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A Comprehensive Comparison of Chassis Systems Coordination Approaches

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Abstract: One of the main concerns of automotive industry is autonomous vehicles. Not only equipment suppliers but also new entering actors are racing to develop safer automotive systems. Consequently, the car manufacturer has to integrate different systems to control the same vehicle. This paper discusses the different architectures adopted to handle chassis systems interactions. Several car manufacturers study the influence of each system on the overall vehicle. They then add an additional coordination layer downstream the different systems to mitigate their possible conflicts. Here we propose a new approach that consists on managing systems interaction upstream the different systems. Coordination is based on control allocation techniques that take into account several vehicle states to find optimal commands distribution. Both approaches have been compared. Simulations showed the benefits that could offer the upstream approach in terms of safety and flexibility. The relevance of this approach is expected to grow with the increasing number of chassis systems.

Keywords: Vehicle motion control, chassis systems coordination, over-actuated systems, control allocation architectures, tire modelling, AMESim[®] simulations.

1. INTRODUCTION

Safety is a major criterion for car manufacturers. In the late 1970's active safety systems were introduced as one of the biggest advances in automotive industry [1]. Thanks to the advances in electronic and software engineering, an increasing number of assistance systems have been proposed. A natural evolution of Advanced Driver Assistance Systems (ADAS) seems to be fully autonomous vehicles. Here, all integrated systems should collaborate to control the vehicle. The main difficulty is the fact that these systems are developed independently by various stakeholders. No matter how efficient assistance systems can be, when put in the same vehicle, one system could compromise the performance of the other. The car manufacturer is therefore obliged to develop an overall chassis systems coordination architecture. One common industrial practice is to study off-line the influence of each subsystem on the overall vehicle, then develop rule-based algorithms to favour subsystems depending on prior objectives [1],[2]. Each subsystem is integrated directly with its controller. The coordination layer is then located downstream the different systems.

Several scenarios should be established to be able to foresee the possible conflicts. However, autonomous vehicles need a large amount of subsystems to operate at the same time. Conflicts would be more numerous and more complex to handle, and it would be hard for automotive engineers to foresee all the possible scenarios. A search for an optimal solution should be carried out. This can only be done if interactions are mathematically formalized and managed before allocating the commands. The coordination should be ensured upstream the subsystems.

Models that gives additional insights on the dynamic

behaviour by taken into account the couplings should be adopted. One of the most important couplings for ground vehicles are those of tire forces. Plenty of coupled tire models exist in the literature. However, whether these models are empirical [3], or they are too complex to handle for control synthesis problems [4]. This is the main reason of the development of a new tire model in [5]. Here the model proposed is linear with varying physical parameters which makes it particularly suitable for Global Chassis Control (GCC).

In this paper, we compare the downstream approach that represent the current situation of most manufacturers, and the upstream approach that should be adopted in the future. To do so, we chose a simple example of a vehicle equipped by an Active Rear Steering (ARS) system and a braking-based Electronic Stability Program (ESP). The comparison is performed in AMESim[®] environment to provide more realistic simulations. The upstream approach showed a smarter way to use the different chassis systems with respect to the downstream approach. These advantages are expected to grow when supplementary systems will be added.

We start in Section 2. by detailing the downstream approach in the context of ARS-ESP coordination. In Section 3., the upstream approach is presented. Section 4. presents and compare results obtained from the AMESim[®] environment. A discussion about the relevance of the upstream approach and its implications is provided in Section 5. Conclusions and future works are outlined in Section 6.

2. THE DOWNSTREAM APPROACH

Here, chassis systems are developed independently to control one specific physical variable, here, the yaw rate.

Controllers are then based on transfer functions from a specific actuator command to directly one of the vehicle control axes. In this configuration, one can only act downstream the different subsystems. Fig.1 illustrates this approach in the context of ARS-ESP coordination.

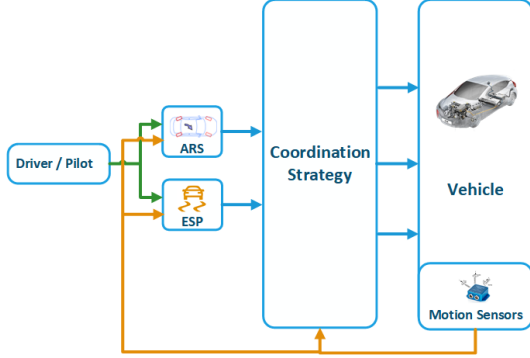


Fig. 1 Structure of the downstream coordination approach (adapted from [2]).

2.1 System Modelling

Both ARS and ESP act on the lateral behaviour of the car. In this context, most of chassis systems developers rely on a reduced vehicle model that depicts well the vehicle's lateral behaviour, which is the bicycle model [6]. This is mainly due to the fact that chassis systems' standalone controllers are developed independently by different suppliers [2]. Tire forces are expressed as follows:

$$F_{y_f} = C_{\alpha_f} \alpha_f, \quad (1)$$

$$F_{y_r} = C_{\alpha_r} \alpha_r. \quad (2)$$

With:

- F_{y_f} and F_{y_r} : the front and rear lateral force,
- C_{α_f} and C_{α_r} : the front and rear equivalent cornering stiffnesses,
- α_f and α_r : the front and rear equivalent side-slip.

Where:

$$\alpha_f = \delta_f - \frac{V_y + \dot{\psi} l_f}{V_x}, \quad (3)$$

$$\alpha_r = \delta_r - \frac{V_y - \dot{\psi} l_r}{V_x}. \quad (4)$$

With:

- δ_f and δ_r : the front and rear steering angles,
- l_f and l_r : the distance between the front/rear axle and the vehicle's centre of gravity,
- V_x : longitudinal velocity of the vehicle,
- V_y : lateral velocity of the vehicle,
- $\dot{\psi}$: yaw rate of the vehicle.

The equations of motion can be expressed using simply Newton's laws of motion as follows:

$$M (\dot{\psi} V_x + s V_y) = F_{y_f} + F_{y_r}, \quad (5)$$

$$J_z s \dot{\psi} = F_{y_f} l_f - F_{y_r} l_r + M_{esp}. \quad (6)$$

With:

- M : vehicle's mass,
- J_z : vehicle's yaw moment of inertia,
- M_{esp} : yaw moment generated by the ESP system,
- s : Laplace operator.

2.2 Control synthesis

2.2.1 Yaw Rate Target

The front steering angle δ_f is used to generate the yaw target. While the dynamical bicycle model is used to synthesis the controllers by taking into accounts the transient behavior, the statical model can be used to generate the yaw rate target, namely, only a gain relating the yaw rate to the front steering angle and depending on the longitudinal speed. In [1], this technique has been used, and additional parameters have been used to tune the response and improve the vehicle's handling.

2.2.2 Coordination Strategy

Here, the influence of the ARS and the ESP on the yaw rate are compared. Not only the potential of each system is taken into account, but additional parameters could be added as tire wear, energy consumption, or the feeling generated. In [7] for example, the Electric Power-Assisted Steering (EPAS) has been compared to the Electronic Stability Program (ESP). The EPAS system has been prioritized in comfort mode, while the ESP system was activated only in an emergency mode. This is mainly due to the fact that the EPAS system improves the vehicle handling which gives better comfort. Also, the ESP system uses the brakes which is in one hand less comfortable, and in the other hand, accelerates tires wear.

Because the ARS system has similar benefits as the EPAS system, the same strategy will be adopted in this paper. Therefore, the ARS system is prioritized and the ESP system will only activates in two cases: if the driver brakes brutally or if the ARS system is saturated.

2.2.3 Subsystems Controllers

Each subsystem has its own controller. Each controller is based on a different transfer function that links the corresponding command directly to the vehicle's yaw rate. The detailed design of these controllers goes beyond the scope of this paper. However, an important remark should be mentioned. Let us note $K_{ars}(s)$ the ARS controller, $K_{esp}(s)$ the ESP controller, and $P(s)$ the plant model. $K_{ars}(s)P(s)$ and $K_{esp}(s)P(s)$ would be stable. But when put together in a downstream configuration, both controllers are placed in parallel. Therefore, $[K_{ars}(s) + K_{esp}(s)]P(s)$ also should be stable. Controllers might have to be redesigned.

3. THE UPSTREAM APPROACH

When each controller is designed independently to act directly on vehicle dynamics, we usually omit a major constraint level, which is tire potential. Designing

each system without considering the possible interactions gives a wrong information about the remaining potential of tires. This could lead to high requests of tire forces when these tires are actually saturated by another system [5]. For future ground vehicles with a higher number of chassis systems, we should bring up the interactions management to a higher level. This can be illustrated by the architecture structure shown in Fig 2.

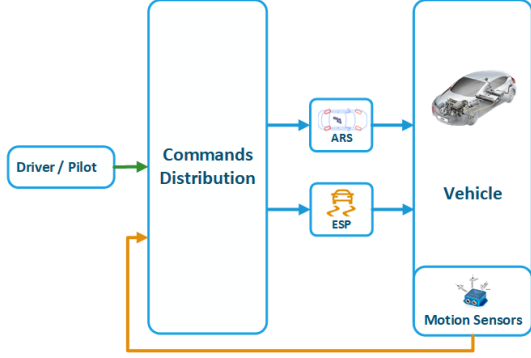


Fig. 2 Structure of the upstream distribution approach (adapted from [2]).

3.1 System Modelling

Interactions should be mathematically described. The modelling should be more complete. A four-wheeled vehicle model should be considered, and tire forces interactions should be taken into account.

3.1.1 The four-wheeled vehicle model

As long as only the ARS and the ESP systems are concerned, a planar vehicle model can be considered [7]. In fact, these systems do not control the vehicle's vertical motions. Variations in vertical forces should be taken into account only in the control allocation strategy. Using Newton's laws of motion, we find:

$$M s V_x = F_{x_{tot}}, \quad (7)$$

$$J_z s \dot{\psi} = M_{z_{tot}}. \quad (8)$$

Where:

$$F_{x_{tot}} = (F_{x_{fl}} + F_{x_{fr}}) \cos(\delta_f) + F_{x_{rl}} + F_{x_{rr}} \quad (9)$$

And:

$$\begin{aligned} M_{z_{tot}} = & (F_{x_{fl}} + F_{x_{fr}}) l_f \sin(\delta_f) \\ & + (F_{x_{fr}} - F_{x_{fl}}) \frac{E}{2} \cos(\delta_f) \\ & + (F_{x_{rr}} - F_{x_{rl}}) \frac{E}{2} - F_{y_r} l_r \end{aligned} \quad (10)$$

Where:

- $F_{x_{f,r}}$: is the front-right braking force,
- $F_{x_{f,l}}$: is the front-left braking force,
- $F_{x_{r,r}}$: is the rear-right braking force,
- $F_{x_{r,l}}$: is the rear-left braking force,
- $F_{y_r} = F_{y_{rl}} + F_{y_{rr}}$ is the lateral rear force.

Note that we have considered only controllable forces. Moreover, in equation (7), the relative motion term,

which make the equation non-linear, has been ignored. In fact, using Relative Gain Array (RGA) analysis, it was shown that it has a minimum effect [7]. The idea is then to design a high-level controller based on equations (7) and (8), and use control allocation techniques to distribute the commands by respecting equations (9) and (10). Tire behaviour and limits should be taken into account.

3.1.2 The new linear tire model with varying parameters

A new tire model has been developed recently to face global chassis control challenges [5]. This model is linear, depicts well the combined slip, and depends on physical varying parameters that could be estimated on-line. It has the same structure as equations (1) and (2), but the stiffness is variable:

$$F_x = C_s^*(\alpha, \mu, F_z) \kappa, \quad (11)$$

$$F_y = C_\alpha^*(\kappa, \mu, F_z) \alpha. \quad (12)$$

where:

- κ : the longitudinal slip,
- μ : the friction coefficient,
- F_z : the vertical load,
- $C_s^*(\alpha, \mu, F_z)$: the tire varying longitudinal stiffness with respect to the side-slip α , the friction coefficient μ , and the vertical load F_z ,
- $C_\alpha^*(\kappa, \mu, F_z)$: the tire varying cornering stiffness with respect to the longitudinal slip κ , μ , and F_z .

Detailed expressions of $C_s^*(\alpha, \mu, F_z)$ and $C_\alpha^*(\kappa, \mu, F_z)$ can be found in [5].

Using these expressions, the longitudinal force is penalized by the side-slip, at the same way that the lateral force is penalized by the longitudinal slip. The combined slip, and therefore interactions at the tire level are taken into account. The overall potential of the tire is limited. Actually tire forces are constrained by a friction ellipse [3]. This can be expressed by means of the following equations [5]:

$$F_x \leq \sqrt{(\mu F_z)^2 - F_y^2} \quad (13)$$

$$F_y \leq \sqrt{(\mu F_z)^2 - F_x^2} \quad (14)$$

3.2 Control Synthesis

3.2.1 High-Level Control

The main goal is to control the yaw rate. However, braking forces could also affect the longitudinal velocity. As long as over-actuated systems are concerned, different solutions can be found to control the yaw rate. We can then choose a specific solution in order to fulfill a secondary objective. Here, the secondary objective would be the minimization of the influence on the longitudinal velocity. To do so, the high level control considers both equations (7) and (8). Two PI controllers are chosen to calculate the generalized forces $F_{x_{tot}}$ and $M_{z_{tot}}$. The integral effect in the high-level controllers is added to face the possible environmental disturbances.

3.2.2 Control Allocation

The control allocation problem can be defined as follows [8]: find the control vector, $\vec{\delta} \in \mathbb{R}^n$ such that

$$\mathbf{B}\vec{\delta} = \vec{d}_{des} \quad (15)$$

subject to

$$\begin{cases} \vec{\delta}_{min} \leq \vec{\delta} \leq \vec{\delta}_{max} \\ \dot{\vec{\delta}} \leq \dot{\vec{\delta}}_{max} \end{cases} \quad (16)$$

where $\mathbf{B} \in \mathbb{R}^{m \times n}$ is a control effectiveness matrix, $\vec{\delta}_{min} \in \mathbb{R}^n$ and $\vec{\delta}_{max} \in \mathbb{R}^n$ are the lower and upper position limits, respectively, $\dot{\vec{\delta}} \in \mathbb{R}^n$ is the control rate, $\dot{\vec{\delta}}_{max} \in \mathbb{R}^n$ is the maximum control rate, $\vec{d}_{des} \in \mathbb{R}^m$ are the desired accelerations, n is the number of control effectors, and m is the number of axes to control ($n > m$).

Here, $\vec{\delta}$ is mainly composed by controllable tire forces:

$$\vec{\delta} = \begin{bmatrix} F_{x_{fl}} \\ F_{x_{fr}} \\ F_{x_{rl}} \\ F_{x_{rr}} \\ F_{y_r} \end{bmatrix} \quad (18)$$

$\vec{\delta}_{min}$ and $\vec{\delta}_{max}$ reflect tire limits with respect to the friction ellipse concept [3].

As we have mentioned, the roll or the pitch cannot be controlled by the ARS or the ESP systems. However, vertical motions should be taken into account as they have a significant influence on tire forces [3],[4]. The vertical load at the level of each tire can be defined as [7]:

$$F_{z_{fl}} = \frac{1}{2}Mg\frac{l_r}{L} - \frac{1}{2}Ma_x\frac{h}{L} - \frac{1}{2}Ma_y\frac{h}{E} \quad (19)$$

$$F_{z_{fr}} = \frac{1}{2}Mg\frac{l_r}{L} - \frac{1}{2}Ma_x\frac{h}{L} + \frac{1}{2}Ma_y\frac{h}{E} \quad (20)$$

$$F_{z_{rl}} = \frac{1}{2}Mg\frac{l_r}{L} + \frac{1}{2}Ma_x\frac{h}{L} - \frac{1}{2}Ma_y\frac{h}{E} \quad (21)$$

$$F_{z_{rr}} = \frac{1}{2}Mg\frac{l_r}{L} + \frac{1}{2}Ma_x\frac{h}{L} + \frac{1}{2}Ma_y\frac{h}{E} \quad (22)$$

where:

- g : gravitational acceleration,
- a_x : vehicle's longitudinal acceleration,
- a_y : vehicle's lateral acceleration,
- h : height of vehicle's centre of gravity.

Here, we assume that μ can be estimated [9].

The desired acceleration \vec{d}_{des} are actually the desired generalized forces in our case:

$$\vec{d}_{des} = \begin{bmatrix} F_{x_{tot}} \\ M_{z_{tot}} \end{bmatrix} \quad (23)$$

\mathbf{B} represents the geometric relations between the vehicle and its tires. Here we use equations (9) and (10) to find:

$$\mathbf{B} = \begin{bmatrix} \cos(\delta_f) & \cos(\delta_f) & 1 & 1 & 0 \\ b_{2,1} & b_{2,2} & -\frac{E}{2} & \frac{E}{2} & -l_r \end{bmatrix} \quad (24)$$

where:

- $b_{2,1} = l_f \sin(\delta_f) - \frac{E}{2} \cos(\delta_f)$,
- $b_{2,2} = l_f \sin(\delta_f) + \frac{E}{2} \cos(\delta_f)$.

Different techniques can be used in order to calculate $\vec{\delta}$. A large overview can be found in [8]. In order to give a reliable comparison between both architectures in terms of the location of coordination strategies, the ARS is also prioritized here with respect to the ESP at first. This can be done by using the “*daisy-chain*” method [7], [8]. When only the ARS is activated, the allocation is made directly using the following equation:

$$F_{y_r} = -\frac{1}{l_r}M_z \quad (25)$$

Unlike the downstream approach, the transition to the second control effectors set is done not only by considering ARS actuator saturation, but also rear tires saturation. In addition, only the error that arises between the desired forces and those generated by the preselected control effectors is transmitted to the second set of control effectors. A second sub-problem can be then defined considering only longitudinal forces:

$$\begin{bmatrix} F_{x_{tot}} \\ M_z + l_r F_{y_r} \end{bmatrix} = \mathbf{B}_2 \begin{bmatrix} F_{x_{fl}} \\ F_{x_{fr}} \\ F_{x_{rl}} \\ F_{x_{rr}} \end{bmatrix} \quad (26)$$

$$\text{where } \mathbf{B}_2 = \begin{bmatrix} \cos(\delta_f) & \cos(\delta_f) & \frac{1}{2} & \frac{1}{2} \\ b_{2,1} & b_{2,2} & -\frac{E}{2} & \frac{E}{2} \end{bmatrix}.$$

Brake forces distribution consists then on inverting the non-square matrix \mathbf{B}_2 . Weighting matrices could also be added to favour one tire over another depending on its potential. *Weighted pseudo-inverse* method can be then used after the “*daisy-chain*” computation [7]. The weighting matrix chosen is expressed as follows:

$$\mathbf{W} = \begin{bmatrix} w_{fl} & 0 & 0 & 0 \\ 0 & w_{fr} & 0 & 0 \\ 0 & 0 & w_{rl} & 0 \\ 0 & 0 & 0 & w_{rr} \end{bmatrix} \quad (27)$$

where $w_{ij} = \frac{1}{\sqrt{(\mu_{ij}F_{z_{ij}})^2 - F_{y_{ij}}^2}}$. We finally find:

$$\begin{bmatrix} F_{x_{fl}} \\ F_{x_{fr}} \\ F_{x_{rl}} \\ F_{x_{rr}} \end{bmatrix} = \mathbf{B}_2^\# \begin{bmatrix} F_{x_{tot}} \\ M_z + l_r F_{y_r} \end{bmatrix} \quad (28)$$

Where $\mathbf{B}_2^\#$ is the pseudo inverse of \mathbf{B}_2 . No optimization solver is implemented. The solution is calculated off-line and then programmed. This takes less time to compute which makes it suitable for real-time operations.

4. AMESIM[®] SIMULATIONS

AMESim[®] gives the opportunity to test control logics using more realistic models by means of a large library of vehicle dynamic components.

4.1 The Vehicle Model

The vehicle model consists on a 3D sprung mass with advanced options¹, a steering column model, an engine model, a braking system based on independent rotary Coulomb friction to enable implementing ESP control logic, suspensions, the non-linear coupled Dugoff's tire model, spindle kinematics, and rotational degree of freedom relative to the spindle of both front and rear wheels to enable implementing the ARS control logic. This results on a 16 Degrees-Of-Freedom (DOF) model. A road model is also given to enable modifying road conditions.

4.2 Simulation Results

Simulations were chosen to shed the lights on the safety that each architecture offers. Two scenarios are highlighted where both ARS and ESP can be activated at the same time: μ -split manoeuvre, and ARS failure.

4.2.1 μ -split manoeuvre

Here, the vehicle is supposed to be in a right cornering. At $t = 8s$ we make the friction coefficient drops from 1 to 0.1 at only the left tires. The yaw rate responses of both approaches are compared in Fig. 3.

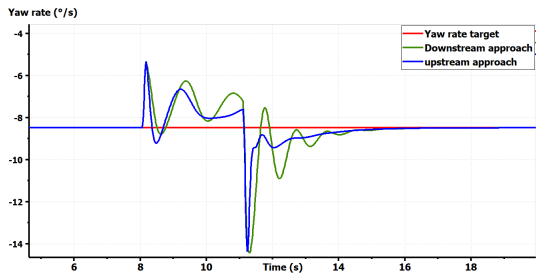


Fig. 3 Yaw rate control comparison - μ -split manoeuvre.

We can see that the upstream approach provides a more stable regulation with respects to the downstream approach. This latter shows an oscillatory behaviour which is unacceptable. Running through commands signals showed that the ESP system has been activated only in the upstream approach as the Fig. 4 illustrates, even though the ARS system is not saturated in both approaches. The reason is that the control allocation made in the upstream approach takes account not only of the actuators saturation, but also effectors saturation. In this case, it is the requested lateral rear tire force that exceeds the tire potential. Taking into account this phenomenon before distributing the commands, enables the yaw rate to stabilize rapidly. Omitting this physical constraints may destabilize the vehicle.

¹Enables anisotropic friction, roll height modification, aerodynamic disturbances and so on.

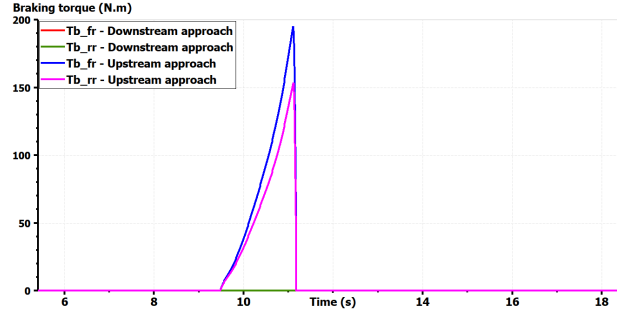


Fig. 4 Brake torques comparison - μ -split manoeuvre.

4.2.2 ARS failure

To simulate this scenario, we choose a repeating slalom manoeuvre, where the ARS fails at time $t = 4s$. Yaw rate responses are illustrated in Fig. 5.

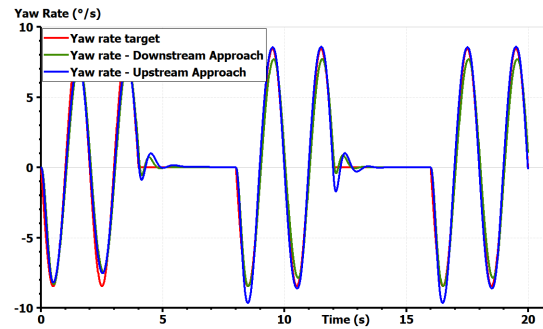


Fig. 5 Yaw rate control comparison - ARS failure.

Unlike the upstream approach, the yaw rate response in a downstream approach start losing its precision at the second set of the slalom manoeuvre. The brake torques allocation differs from an approach to another. The difference of brake torques between the right and left wheels is less important in the upstream approach. In fact, the vertical load distribution varies. This criterion is taken into account in the control allocation by means of the weighting matrix in (27). This weighting matrix reduce the weight when the vertical load increases, which favours the corresponding brake torque command.

5. ARCHITECTURES COMPARISON

The industrial practice that favours the least complex solution or cheaper ones results on a downstream approach when conflicts could be avoided only if interactions could be foreseen. The higher the number of integrated systems gets, the more fuzzy these interactions become. The coordination approach should be reinvented to enable developing safer autonomous vehicles.

5.1 Safety Architecture

Active chassis systems were invented to assist the driver in complex manoeuvres. The vehicle is becoming more and more automated, and it is expected that safety will increase with this practice. Chassis systems should be ready to operate safely at every possible situation. In a

downstream approach, rule-based algorithms are used to favour one system over another. These rules can be formulated when a small set of parameters are concerned. With the simple example that we have exposed, subsystems influence on the overall vehicle depends on longitudinal deceleration and actuators saturation. Other important parameters, as tire saturation, were ignored. The reason is that these parameters should be taken into account before the command generation. As only specific rules are proposed in a downstream configuration, we cannot prove if optimal solutions are provided. Therefore, the performance is not maximized. This is why this approach is used to handle single-objective problems while secondary objectives can be fulfilled [2]. Regarding the upstream approach, an optimization-based method is proposed. Performances can be maximized. This is mainly due to the fact that the interactions management has been brought up to a higher level. In this configuration, tire limits and other parameters could be taken into account before allocating the requests. This, however, can only be done if a more complete description of tasks is ensured.

5.2 Industrial Constraints

In this paper, we have demonstrated that the number of controllers can be reduced in an upstream approach. The command distribution is ensured directly after with control allocation methods. In the downstream approach, the controllers are placed in parallel. In the upstream approach, the number of CANs, cables, ECUs, and space are expected to be reduced [1]. The upstream approach could be then cheaper. Another characteristic that have kept industrials skeptical about optimization-based methods is their time of execution. Here, we insist on the fact that the optimization problem is solved off-line and only a pseudo-inverse of the effectiveness matrix is implemented. This results in faster computations. The high performance that these algorithms can offer enables higher number of interactions management that the downstream approach cannot handle. So, the more complicated the problem gets when approaching to fully autonomous driving, the more pertinent the upstream approach becomes. Therefore, to accelerate autonomous vehicles development, the different stakeholders should have a common framework. For this reason, in [2], several characteristics that the overall control architecture should respect have been determined: *Adaptability*, *Dynamic reconfiguration*, *Extensibility*, *Modularity*, and *Openness*. Unlike the downstream approach, the upstream one ensure most of these characteristics. Only *Openness* still need a close stakeholders' collaboration.

6. CONCLUSION

In this paper, two chassis systems coordination approaches have been compared. The ARS system and the braking-based VDC system were chosen as an example to study both architectures as they can both influence the

yaw rate. AMESim[®] software was used to provide more realistic simulations. The results showed that the upstream approach enables improvements in terms of safety and extensibility. These improvements were possible also thanks to a new linear tire with varying parameters.

This implies that the global chassis control strategy should change. Car manufacturers and suppliers should join forces to establish new standards and a common framework to accelerate autonomous vehicle development. The authors recognize that more evidence are required by means of real experiments. We expect more collaboration by car manufacturers.

Finally, this paper showed that when taking account of only tire forces couplings, safety is improved. Couplings at the vehicle level should be also taken into account, especially when vertical motions of the sprung mass can be controlled. This constitutes one of our future works which focus on the high-level control. More subsystems with different abilities can be then integrated to ensure a global vehicle motion control.

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