

Design and Validation of a Full Body Control Semi-Active Suspension Strategy for a Supercar

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Abstract: The paper designs a semi-active suspension control strategy for a supercar. The control strategy aims at providing sprung mass stability in terms of heave, pitch and roll dynamics. Supercars dynamics are very demanding in that the vehicle has to provide excellent stability during pilot-induced sprung mass movements (braking, acceleration and turning) and a reasonable road disturbance isolation. The proposed control system is based on four independent modified sky-hook controllers and a centralized high-level controller that schedules the parameter of the sky-hook algorithms taking into account the driver's input. The paper implements the proposed algorithm on an instrumented supercar and validates the approach on a number of maneuvers.

Keywords: Suspension Control, Semi-Active Suspension, Magneto-Rheological Suspension

1. INTRODUCTION

Suspensions have a large impact on the drivability and safety of any wheeled vehicle (see for example Savaresi et al. (2010)). Over the years, both academia and industry have developed and explored a plethora of different technologies and approaches: starting from hydro-dynamic suspensions to fully-fledged fast acting active suspensions. Today, the community regards two main technologies as practical and viable: Electronic hydro-pneumatic suspensions and semi-active suspensions.

Semi-active suspensions achieve a higher frequency control. Semi-active suspensions modulate the suspension damping. In most cases, the change happens in the order of milliseconds; this makes them ideal to control unsprung mass dynamics as well as the sprung mass dynamics. Semi-active suspensions have other advantages: 1) size and weight: their simple architecture makes for light and compact products. 2) Energy requirements: being passive, their energy requirement is negligible in automotive applications. On the other hand, their main disadvantage is the impossibility of exerting active forces. They are inherently passive devices which only modulate the dissipation term and thus cannot exert static forces.

Several technologies implement the semi-active philosophy: electro-hydraulic (EH), in which a valve is used to control the damping; magneto-rheological (MR) and electro-rheological (ER), in which either a magnetic or an electric field is used to change the viscosity of a fluid.

The control literature is rich and diverse. The bulk of the research studies the dynamic properties and algorithms

aimed at controlling the chassis and wheel vertical dynamics: the most successful approach is the sky-hook concept. It was first developed in Karnopp (1995) and then refined and extended in a number of other contributions (such as Savaresi et al. (2007); Song et al. (2005)). Other authors have investigated more modern control methods as Linear Parameter Varying Control (such as in Sammier et al. (2003); Poussot-Vassal et al. (2008); Gaspar et al. (2007); Zin et al. (2006)). Other approaches include Fuzzy control (Tang et al. (2017)). Most researchers base the control system design on the quarter car model and the validation is often based on a quarter car test rig Tang et al. (2017). Working on a quarter-car test rig has many advantages. It allows one to easily quantify the effect of the control strategy and compare results in a repeatable way. On the other hand, the quarter car test rig does not account for many aspects that influence the ride quality: pitch and roll dynamics to cite two. When it comes to study the effect of the suspension on the roll and pitch dynamics, the most common approaches consider active suspensions (as in Brezas and Smith (2014)). The literature on semi-active suspensions is scarce and works presenting an experimental validation even scarcer: Fleps-Dezasse et al. (2018) describes an LPV control augmented with a feedforward term a to improve roll stability; Brezas et al. (2015) designs an LQ controller with the same objective.

The paper presents a complete semi-active controller with the aim of proving road isolation and at the same time improve the roll and pitch stability. As shown in Figure 1, we propose a hierarchical structure where four independent low level controllers impose a desired equivalent damping on the four dampers; four mid-level controllers, based on

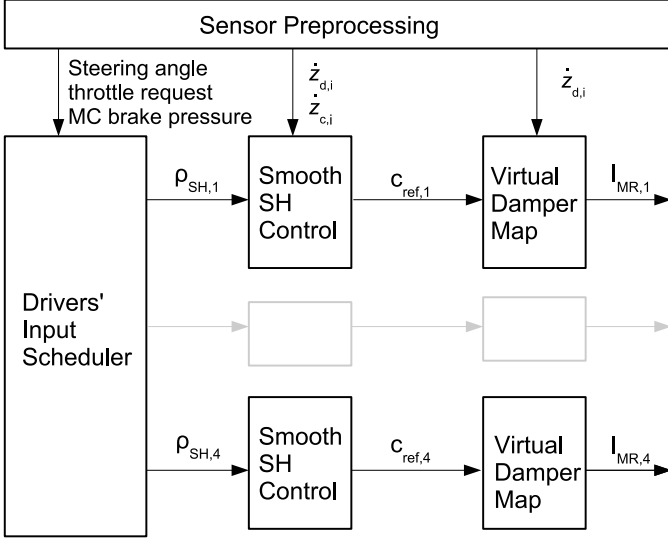


Fig. 1. Schematic representation of the proposed control system.

the sky-hook idea, compute the desired equivalent damping based on each corner movement; finally, a centralized high-level controller schedules the parameter of the sky-hook algorithms taking into account the driver's input.

The main contributions of this work revolve around three points:

- The classical Sky-Hook strategy is adapted with the objective of reducing corner jerk. Conventional sky-hook control tends to be detrimental to the corner jerk. The present work, taking inspiration from Ahmadian et al. (2004), proposes and analyzes a smooth implementation of the two-state logic.
- The Sky-Hook strategy is scheduled based on the drivers input. The scheduling of the SH parameters allows for an improvement of the roll and pitch stability without limiting the filtering properties.
- The proposed solution is extensively tested on an instrumented vehicle showing that an adequate damping control can effectively increase the perceived roll and pitch stiffness during aggressive maneuvers.

2. CORNER CONTROL

Figure 1 represents the overall architecture. The controller has 4 main elements: the centralized system comprises the sensor preprocessing, and drivers's input scheduling; at the corner level, we have the Smooth SH controller and the Virtual Damper Map.

In this work, we assume a classical sensors configuration:

- 4 corner single-axis body accelerometers (used for control);
- 4 corner suspension potentiometers (used for control);
- steering angle (δ), Master Cylinder pressure and driver's throttle request (used for control);
- 4 wheel single-axis accelerometers (used for analysis and validation);
- a central vertical accelometer (used for analysis and validation);

- high accuracy GPS antenna (used for analysis and validation).

The pre-processing module processes the signals available for control in order to provide the vertical chassis speed z_i and the damper stroke speed $z_{d,i}$. The processing is done through linear pass-band filters (see Savaresi et al. (2010)).

The vehicle is equipped with 4 MR dampers. Each damper has a current controller that is outside the scope of this paper

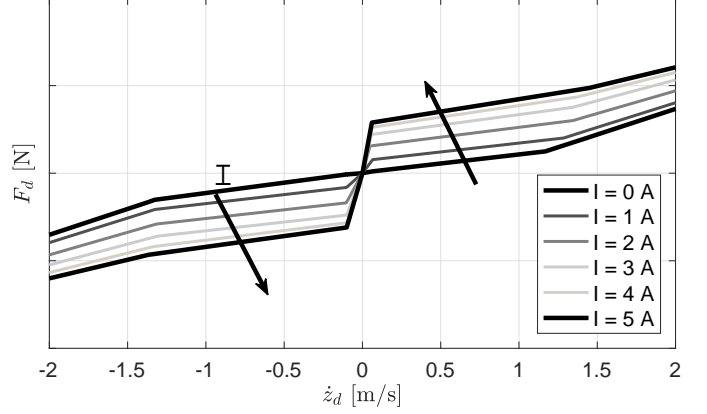


Fig. 2. Force-stroke speed static characteristic of the MR damper.

Each corner module takes the available measures relative to that corner and determines the damper current according to a modified SH philosophy.

2.1 Virtual Damping Map

Figure 2 shows the nominal characteristic of the MR damper¹: it clearly has a maximum and minimum force that it can exert and it is highly nonlinear. The nonlinearity is potentially troublesome at low stroke speeds where the characteristic has a very steep force increase. This is equivalent to having an extremely high damping. As it will become clearer later, this is useful to improve roll and pitch stability, but it makes it difficult to tune the SH controller to provide good heave stability when driving straight. In fact, the classical SH rationale has been developed under the assumption of more regular curves.

The Virtual Damper Map module has the goal of regularizing these curves. The user can draw a family of desired damping characteristics (parametrized by a normalized $c_{ref,i}$) and the module computes the current that yield the desired force for the current stroke speed. The computation is based on inverting the static map of Figure 2. A detailed analysis of the inversion problem reveals that the map is not invertible for $z_{damper} = 0$; to avoid chattering, a dead-band is implemented. In the z_{damper} dead-band the current is held at the previous value. Thanks to this low level control, the SH controllers use, as a control variable, the normalized $c_{ref,i}$ for each corner. It is worth noticing that drawing these curves is still more of an art, than a

¹ Note that, through the paper, some units of measure have been normalized for confidentiality reasons. In particular, the stroke are expressed in thr $[-1, 1]$ range, where the extremes are the end-stroke position.

science. The simplest solution is to smooth the angles in the nominal characteristics with a spline.

2.2 Smooth Sky-Hook

The SH is a common approach to the comfort oriented control of four wheeled vehicles. Many implementations exist, one of the most common is the so-called two-state SH control. In this implementation, the damping is computed as

$$c_{ref} = \begin{cases} c_{min} & \text{if } \dot{z}\dot{z}_d \leq 0 \\ c_{max} & \text{if } \dot{z}\dot{z}_d > 0 \end{cases} \quad (1)$$

In the above control law, c_{min} and c_{max} represent the minimum and maximum actuatable damping. The control algorithm of Equation (1) has many advantages: it is computationally efficient and simple to understand for practitioners. Unfortunately, the control law cannot be implemented as formulated. Figure 3 illustrates this limitation. The figure represents the equivalent damping as a function of \dot{z} and \dot{z}_d . The control algorithm is sub-

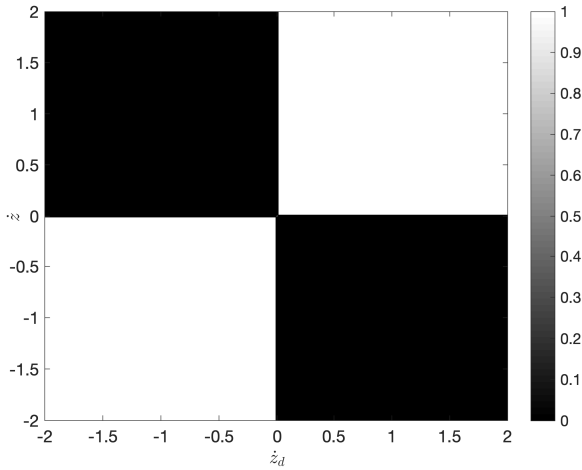


Fig. 3. Graphical representation of the two-state SH law.

ject to unwanted chattering, when either \dot{z} or \dot{z}_d change sign. Formally speaking, the switching corresponding to a change of sign of \dot{z}_d should not affect the chassis dynamics as in those conditions, the damper force is negligible. However, inevitable sensor noise and computation delay may cause a non negligible jerk. The industrial practice to solve these issues is to introduce dead-zones and hysteresis. These heuristics tend to be difficult and time consuming to tune. Here we propose a linear approximation of the two-state SH that guarantees smoothness also in case of noise and delays while introducing a limited number of new parameters. The approximation is:

$$c_{ref} = \underset{c_{ref} \in [c_{min}; c_{max}]}{\text{sat}} (k_{sky} \dot{z}\dot{z}_d + c_{nom}) \quad (2)$$

Figure 4 depicts the resulting damping on the \dot{z} , \dot{z}_d plane. From figure, it is clear how the proposed solution represents a smoothing of the two state approach where the smoothing happens on hyperbolic isoclines. Alternatively, the two-state implementation could be interpreted as the smooth SH law for $k_{sky} \rightarrow \infty$. The smoothed SH introduces two parameters, the gain k_{sky} and the nominal damping c_{nom} . These parameters will be employed by the

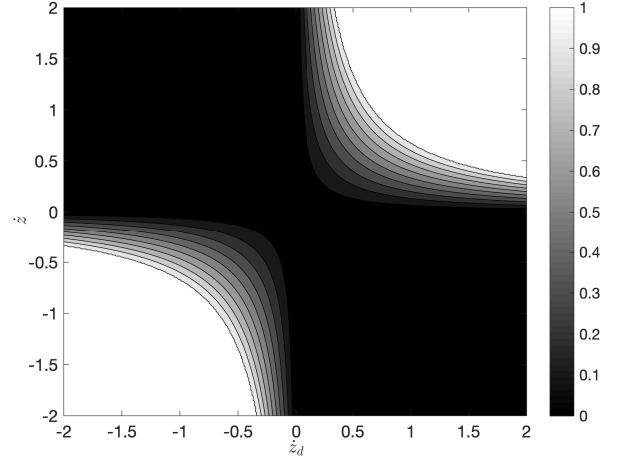


Fig. 4. Graphical representation of the smooth SH law for $c_{nom} = 0$.

Driver's Input Scheduling to influence the roll and pitch dynamics.

3. DRIVER'S INPUT SCHEDULING

The SH logic presented in the previous section is designed to control the chassis movement. Strictly speaking, the SH rationale is designed to limit the chassis velocity. As such, it should also be able to stabilize the chassis during load transfer phenomena. In fact, dynamic load transfer caused by drivers' input excite the same dynamics as road disturbances. However, there are two main differences: 1) dynamic load transfer phenomena act at lower frequency than road disturbances (usually considered a broad band excitation) 2) the dynamic load transfer is predictable based on the driver's input.

In this section, we augment the closed-loop SH with a scheduling approach that acts on the SH tuning parameters to improve the stability of the chassis dynamics during maneuvers. One of the main advantages of the scheduling approach is that it does not override the SH strategy and thus road disturbance filtering is always active. The proposed scheduling strategy is model-based and composed of two terms: a longitudinal scheduling and lateral scheduling. The two terms act according to the same rationale: they increase the nominal damping in (2) during load transfer transients. The scheduling acts only during the transients because that is the only part of the maneuver that the damping modulation can influence. Once the load transfer has occurred, the SH control is active with the nominal parameters so not to negatively affect road filtering.

3.1 Lateral Scheduling

When the vehicle is steered, the consequential lateral acceleration produces a load transfer to the outer part of the vehicle which generates a rolling movement. By increasing the damping during this transient, it is possible to slow down the roll dynamics and improve the perceived stability. This simple idea is implemented in three steps:

- (1) The velocity of the vehicle and the steering angle are fed into a static lateral dynamic model to compute a lateral acceleration

$$a_y^* = \frac{v^2 \delta}{K_{us} v^2 + L} \quad (3)$$

where K_{us} is a reference understeering coefficient (see Kiencke and Nielsen (2000)) and L is the wheel base of the vehicle

- (2) the modeled lateral acceleration is filtered through a high pass filter to get $a_{y,HP}^*$
- (3) an additive term to the nominal damping in (2) is computed as

$$c_{nom}^{lat} = K_{lat} \|a_{y,HP}^*\| \quad (4)$$

where K_{lat} is the lateral gain scheduling.

It is interesting to comment the choice of using a model-based approach rather than employing the lateral acceleration. The main reason is that the modeled lateral acceleration, by neglecting the vehicle and tyre relaxation dynamics, anticipates the measured lateral acceleration. Thus, the model-based approach yields a faster adaptation to load transfer. Figure 5 compares the modeled and measured acceleration acquired in a track lap. From figure, one

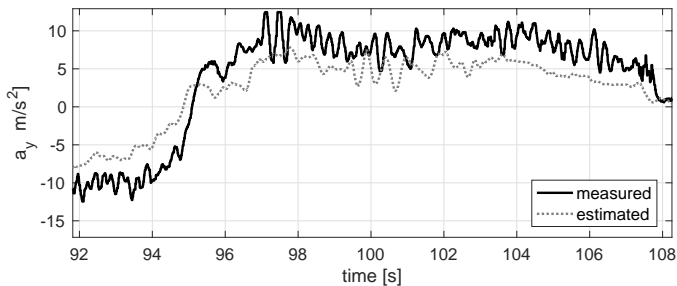


Fig. 5. Comparison of the modeled and measured lateral acceleration during a segment of a track lap.

can see that the modeled acceleration anticipates the load transfer. In this example, the phase between the modeled lateral acceleration and the measured one can be as long as 100 ms. On the other hand, it should also be noted that the model is not always accurate, especially at high lateral acceleration when the K_{us} approximation breaks down. This lack of accuracy is not critical as it can be managed through the tuning of the lateral gain K_{lat} .

3.2 Longitudinal Scheduling

The longitudinal scheduling follows the same rationale as the lateral one. It uses a simplified, steady state longitudinal acceleration model, that is fed with the vehicle velocity, throttle request and MC braking pressure. Also in this case a high pass filter and a scheduling gain K_{long} are employed.

The proposed longitudinal and lateral scheduling may be easily adapted to act differentially on the four corners. For brevity's sake, in this contribution, we assume to act on all corners in the same way.

4. EXPERIMENTAL VALIDATION

In this section, the proposed semi-active suspension control system is implemented and validated on a supercar.

The control algorithm runs at 1 kHz and the MR current drivers guarantee a closed loop current control bandwidth of around 150 Hz. Directly working on an instrumented car, rather than a test-rig, has advantages and disadvantages. The main disadvantage is that it is difficult to rigorously compare different tunings and quantify performance as the conditions are never repeatable. On the other hand, performing tests on the vehicle is the only way to assess the coordinated control of the four corners while performing aggressive maneuvers.

The discussion is organized following the module discussions.

4.1 Smooth SH implementation

The first analysis concerns the comparison of the two-state SH against the smooth SH control logic. Figure 6 compares the results of a sweep test simulation (performed on a validated model that also considers sensor noise) in terms of vertical chassis acceleration. The plot focuses on the sprung mass resonance frequency, where the SH control is most beneficial. In this case, using a simulation envi-

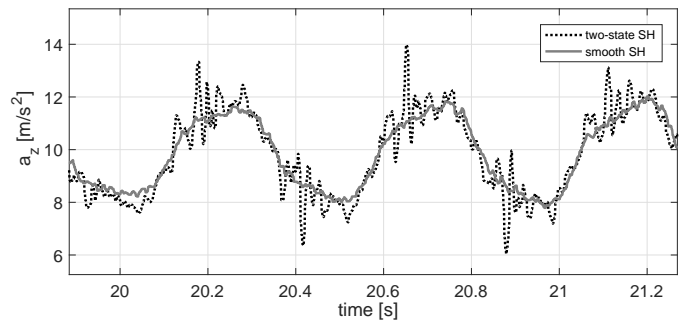


Fig. 6. Comparison of the simulated vertical body acceleration for the two-state SH control and the Smooth SH control.

ronment guarantees the repeatability needed to compare the two settings. The figure clearly shows that the two-state switching generates very large acceleration peaks. The introduction of the smoothing strategy completely removes these peaks without affecting the amplitude of the sprung mass resonances.

As already pointed out, extracting a quantitative performance assessment from data collected on the real vehicle is challenging. One way to compare settings is through the body and wheel vertical acceleration spectra. In order to obtain a sound comparison, tests performed at the same constant speed on the same road are compared. Figure 7 quantifies the effect of the active control with respect to the c_{min} case. From figure, it is easy to assess that the semi-active control is capable of considerably reducing the oscillations around the unsprung mass resonance without negatively affecting the wheel dynamics and only marginally affecting the filtering at higher (≈ 20 Hz) frequency.

Following the proposed approach, by tuning the two parameters of the corner module, one can obtain a more comfortable vehicle or a more *sporty* one. Figure 8 shows the spectra of two proposed tunings obtained with the collaboration of professional drivers. Apparently, the comfort

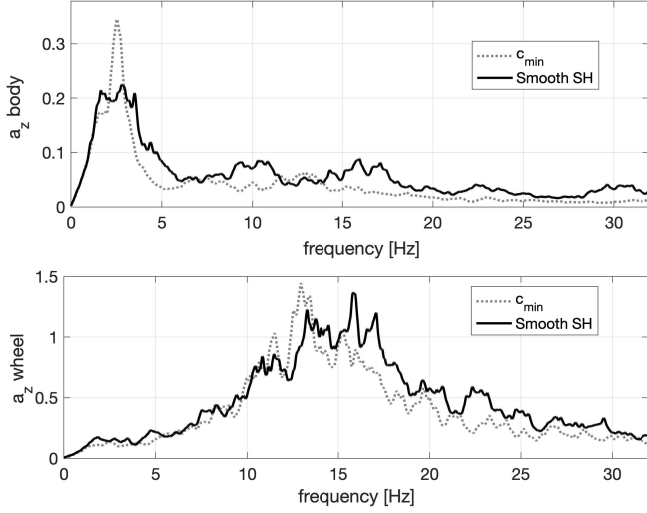


Fig. 7. Comparison of the vertical body and wheel acceleration spectra for the Smooth SH control and c_{min} .

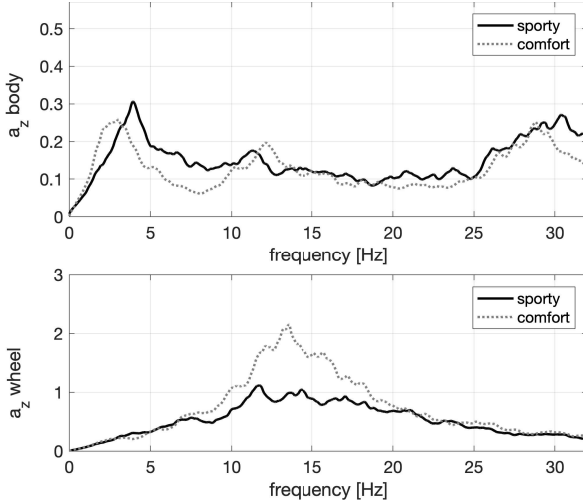


Fig. 8. Comparison of the vertical body and wheel acceleration spectra for the two proposed tunings.

configuration provides a more stable chassis around the unsprung mass resonance, at the cost of higher oscillations of the wheel around the unsprung mass resonance. This confirms that drivers associate a sporty feeling to a more stable wheel.

4.2 Driver's Input Scheduling

The experiments discussed so far were performed at a constant speed; the chassis movements were caused by road input. In the remainder of the section, we will consider the effect of the proposed scheduling. First in case of longitudinal maneuvers and then in case of lateral excitation. Figure 9 plots the front and rear left suspension stroke and the reference damping for a hard braking maneuver executed at around 1.2 g. The stroke indicates that without the longitudinal scheduling, the pitch dynamics is considerably faster. During the pitching phase, the Smooth SH increases the damping. This increase is however not enough and the fast pitching movement is perceived as unpleasant by

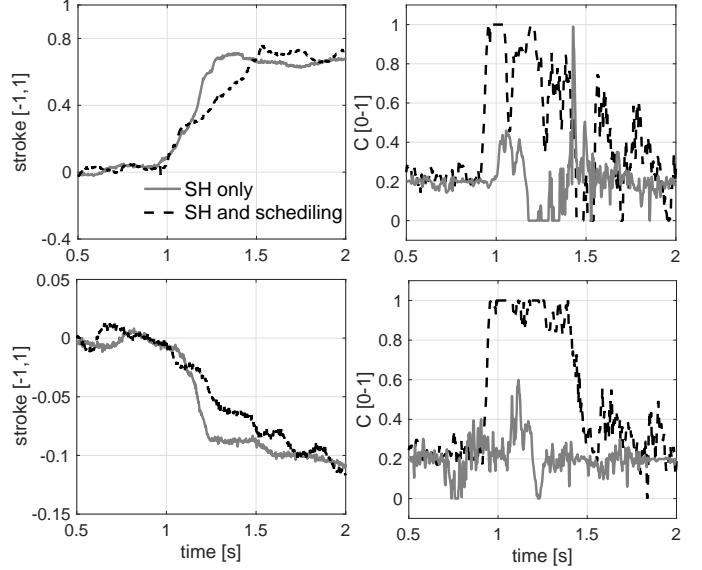


Fig. 9. Comparison of two hard braking maneuvers: only the SH module active, and the complete scheduled controller. Front and rear left strokes (left plots) and front and rear reference damping (right plots).

the driver. The scheduled controller, on the other hand, rapidly increases the damping yielding a slower pitch dynamics. Overall, the scheduling is capable of slowing the pitch dynamics by a factor of 2 without affecting the road filtering one when the load transfer transient is over.

The performance comparison in terms of lateral dynamics is more challenging. The driver changes the steering input depending on how the vehicle responds. For this reason, Figure 10 first compares the stroke dynamics during a repeatable steer step with different open-loop suspension tunings and subsequently Figure 11- 12 show the behavior of the scheduled SH controller during a high speed maneuver.

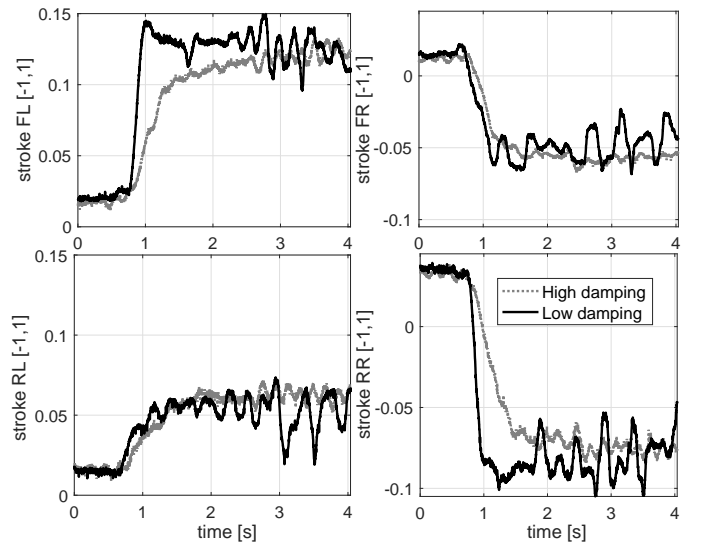


Fig. 10. Suspension stroke behavior during a righthand steering step for two suspension settings: high damping and low damping.

Figure 10 shows two 120 degrees steering wheel step maneuvers performed at 75 km/h. It is interesting to note that the damping setting of the suspension has a large impact on the load transfer transient. The front left suspension settling time is increased from 0.3 s to 1.2 s. This proves that MR suspensions can increase the perceived roll stiffness during transients and consequently improve stability. It is also interesting to notice that, in this righthand turn, the suspensions that mostly influence the roll dynamics are the front outer and rear inner ones. This leaves margin to use the calibration of the suspensions on the other diagonal to modify other vehicle dynamics aspects.

Figures 11-12 finally plots the results of during a chicane performed with a velocity varying from 80 km/h to 180 km/h and maximum longitudinal and lateral accelerations of respectively -1 g and 1.1 g. One can see that,

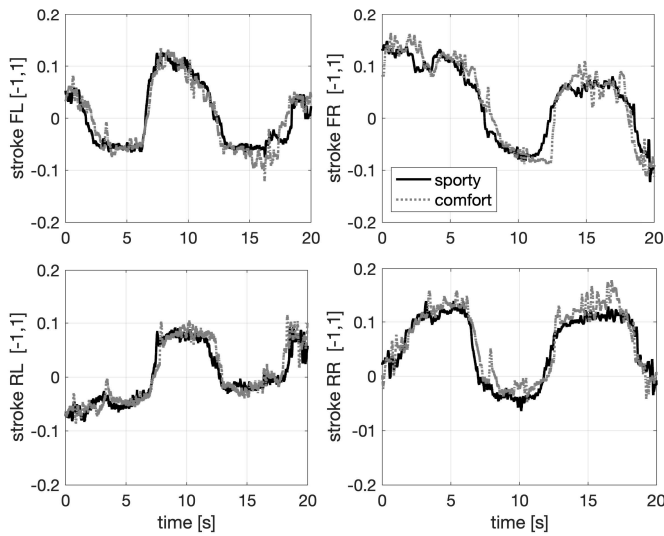


Fig. 11. Suspension stroke behavior during an aggressive chicane maneuver for two different control tunings.

as expected, the scheduling increases the damping only during the transients, during the steady state part of the maneuver the damping returns to the nominal behavior dictated by the Smooth SH control. Furthermore, the comfort setting, during the second part of the maneuver, yields a less stable roll dynamics. The front right suspension compresses more rapidly at around the 12 s mark; this shift is clearly also seen in the reference damping dynamics.

5. CONCLUSIONS

This paper presents a full body semi-active suspension control for a supercar. The controller is built around the idea of scheduling four independent SH controllers depending on the lateral and longitudinal jerk. The SH controller is implemented in a smooth version to avoid unnecessary switching. The scheduling variables are model-based, this allows to have a prompter response to the driver's input.

The proposed approach is extensively validated on an instrumented super car. Different maneuvers are considered and the effect different tuning choices illustrated.

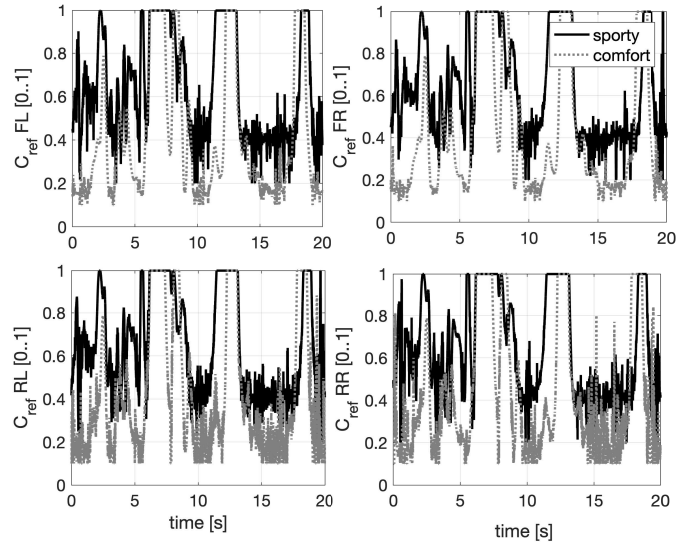


Fig. 12. Reference suspension damping behavior during an aggressive chicane maneuver for two different control tunings.

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