* additional functions in the controller, such as a comfort-oriented algorithm or a end-of-stroke management policy, without modifications of the controller structure itself.

## V. SIMULATION RESULTS

This section discusses the performance of the combined stiffness/damping control strategy over significant examples of handling maneuvers. Subsequently, the handling controller is tested in combination with a comfort oriented algorithm to show the interaction of the two in a comfort scenario.

### A. Control performance in handling scenario

The functioning of the handling controller is described in Fig. 4, which presents a mixed throttle/steering action followed by an emergency brake. This maneuver simultaneously activates the three damping modules, and represents a common scenario of maneuver inversion.

- *passive benchmarks*: irrespective of the stiffness configuration, when the damping is minimum, the vehicle angles present wider oscillation peaks and higher settling time than the case of maximum damping. On the opposite, a change in the stiffness mainly affects the angular steady-state values, with the hard mode ensuring reduced pitching and rolling.
- *semi-active control*: the damper control action is reported in the fourth plot, where each module activation is underlined. The scheduling function, acting as a priority-based maneuver selector, ensures that the prevailing control action corresponds to the module targeting each specific chassis movement. Indeed, at the beginning of the throttle action and during the braking recoil, at times $t = 4s$ and $t = 12s$, the Squat module kicks in, while during the first part of the braking maneuver, around time $t = 9s$, the Dive module activates. The Roll module manages the rolling stability during steering in the interval between $t = 5.5s$ and $t = 9s$. Each module is activated only when an aggressive maneuver occurs, keeping minimum damping in all other cases, to prevent any detrimental effect of the handling controller on comfort performance. The tuning of the control effort, obtained through a sensitivity analysis, is different for each module, and it aims at reflecting the different driving needs. Therefore, the action of the Dive and Squat modules is determined by the transient control, that rapidly increases the reference damping to maximize the angular rate reduction; subsequently, the control action slowly descends to zero in a manner that prevents abrupt discontinuities in the damping force. Conversely, for the Roll module the steady-state control ensures a residual damping action different from zero in the interval between the two opposite lateral acceleration steps. This can be justified considering that a cornering action has an average longer duration with respect to maneuvers in the longitudinal direction, where the focus of the stability lies almost exclusively in the impulsivity of the maneuver.
  The effect of the semi-active controller is visible in the Roll and Pitch angles: in the controlled case (dashed line) both signals are free from any overshoot in the transient and oscillations and settling phase. This is the result of an effective reduction of the angular rates, with an overall behavior comparable to the passive maximum damping configuration.
- *stiffness control*: the valve control actions are reported in the bottom of Fig. 4. During the whole experiment, the steady-state angles are kept limited by properly switching to the hard configuration. In particular, the switching occurs during transients, which is when the static loads redistribute among the vehicle corners; it also happens symmetrically for diagonally-placed corners, due to the vehicle geometry. The overall angles obtained with the switching mechanism are comparable to those of the hard mode, which is the best passive configuration for steady-state handling stability. In addition, the aggressive brake highlights the peculiar feature of the multichamber suspension control in case of maneuver inversion. The fast opening and closing action at $t = 9s$ leads to a change in the stroke equilibrium positions which allows the controlled suspension to outperform the angular value obtained with the hard passive stiffness. For a detailed explanation on this control principle, see again [9].

### B. Handling indexes

The coupling of damping and stiffness controllers leads to a simultaneous regulation both of the transient oscillations and of the steady-state angular values. Therefore, two handling indexes are defined to quantify the performance:

1) the Root Mean Square (RMS) of the chassis angles, which contains information on the vehicle attitude throughout the whole maneuver with particular emphasis on the reduction of the steady-state angular value. For

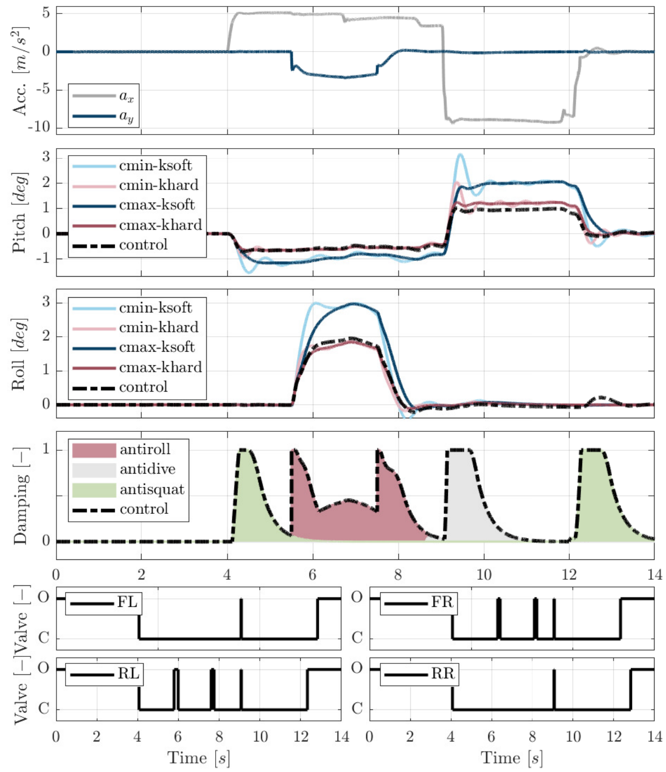


Fig. 4. Performance of the control strategy in time.

the pitch angle (and similarly for the roll angle) it can be expressed as:

$$J_{RMS}(\theta) = \sqrt{\frac{1}{T}\int_0^T \theta(t)^2 dt} \quad (6)$$

where $\theta(t)$ is the vehicle pitch angle.

2) the maximum of the roll and pitch angular rate, that underlines the effect of the damping control in slowing the chassis movement during handling maneuvers. For the pitch rate (and similarly for the roll rate) it can be defined as:

$$J_{\max}(\dot{\theta}) = \max(|\dot{\theta}(t)|) \quad (7)$$

where $\dot{\theta}(t)$ is the vehicle pitch rate.

Fig. 5 shows the numerical indexes normalized with respect to the full soft configuration, *i.e.* having open valves and minimum damping. With reference to the passive configurations, a higher damping is mainly responsible for attenuating the angular rates, which directly translate to a smoother transient behavior. Dually, a harder spring has direct effect on the angles themselves, reflected in lower $J_{RMS}$ index values. As a consequence, by applying the decoupled control actions (only stiffness or only damping) the predominant effect is reducing either the angular variation or the angular rate, with performance comparable to *khard-cmin* or *ksoft-cmax* respectively.

On the opposite, combined control merges the stiffness and damping regulation benefits and allows to minimize both indexes, with improvements up to 49% and 36% in pitch and roll respectively, and up to 44% and 29% in pitch rate and roll rate respectively, with respect to to the full soft configuration. These results are comparable with the full hard configuration, with the advantage of increasing the damper and spring force only when needed. In case of maneuver inversion, combined control outperforms the passive configuration, as visible in the $J_{RMS}$ of the pitch, with an additional 9% improvement compared to the *cmax-khard* case.

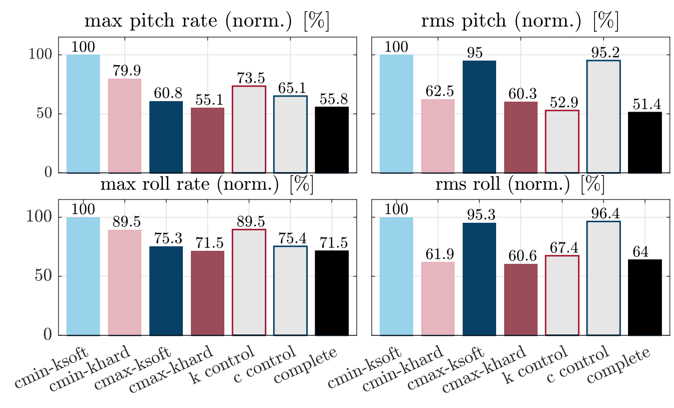


Fig. 5. Handling indexes obtained in simulation.

### *C. Controller performance in comfort scenario*

The handling controller is now tested in a comfort scenario to verify its effects on the vehicle comfort performance.

The stiffness modulation strategy, thanks to the switching logic based on a load transfer estimation, ensures a soft spring configuration unless a handling maneuver is detected. Therefore the focus of this analysis is shifted on the damping control, where the scheduler is exploited to add to the controller modules a continuous SkyHook formulation (proposed in [7]) in the form:

$$c_{\text{ref}}^{XX} = \max\left(c_{\text{ref}}^{\text{Roll}}, c_{\text{ref}}^{\text{Dive}}, c_{\text{ref}}^{\text{Squat}}, c_{\text{SH}}^{\text{XX}}\right) \quad (8)$$

where $c_{\text{ref}}^{\text{Roll}}$, $c_{\text{ref}}^{\text{Dive}}$, $c_{\text{ref}}^{\text{Squat}}$ are the outputs of the handling modules, as seen in the previous sections, and $c_{\text{SH}}^{\text{XX}}$ is the SkyHook control action for the corner $XX$. The results in time domain are reported in Fig. 6: the experiment consists in driving with straight trajectory and constant speed on an irregular road profile followed by a speed bump.

It is visible how the dead-zones $\bar{j}^{\text{Module}}$ and $\bar{a}^{\text{Module}}$, whose value is reported in dashed lines in the second and third plot respectively, act as decoupling factors on the jerk and acceleration signals. They determine a threshold on the level of excitation in the longitudinal and lateral direction below which there is no activation of the handling modules. This ensures that in a comfort scenario such as the one in analysis, the prevalent control action is determined by the SkyHook, as reported in the lower plot. Indeed, on the speed bump there are no intervention from the handling modules while on the rough road there are some sporadic activation of the roll module, with an amplitude of the reference damping that is less than $15\%$ of the maximum damping. The effect is reflected in the comfort performance index, defined as the root mean square of the chassis vertical acceleration:

$$J_{comf} = \sqrt{\frac{1}{T}\int_0^T a_z(t)^2 dt}, \qquad (9)$$

reported in Fig. 7 normalized with respect to the full soft configuration. As expected, the SkyHook algorithm constitutes a good trade-off between the passive configurations, even surpassing the soft benchmark on the irregular road. When coupled with the handling modules, the overall controller still ensures good comfort performance on the speed bump: it is equivalent to the case where only the vertical control logic is active. On the rough road, it presents a slight deterioration of performance of less than $6\%$, thus still ensuring an improvement of $9.4\%$ with respect to the soft configuration.

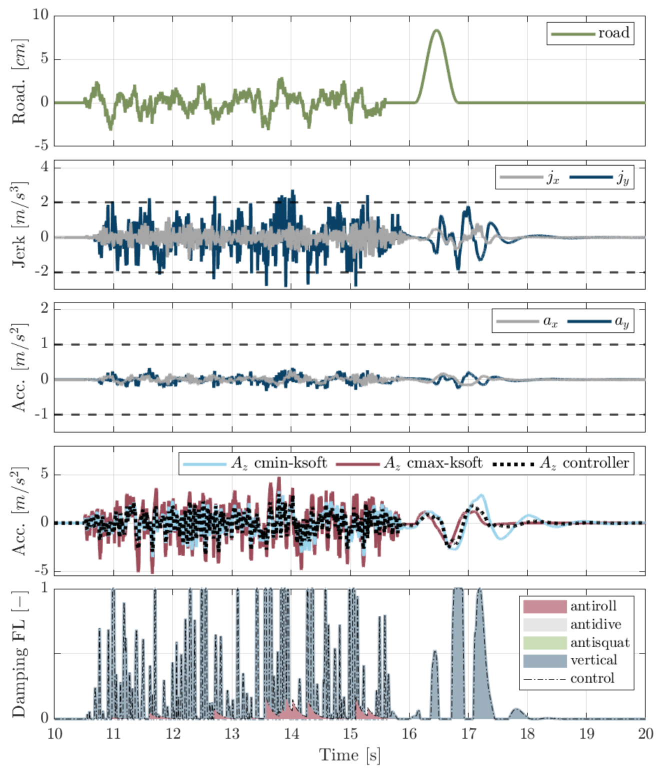


Fig. 6. Performance of the control strategy in time in a comfort scenario

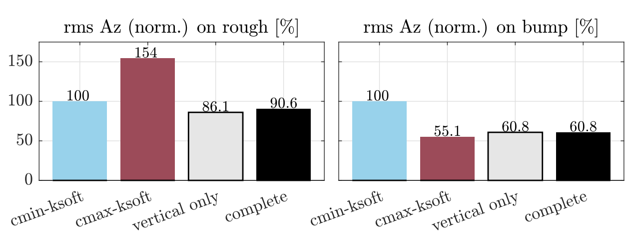


Fig. 7. Comfort indexes obtained in simulation

## VI. Conclusions

This paper proposes a handling-oriented control strategy based on the combined action of damping and stiffness regulation. The damping control mainly aims at smoothing the angular oscillations during the maneuver transient, whereas the stiffness control is in charge of reducing the steady-state angular variation. These results lead to an increased sense of safety and stability perceived by the passengers whenever a medium to aggressive maneuver occurs, being it a throttle, brake or steering event.

The overall simple and agile formulation, that requires a reduced number of measurements, aims at ensuring ease of implementation and efficiency of computation, while guaranteeing enhancement of performance with respect to the passive suspension framework. The choice of the damping scheduling function is able to set the amount of damping request based on the specific maneuver. Moreover, the controller modular structure allows to introduce or remove any additional functions, such as a comfort module, without compromising its functioning or changing the scheduling function. Finally, the integration of damping and stiffness controllers is done so that their benefits are preserved and added to each other in the combined controller version.

All being said, to the best of the author's knowledge, the proposed strategy represents the state-of-the-art suspension control for the car real-time handling regulation. Future developments concern the controller implementation and validation on a real vehicle.

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