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Systematization of Integrated Motion Control of Ground Vehicles

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ABSTRACT This paper gives an extended analysis of automotive control systems as components of the integrated motion control (IMC). The cooperation of various chassis and powertrain systems is discussed from a viewpoint of improvement of vehicle performance in relation to longitudinal, lateral, and vertical motion dynamics. The classification of IMC systems is proposed. Particular attention is placed on the architecture and methods of subsystems integration.

INDEX TERMS Motion control, road vehicles, vehicle dynamics, vehicle safety.

I. INTRODUCTION

Development of novel mechatronic systems and advanced control engineering methods as well as a permanent demand on environment-acceptable and safe intelligent technologies has a profound impact on ground vehicle engineering in general. This impact results in both an increasing degree of automation of systems employed in ground vehicles and emerging new concepts like the integrated motion control (IMC). In this context, a coordinated operation of automotive chassis and powertrain systems undertake the function of controlling the vehicle motion to enhance safety, eco-friendliness, comfort and other vehicle characteristics. IMC application areas relate not only to the traditional transportation sector of passenger cars and commercial vehicles but also to agricultural, mining, construction and forestry machinery. The IMC systems are also substantial components of automated driving, which has dartingly increased importance for the future transportation paradigm.

The integration of active chassis and powertrain systems can be considered in relation to three domains of vehicle dynamics, Fig. 1. From this standpoint the IMC can specifically address longitudinal, lateral and vertical motion of the vehicle or can have cross-domain applications. By the cross-domain application is meant that integrated systems are targeting simultaneous improvement of the performance in terms of parameters related to vehicle dynamics in XY-, XZ-, YZ- or even XYZ-frame of reference.

Basic works related to the IMC have originated in the middle of 1980s. In particular, Fruechte and co-authors have introduced in [1] the results of General Motors project *Trilby*,

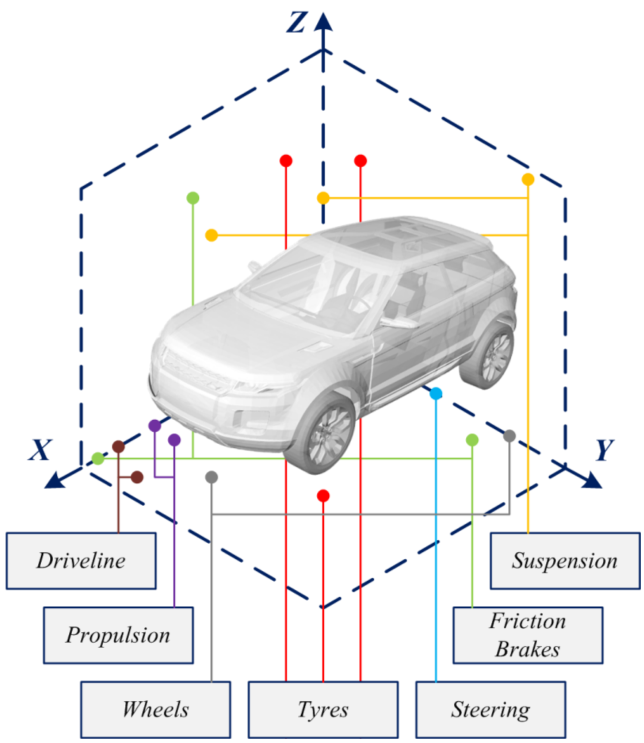

FIGURE 1. A frame of reference of ground vehicle dynamics.

where global integration architecture for various vehicle subsystems has been developed, Fig. 2. On this early stage of the development of IMC technologies another studies have been

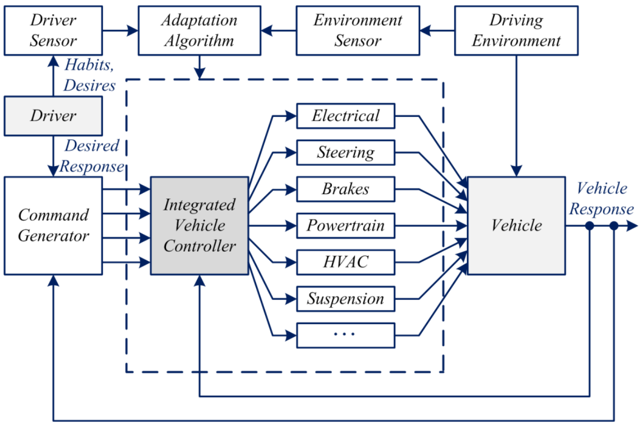

FIGURE 2. Integrated control with driver and environmental adaptation (adapted from [1]).

also presented by researchers from Hitachi [2] (coordination of engine, drivetrain, brakes, steering, and suspension operation), and Toyota and Aisin Seiki [3] (integration of active suspension and four-wheel steering).

Further progress in the integrated motion control led to numerous academic and industrial studies proposing different combinations of active chassis and powertrain systems. As a result, a demand arose for systematization and classification of available and potential integrated systems that could contribute to an efficient advance of the IMC. It is worth noting that several surveys in this area have been previously published. In particular, Trächtler and Niewels proposed in [4] a classification and corresponding analysis of the active system integration variants for the purposes of lateral vehicle dynamics control. Yu, Li, and Crolla have made in [5] an overview of different architectures of integrated vehicle dynamics control systems. However, an extended survey of various IMC configurations in relation to longitudinal, lateral and vertical dynamics of the vehicle is not found from the analysis of available research literature. Therefore the presented paper has an intention to close this gap and will discuss the following topics in next sections: IMC classification in relation to a frame of reference of vehicle dynamics; overview of IMC systems; overview of methods for integration of active chassis and powertrain systems.

II. INTEGRATED MOTION CONTROL IN CONTEXT OF GROUND VEHICLE DYNAMICS

A frame of reference of vehicle dynamics form Fig. 1 can be used as a basis for IMC classification. In that case the classification includes seven possible classes, which can be called hereinafter as *X-, Y-, Z-, XY-, YZ-, XZ-,* and *XYZ-integration* for convenience.

The *X-integration* means a combined operation of systems for the purposes of longitudinal vehicle dynamics control, e.g. braking or traction control. A case in point is the electric vehicle brake system integrating conventional friction brakes and electric motors operating in a regeneration mode.

Next example is a variant of the traction control uniting the motor management and active centre differential. For the X-integration, the most important common IMC performance criterion is the utilization of longitudinal friction of tyres μ_x. An ideal case in this regard is $F_x = \mu_x \cdot F_z$, where F_x and F_z are the longitudinal and vertical tyre forces. Among other criteria, minimization of tyre energy dissipation E_{tyre} and wheel slip power losses P_λ can be also included for the IMC performance estimation. These parameters can be calculated in a general case for a four-wheel vehicle as

$$E_{tyre} = \sum_{i=1}^{4} \int_{0}^{T} F_{x,i} V_{\lambda,i} dt, \tag{1}$$

$$P_\lambda = \sum_{i=1}^{4} \int_{0}^{T} P_{w,i} \left(1 - \lambda_i\right) dt, \tag{2}$$

where T is the vehicle manoeuvre duration, λ is the wheel slip, V_λ is the wheel slip velocity, and P_w is the power supplied to the wheel.

The *Y-integration* is attributed to the system interaction aiming at the lateral vehicle dynamics control. The active steering and brake-based or torque-based yaw moment control is a widespread combination in this regard. The corresponding IMC performance is traditionally assessed through parameters related to the vehicle side slip angle β and yaw rate $d\psi/dt$.

The *Z-integration* implies a joint operation of systems influencing the vertical vehicle dynamics. As an example, the coordination of the active suspension and the dynamic tyre pressure control can be considered in this context. Different indicators of ride comfort like the vertical acceleration, and road holding can characterize the performance of corresponding IMC systems.

The *XY-, XZ- or YZ-integration* supposes such a system combination that influences simultaneously two corresponding domains of vehicle dynamics. For instance, the brake control and active suspension can illustrate the XZ-integration; the active roll control and torque vectoring/electronic stability control belong to the YZ-integration. The IMC performance in such cases has to be estimated with integrated criteria uniting particular parameters of longitudinal, lateral or vertical vehicle dynamics. By way of example, Fig. 3 shows graphical interpretation of combined utilization of longitudinal and lateral tyre friction proposed by researchers of Toyota Motor Corporation in [6]. This approach can be used for the performance assessment of the IMC systems under the XY-integration class. For the XZ and YZ classes assessment of pitch and roll vehicle dynamics can be utilized correspondingly.

The most complex class, the *XYZ-integration*, includes the IMC systems configurations influencing all the domains of vehicle dynamics. Several concepts can be mentioned in this regard. For example, ZF Group has introduced in [7] the system network uniting the active steering, the torque vectoring,

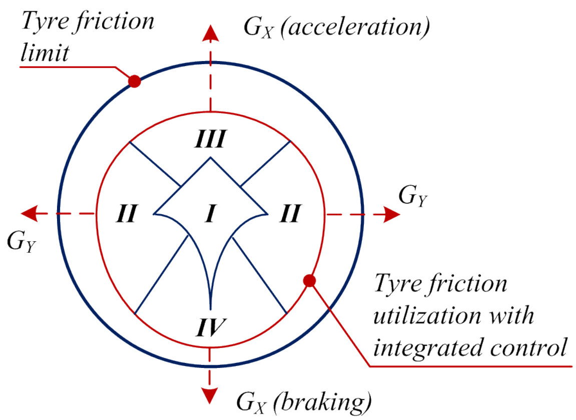

FIGURE 3. Influence of integrated chassis control systems on tyre friction utilization. Regions: I - without control; II - integration of active suspension and active front and rear steering; III - integration of traction control and active front and rear steering; IV - integration of anti-lock braking control and active front and rear steering (adapted from [6]).

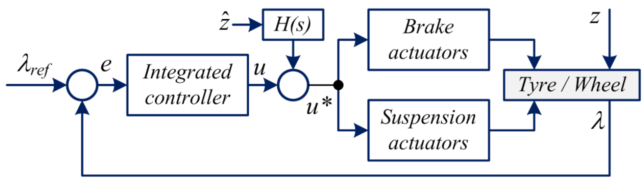

FIGURE 4. Structure of integrated brake and suspension control.

the brake-based stability control, the variable damping control, and the active roll stabilization system. Many concepts of single-wheel actuators or active wheel corners [8], [9] are also being classified under the XYZ-integration. These concepts are herein taken to mean that driveline, brake, suspension, steering and other relevant actuators (e.g. for the camber control) are integrated for each individual wheel unit (mainly in the wheel hub and adjacent packaging space). The IMC systems under the XYZ-integration class require very comprehensive criteria to assess their functional performance. These criteria must include parameters of longitudinal, lateral and vertical vehicle dynamics and are most often composed as complex cost functions that will be further illustrated.

Next part of the study will discuss each IMC class in more details in respect to the most relevant variants of the system integration and their control architecture and performance.

III. X-INTEGRATION CLASS

The X-integration of automotive chassis and powertrain systems contributes to the improvement of traction and/or braking performance of the vehicle. The IMC systems targeting the traction performance only are rarely presented in research studies. An example is the combination of the traction control and the active interaxle differential for the all-wheel drive vehicle. The system integration for the purpose of the braking performance enhancement is being more investigated. Two particular and most encountered integration variants will be further discussed.

A. INTEGRATION OF THE BRAKE AND SUSPENSION CONTROL

The integration of the brake and suspension control from viewpoint of longitudinal vehicle dynamics enhances the braking performance, especially under conditions of a rough, uneven surface. Road unevenness has an adverse influence on the tyre-surface contact: the contact is becoming a high grade of inhomogeneity that reduces the utilization of tyre friction. Moreover, road disturbances can affect the signal quality of the wheel speed sensors required for the anti-lock braking system (ABS). As a consequence, most of known ABS algorithms are being deactivated in off-road conditions. These shortcomings can be avoided with the combined operation of the brakes (ABS control) and suspension control. General IMC scheme under discussion is depicted on Fig. 4. The global control task in this case is to minimize the control error e between the actual λ and reference λ_{ref} wheel slip at braking. As a rule, the reference wheel slip lies in the range corresponding to the maximum tyre-road friction for actual surface conditions. The initial control demand u produced by the integrated controller is then splitting up between the brake and suspension actuators. The brake actuators produce the brake torque to be applied to the wheel. The suspension actuators change the lifting and pitching forces on the wheel to compensate road disturbances z. It is reasonable for the presented controller architecture that the road disturbances are also estimated and, after applying a certain transfer function $H(s)$, taken into account for the corrected control demand u^*.

The effect on the braking performance from the discussed IMC has been demonstrated in research literature for different suspension systems. In particular, one of the first relevant studies [10] has investigated the ABS and the suspension equipped with electro-hydraulic actuators. The simulation of the car braking from 16 m/s on the wet asphalt demonstrated for the quarter vehicle model that the stopping distance is reduced by 7-8% with the integrated control as compared with the conventional ABS operation. The integration algorithm in this case handles the dependence of the suspension force from the brake torque:

$$F_z = A \cdot sign\left(T_{br} - T_{br\_mean}\right), \tag{3}$$

where F_z is the desired force profile of the suspension actuator, A is the amplitude of F_z, T_{br} is the actual brake torque supplied to the wheel, and T_{br} is the mean brake torque on the wheel during braking. A similar integration approach has been also investigated in [11], where the controller is realized using the nonlinear backstepping procedure to the error state variable

$$\begin{aligned} e_z &= F_z - F_{z\_dem} \\ F_{z\_dem} &= m \cdot g + K_{AS} \cdot T_{br}, \end{aligned} \tag{4}$$

where F_{z_dem} is the demanded normal force, m is the vehicle mass (reduced to the wheel in the case the quarter-vehicle model), K_{AS} is the operational coefficient of the active suspension actuator. The simulation of passenger car braking on the dry asphalt demonstrated the stopping distance reduction on ~7,4% for the integrated control as compared with the conventional ABS control.

As differentiated from previous simulation-based cases, the experimental variant of the integration is discussed in [12], where the ABS and the semi-active continuous damping control are considered with the possibility to switch the suspension to a higher damping mode during the ABS operation. This approach reduces the wheel slip oscillations $\lambda(t)$ around the reference value λ_{ref} that increases the utilization of the longitudinal tyre friction and can be realized through the controlled damping of F_z-fluctuations in the course of the integrated brake and suspension control. This effect can be seen by analyzing the following quarter-vehicle formula of the difference $\Delta\omega$ between the actual wheel angular velocity and theoretical velocity of the free rolling wheel

$$\Delta\omega = \frac{\int \mu_x(\lambda(t)) \cdot r \cdot F_z(t)\, dt - \int T_{br}(t)\, dt}{I_w}, \qquad (5)$$

where r is the wheel radius, I_w is the moment of wheel inertia. For the method under discussion, the work [12] implemented the MinMax control strategy for the damping mode switching and tested the integrated system on the real car in dry and wet highway conditions. Based on the test results the mean braking distance has been reduced on 1…3,5% (depending on the road roughness and reference mode of the suspension) as compared to the non-integrated system with the fixed damping.

In spite of certain improvements in the braking performance demonstrated in previous paragraphs, the integration of the brake and suspension control is more reasonable for the vehicles operating in off-road conditions. This can be confirmed with results presented in [13]. This study investigated the braking of a sport utility vehicle (SUV) on two types of rough terrain. The integration has concerned ABS and a semi-active 4S$_4$ suspension unit. Considering braking from 80 km/h to 10 km/h, the optimal integration variants allowed to reduce the stopping distance on 1,9 m for the road with the parallel corrugations and on 9,0 m for the Belgian pavement.

Finally, some remarks can be inferred for the integrated brake and suspension:

- Published studies confirm the effect in the braking performance from the integrated control, however the most of known integration approaches have been validated in simulation only;
- There are no priority variants of the suspension control for the integration with ABS - both semi-active and active units as well as the suspension stiffness and damping control are finding implementation in various research works;
- More benefits from the integration can be expected for braking maneuvers on pavements and uneven surfaces, rather than on flat road.

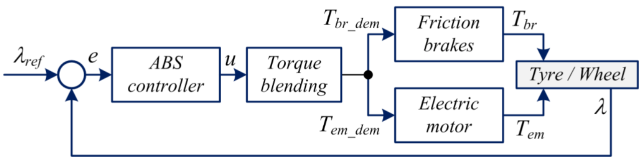

FIGURE 5. Structure of integrated ABS control of electric vehicle.

B. INTEGRATION OF THE REGENERATIVE AND FRICTION BRAKE CONTROL

Hybrid and full electric vehicles have possibility for a combined operation of a conventional brake system actuating the friction brakes and electric motors in a regeneration mode. Hence the corresponding ABS of the electric vehicle can be considered as an integrated system because the control functions are being realized with two independent actuators (friction brakes and electric motors). A simplified ABS configuration for the discussed case is depicted on Fig. 5. Here the ABS controller processes the error e between the reference λ_{ref} and actual λ wheel slip and produces the control demand u. The torque blending defines the shares of the total brake torque demand, which have to be realized by the friction brake system and the electric motors, T_{br_dem} and T_{em_dem} correspondingly. Then the actual brake torques T_{br} and T_{em} are applied to the wheel. First of all, this kind of integration improves the braking performance due to higher response speed of electric motors as compared with friction brakes. Secondary benefits like a more smooth operation of the electric ABS (it provides a better comfort at braking due to the reduced oscillations in deceleration) and the braking energy regeneration are also of importance. However the conventional ABS with friction brakes can not be fully replaced with the electric ABS due to safety requirements and limited braking torque of electric motors, therefore the integration of both systems is called for.

Most of relevant solutions, which are known from research publications, are proposing a rule-based approach to the system integration. The required set of rules is usually aimed at preferential operation of the electric motors as the ABS actuators to increase amount of energy regenerated at the braking manoeuvre. The main limitation to be considered in this case is the state-of-the-charge (SOC) of the battery. Other specific limitations concern the type of electric powertrain (central motor, individual axle motors, individual wheel motors) and the speed and thermal modes of the motors.

A variant of the rule-based integration method is proposed in [14] and [15] for the electric vehicle with the front and rear axle motors: (i) Friction brakes and electric motors have an integrated operation when the front or rear slip ratio is below *0.1*; (ii) Within the slip range between *0.1* and *0.2*, only the electric ABS is working; (iii) When the front or rear slip ratio exceeds *0.2*, the brake torque on corresponding wheels is compulsory set to zero to avoid the wheel lock; (iv) When the battery SOC reaches the overcharged level, the brake torque is supplied from the friction brakes only.

The tests of this approach both on the vehicle software simulator and the real vehicle demonstrator have functionally validated this approach, especially on the low-friction and split-μ surfaces. Another, more extended base of rules is proposed in [16], where both the wheel slip λ and the specific vehicle deceleration $z = a_x/g$ are used for the integration thresholds. In particular, four z-levels determine the transition between the electric and hydraulic ABS mode as well as the required brake torque rate. Pure electric ABS actuation at low torque rate is used for low friction road surfaces ($z<0.3$). At high deceleration level ($z>0.7$), purely hydraulic ABS actuation is applied. Both electric and hydraulic actuators are simultaneously operated when braking dynamics is situated at intermediate deceleration levels. The simulation results, presented in [16], indicate that this rule-based algorithm can keep the wheel slip in the range $0.2 \ldots 0.3$ on the low-friction road ($\mu = 0.4$). Other relevant variants of rule-based integration of the friction brake system and regenerative brakes are also introduced in [17] and [18].

Despite good applicability of rule-based approach to the integration of different brake actuators from viewpoint of robustness and low demand to the processing resources of the controller, another (non-linear and continuous) integration methods are also of particular interest in respect to a better performance utilization of electric motors. An example of corresponding system architecture with feedforward and feedback compensation elements has been considered in [19] and [20]. Here the integration procedure is defined by a number of factors as SOC, the electric motor temperature and speed etc. The validation of the controller has been discussed in [20] both for simulation of the braking on a road surface with $\mu = 0.5$ and for real car tests on a low friction road. These tests confirmed that the system can ensure required braking performance and, in addition, can contribute to a slight increase (0.5 %) of the battery SOC caused by brake energy recuperation.

Other non-rule-based approaches are also known from published studies. For instance, methods based on the iterative learning control [21], the combined open-loop/closed loop control [22], and the sliding mode control [23] can be mentioned in this context. The continuous integration is of particular interest for all-wheel drive electric vehicles with individually controlled electric motors because of more flexible system architecture. The variant of the corresponding integrated ABS control is investigated in [24] and [25] for SUV equipped with four switched-reluctance on-board electric motors. In accordance with the IMC architecture, Fig. 6, the system operates with the feedforward, predictive T_{pred} and feedback, reactive T_{react} part of the total brake torque demand, which is then split up between the friction and electric brake systems, T_{br_dem} and T_{em_dem} correspondingly. The predictive part of the total brake demand is formulated from the brake pedal travel and actual road friction conditions. The reactive part is produced by the reactive controller (proportional-integral controller with gain scheduling by the vehicle velocity V_x) from the error between the reference λ_{ref} and actual λ wheel slip. A specific feature of the proposed integration principle is that the electric powertrain has to modulate the high-frequency torque variations whilst the rest brake torque demand is split between friction brakes and regeneration, taking into account motor and battery limitations but giving priority to regenerative brakes. The discussed integrated control demonstrated a better performance resulting in the reduction of the stopping distance in low-friction and transient surface conditions on more than 20% as compared with the benchmarked conventional ABS.

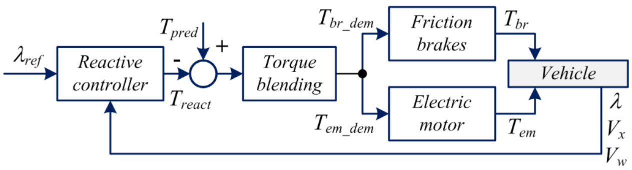

FIGURE 6. Integrated ABS control with continuous direct slip control.

Summarizing, the following observations can be done for the integrated regenerative and friction brake control:

- Majority of studies confirms better braking performance for the integrated ABS as in comparison with conventional ABS actuating the friction brakes;
- For the hybrid electric vehicles and full electric vehicles with the central electric motor, most of analyzed integration principles are based first of all on the parameters of electric powertrain as SOC, motor speed and temperature and are mainly using the rule-based control methods.
- For the full electric vehicles with individual in-wheel and on-board motors, the non-linear, continuous integrated controllers can have certain advantages due to better utilization of high response speed of the regenerative braking system.

IV. Y-INTEGRATION CLASS

The aim of the Y-integration is first of all to improve the vehicle stability and handling. Several most relevant IMC variants can be mentioned. The first largest group covers the integration of active steering and brake-based/torque-based yaw dynamics control. The second group includes active suspension (AS) elements in addition to the active steering and yaw dynamics control. Both groups are being analyzed next. The Y-integration class supposes the systems influencing only yaw dynamics in relation to the vehicle stability. Additional consideration of roll dynamics is subjected to the Y+Z-integration class discussed in Section VIII.

A. INTEGRATION OF THE ACTIVE STEERING AND BRAKE-BASED/TORQUE-BASED YAW DYNAMICS CONTROL

The active influence on yaw vehicle dynamics can be basically realized through the control on wheel torques and steering wheel angles. Hence possible integration variants can involve combinations of the brake-based yaw control

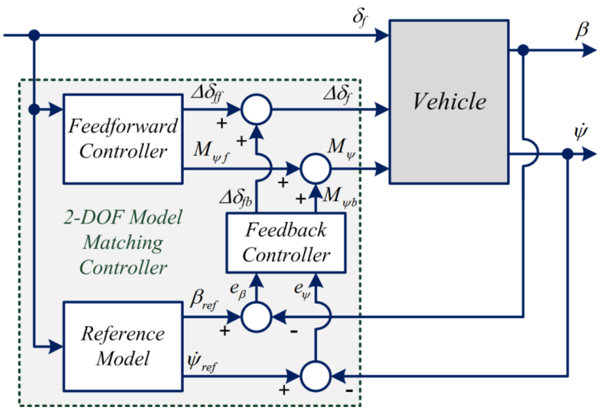

FIGURE 7. Integrated active front steering and direct yaw control system with model-matching control (adapted from [27]).

(electronic stability control ESC, yaw moment control YMC), the torque-based yaw control (torque vectoring TV), the active front and rear steering (AFS, ARS), and the active all-wheel steering (AWS). However, these systems each have certain limitations [26]. Many studies are investigated the integration of the yaw dynamics control and the active steering on the front wheels only. An example of such a combination has been given in [27] for the controller architecture depicted on Fig. 7. Here a model-matching controller based on the 2-DOF model of the lateral vehicle dynamics is used. It includes feedforward and feedback parts for the generation of predictive and reactive components of additional steering angle $\Delta\delta_f$ from the AFS and correcting yaw moment M_ψ from the ESC. The feedback is built up using the optimal control method. The simulation results presented in [27] did not display essential effect from integration during the control of the lane change or J-turn manoeuvres. However, the considerable improvement of the vehicle dynamics has been obtained for the situation with the side wind disturbance.

Another variant of the hierarchical AFS/ESC integration is proposed in [28]. As differentiated from the previous example, the system has the upper level controller, which is based on the sliding mode control and defines the reference yaw rate, and the lower level controller that is responsible for the system integration. The integration principle is based on an optimal coordination of the active lateral and longitudinal tyre forces for the reference yaw moment, and the optimization procedure is determined by the Karush-Kuhn-Tucker conditions. The simulation results for this approach given for a single lane change maneuver on the dry road from 90 km/h have demonstrated that a certain effect can be observed for a better tracking of the reference yaw rate only as compared with the ESC operation. The reduction of side slip angle was comparable for both systems (ESC and the AFS/ESC integration). Further advancement of this method is discussed in [29], where the coordination of AFS/ESC is subjected to the optimization procedure applied to six active tyre forces: lateral forces for four tyres each and longitudinal forces for two front tyres). The shares of the AFS/ESC systems in the formulation of the total control demand are computed depending on the cost function, an equality constraints for the reference yaw moment and inequality constraints for the performance limits of tyres and the brake-based ESC actuators. This method has been studied through simulation of cornering and double lane change maneuvers and demonstrated a possibility for simultaneous improvement of lateral stability, manoeuvrability and agility.

Among other well-accepted approaches applied to the integration of the active steering and yaw moment control, the methods that are based on the fuzzy control [30]–[32] and the model-based/model-following control [33], [34] can be mentioned. The works related to the latter approach have pointed out its good performance for the control tasks, where not only the lateral stability but also the path tracking is of particular importance.

Recent studies give much attention for the integration not only the AFS but also the active rear steering and the all-wheel steering control. A comparative investigation in this regard has been presented in [34] and [35] for AFS/AMC, ARS/YMC and AWS/YMC configurations based on model-predictive and sliding mode controllers. It was demonstrated in particular by the simulation of lane change maneuvers that (i) the integration with AFS gives benefits in minimization of the side slip angle and more precise vehicle path following, (ii) the ARS/YMC is efficient for the situations with deterioration in the rear wheel cornering stiffness, and (iii) the AWS/YMC integration can be characterized by high stability limits with high responsiveness.

The integration of the active steering can be realized not only with the brake-based YMC but also with the torque-based control involving, for instance, the torque vectoring [36] or the active differential (AD) [37]. It should be mentioned that the integration of the torque-based yaw moment control with the active steering is specifically promising in context of electric vehicles of various architecture. In particular, the study [32] has introduced a hybrid vehicle with the integrated fuzzy control of the AFS and the axial motors with the inter-wheel differential generating individual wheel torques. Such control architecture has demonstrated a potential to reduction of side slip and yaw rate both of oversteer and understeer vehicle in the simulation of J-turn and single-lane change maneuvers. Another architecture has been investigated in [38] and [39], where a full electric vehicle is equipped with in-wheel motors. The integration has been proposed on the basis of a decoupled, observer-based side slip and yaw rate control and covered the joint operation of the electric motor torque control with the AFS and the AWS. Both variants have been tested on the experimental vehicle and demonstrated, in particular, the robustness of the integrated control to cornering stiffness variation.

Previous paragraphs are illustrated the integration of the active steering with yaw dynamics control by only few particular examples. Nevertheless, a number of published studies in this topic is high that allows to deduce some common observations from their analysis:

- Many studies consider the active steering on only front wheels for the integration. However, such integrated systems are of interest rather for the research than for practical implementation because of minimal effect in stability improvement as compared with stand-alone yaw motion control systems;
- The integration of the YMC with the AWS has more potential for the enhancement both of stability and handling, especially for vehicles equipped with the steer-by-wire and for HEV/FEV. But the complexity of the control is growing in this case due to high nonlinearity of the cornering dynamics of the vehicle with all steered axles;
- There are no considerable differences in the integration architecture, when one compares the brake-based and torque-based yaw dynamics control. But it should be mentioned that the published investigations in the integration of the active steering with torque-based YMC are more concerning electric vehicles with individual wheel motors.

B. INTEGRATION OF THE YAW DYNAMICS CONTROL AND ACTIVE SUSPENSION ELEMENTS

The inclusion of the active suspension elements into the IMC supposes first of all the contribution to the roll stability improvement, and the resulting system configuration should belong in particular to the Y+Z-integration class. However, some studies consider the AS as a tool for supporting the yaw dynamics control only. Such IMC systems can be attributed to the Y-integration class. Several corresponding variants are discussed below.

The work [40] proposes a set of basic requirements of particular relevance to integrated yaw dynamics control systems uniting the active steering and the active suspension control: (i) improvement of steering responses/yaw rate tracking in normal driving conditions; (ii) minimization of the influence of the integrated controller on the longitudinal vehicle dynamics; (iii) avoidance of the control discontinuity with a change in the control priority. These targets were realized in the integrated system, Fig. 8, where the yaw rate tracking error e_ψ is the input point for the integration. Here the AFS is active for the entire cornering manoeuvre, and the AS is active for the certain time T. The required change in the vertical forces F_{zc} during the activation time is defined by look-up tables (LUT). The proposed integrated system has been realized using the fuzzy control and simulated for the single- and double-lane change manoeuvres on a dry road at different velocities. The simulation results confirmed the feasibility of this approach. However, the obtained improvements in yaw rate tracking and lateral stability were of a minor nature as compared with the stand-alone AFS control.

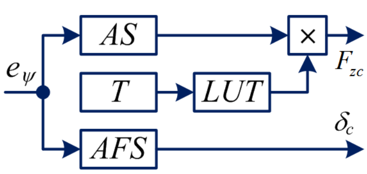

FIGURE 8. Integrated active front steering and active suspension (adapted from [40]).

Nevertheless, the active suspension can bring more collateral advantages that are not directly related to the vehicle stability. In particular, the study [41] has investigated an IMC covering the brake-based yaw moment control, the active steering and the active suspension represented by the active roll bars and controlled magnetorheological dampers. The IMC system is realized on the model-based approach with the control authority defined by the yaw moment difference

$$\Delta M_z = M_{z2} - M_{z1}, \tag{6}$$

where M_{z1} is the vehicle yaw moment calculated from the tyre forces defined through the equations of the vehicle motion for the condition of the vehicle steady state response to a constant steering angle, M_{z2} is the vehicle yaw moment calculated with consideration of the change in the tyre normal and lateral forces caused by the operation of active subsystems. The starting point and time of the activation of the active subsystems (brakes, suspension, steering) is defined by a number of criteria like quickness of the subsystem response, the level of obtrusiveness to the driver, and the required M_z-magnitude to be corrected. The functional tests of the discussed IMC for a midsize rear wheel drive car have demonstrated that the inclusion active suspension elements allows to reduce the time of brake system control activation by 23-25% for the sinusoidal steering and J-turn manoeuvres, by 58% on circular track, by 69% in double lane change, and by 72% on handling track [41]. These results can be considered as beneficial in terms of (i) lesser losses of the vehicle velocity during the manoeuvre and (ii) a certain increase of the energy efficiency due to lower demand in the operation of the brake actuators.

The active suspension elements involved in integrated control systems can have various hardware implementations. Along with magnetorheological dampers and active roll bars mentioned in [41], other variants like the active camber control [42], [43] are also investigated. However, it should be mentioned that the value engineering of the active suspension as an IMC component is rarely discussed in relevant research literature. Another general observation regarding the role of active suspension in the integrated control is that this topic every so often is being investigated for a simultaneous impact of the AS operation both on the lateral dynamics and the roll stability. This will be illustrated in Section VIII.

V. Z-INTEGRATION CLASS

Combined operation of active chassis systems under the Z-integration class should target the improvement of such factors as ride comfort, roll stability, pitch dynamics and road holding. The analysis of published studies has detected too little information about pure Z-integration. Nevertheless, this gap can be potentially covered, for instance, by the integration of active tire pressure and active suspension systems. Nowadays most of appropriate IMC systems consider the factors of longitudinal and lateral dynamics in addition to criteria of vertical dynamics and, therefore, are subjected to the X+Z- and Y+Z-integration classes discussed later.

VI. X+Y-INTEGRATION CLASS

Along with the vehicle performance enhancement by common criteria of the longitudinal and lateral dynamics, the IMC systems under the X+Y integration class have following specific functions: (i) Supporting the longitudinal manoeuvres, where the vehicle stability can be deteriorated, e.g. braking on a split-μ surface; (ii) Improvement of the lateral vehicle dynamics control performance in terms of additional criteria as minimization of vehicle velocity losses or the tyre and vehicle energy dissipation. Following paragraphs discuss principles of the corresponding integrated motion control.

A. INTEGRATION OF ACTIVE STEERING FOR BRAKING AND YAW DYNAMICS CONTROL

The integration of the braking and yaw dynamics control system with the active steering can be considered for a better utilization of tyre friction in general that also improves both the lateral and longitudinal vehicle dynamics. One of obvious relevant examples is the integrated control for the split-μ braking. A corresponding option has been demonstrated in [44], where the IMC unites the active steering based on the H∞ controller and the rule-based ABS. In accordance with the integration scheme, Fig. 9, the event detector identifies the split-μ braking situation. When it takes place, the brake trigger disables the brake pressure attenuation, and the steering controller is activated in order to correct the yaw motion of the vehicle. The yaw dynamics control in [44] has the disturbances compensation caused by asymmetric brake pressure distribution. This method has been investigated both on the HIL test rig and the test vehicle and demonstrated a permanent reduction of the stopping distance at the split-μ braking within the initial velocity range 60…100 km/h.

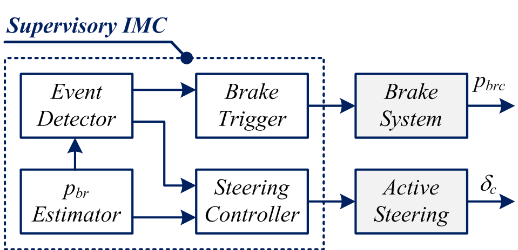

FIGURE 9. Integrated active steering and brake control (adapted from [44]).

Another variant of the integrated steering and braking control is presented in [45], where the fuzzy rules have been used for the coordination of both subsystems. This IMC has a target to compensate the control errors both for the reference wheel slip and yaw rate. To take into account an influence of the lateral dynamics on the braking process, the reference wheel slip is calculated as

$$\lambda_{ref} = k \frac{a_y \cdot g}{a_x \cdot V_x^2}, \tag{7}$$

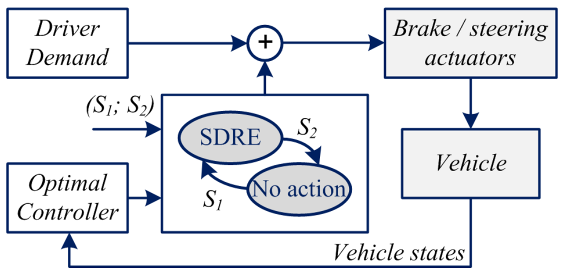

FIGURE 10. Optimal integrated active steering and brake control (adapted from [46]).

where k is the specific correction coefficient, V_x is the vehicle velocity, a_x and a_y are the longitudinal and lateral acceleration correspondingly. The HIL-test on the vehicle for the steady circle test with emergency braking has indicated that the proposed approach can improve the stability but at the cost of a minor deterioration of the brake efficiency [45]. Such a conflict between vehicle performance factors can be overcome by proper optimization of the integration strategy that has been illustrated for the IMC presented in [46]. The system under discussion, Fig. 10, uses the nonlinear optimal controller based on the state-dependent Riccati equation (SDRE) and performs the correction of the vehicle dynamics in accordance with the states S_i corresponding to certain expert driver characteristics. The optimization target of the integrated control is to minimize the performance index calculated for the following cost function:

$$J = \int_{t_0}^{\infty} \left[\begin{array}{c} \left(V_x - V_{xref}\right)^2 + \delta_c^2 + w_V \cdot V^2 + w_\psi \cdot \dot{\psi}^2 \\ + \sum_{i=1}^{4} w_{\lambda i} \left(\lambda_i - \lambda_{ref}\right)^2 + \sum_{i=1}^{4} w_{Ti} \cdot T_{bri}^2 \end{array} \right] dt, \tag{8}$$

where w_j are the weighting coefficients, λ is the wheel slip, T_{br} is the braking torque, δ_c is the steering angle assigned by the controller, index "*ref*" is for the reference values. The proposed approach has been investigated by the simulation of the split-μ braking with the friction coefficients 0.8/0.4 and actuator failure scenario. The test results have confirmed the robustness of the developed optimal controller and sufficient performance in terms of yaw rate reduction, optimal brake force distribution and disturbances compensation by tracking of the target longitudinal velocity.

It should be mentioned that the improvement of the split-μ braking performance can be also achieved through integration of the active steering and torque-based yaw dynamics control. A corresponding example can be found in the work [47], where a combination of the AFS and the semi-active rear differential is proposed.

An important criterion indicating the performance of the IMC systems under the X+Y integration class is the

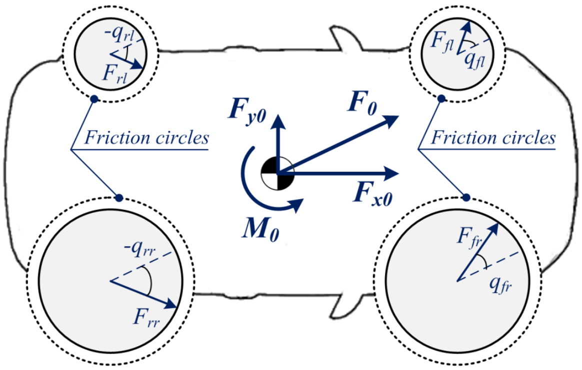

FIGURE 11. Definition of target reference force and moment for IMC in accordance with tyre friction circles.

utilization of tyre friction forces both in the longitudinal and lateral domains. In this regard several studies are demonstrated that it is reasonable to integrate the active steering and the yaw dynamics control with the consideration of tyre force allocation and constraints provided by tyre friction circle/ellipse. This statement can be illustrated with Fig. 11 that explains a concept introduced by Toyota's researchers in [48]–[50]. In accordance with this concept, the IMC systems uniting the individual active steering, traction and braking control for each wheel, has to minimize the friction rate γ for each wheel that is calculated as

$$\gamma_i = F_i \big/ F_{\mu i} = \sqrt{F_{xi}^2 + F_{xi}^2} \Big/ F_{\mu i}, \tag{9}$$

where index "i" is for the front/rear left/right tyre position (FL, FR, RL, RR), F_{xi} and F_{yi} are the longitudinal and lateral tyre forces correspondingly, F_i is the resulting tyre force, $F_{\mu i}$ is the tyre friction force (defining the friction circle). Additional target is that the friction rate γ has to be the same for all tyres. The reference vehicle satisfying all mentioned conditions can be represented with certain target force F_0 and moment M_0. The angles q_i between the vectors F_i and F_0 are being used as the control inputs for the required task of the minimization of the γ-value. This task can be also introduced as the maximization of the following cost function:

$$\begin{aligned} J &= \frac{(y_0 F_{x0})^2 + (x_0 F_{y0})^2 + M_{z0}}{\gamma} \\ &= y_0^2 F_{x0} \sum_i F_i \cos q_i + x_0^2 F_{y0} \sum_i F_i \sin q_i \\ &\quad + M_{z0} \sum_i F_i (x_i \sin q_i - y_i \cos q_i), \end{aligned} \tag{10}$$

where x and y define the centres of tyre forces in relation to the centre of vehicle mass in longitudinal and lateral directions correspondingly. Using the optimal control for Eq. (9), the required control demands for the steering and traction/braking actuators are being defined on the next step. The simulation results presented in [49] and [50] have shown that the proposed approach both improves the tyre fiction utilization and demonstrates better stability at different manoeuvres like split-μ braking, obstacle avoidance et al. as compared with non-integrated control. In particular, for single-shot sine-wave steering at 22 m/s of the vehicle velocity, the maximum resultant force was increased up to 10,5% with this method in comparison to conventional yaw dynamics control.

Some variations of the IMC with an optimal tyre force distribution are presented in other studies [51]–[53]. For instance, the work [51] has investigated the method where the generalized control forces are defined using the sliding mode control, and their individual optimal distribution to the tyres is realized with the servo-loop part implementing the control allocation based on a constrained quadratic programming. It was illustrated with the simulation results that the corresponding integrated AWS and YMC can also sustain the vehicle stability under critical disturbances like an actuator failure and crosswind effect. Another comprehensive IMC with the optimal tyre force distribution and servo-loop has been discussed in [52], where the system performance is evaluated with the normalize tyre force reserve NTFR (can be introduced as $1 - \gamma$ in relation to Eq. (8)). The simulation of different steady-state and transient manoeuvres has demonstrated that optimal coordination of the AWS and YMC reduces the NTFR value in range 30…80% as compared with conventional systems.

Examination of studies investigating the active steering, braking and yaw dynamics control under the X+Y integration class allows to note:

- An observable effect from the integration can be expected with the use of an all-wheel steering; In many situations an integrated use of the AFS or ARS solely is not reasonable when one compares possible level of vehicle performance improvement and required complexity of the controller;
- Analysis of proposed control tools for the system integration points to dominating application of various control allocation approaches relative to other methods; however, the validation of such integration technique on real hardware objects (vehicles) is still rarely addressed in the publications.

B. IMC SYSTEMS OF X+Y INTEGRATION CLASS FOR OVER-ACTUATED VEHICLES

A ground vehicle as a complex system can be considered as over-actuated one when it has more actuators than degrees-of-freedom. One of the first variants of such integration was proposed by Toyota [6], where the IMC united the active all-wheel steering, the hydropneumatic active suspension as well as the ABS and the traction control system. The vehicle tests on emergency obstacle avoidance, lane change with accelerating on a slippery surface and braking on a split-μ surface have confirmed high performance for this IMC configuration. Particular benefit has been achieved here for the vehicle controllability in the non-linear range of cornering forces and for overall system redundancy.

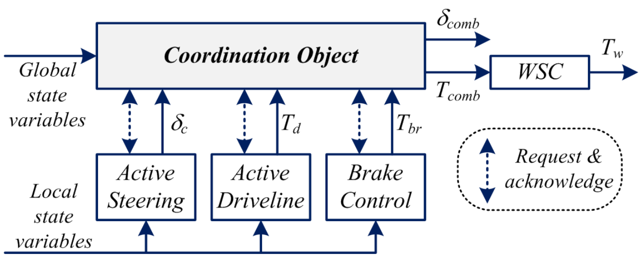

FIGURE 12. Agent-based IMC system (adapted from [54]).

Of special interest for the IMC under discussion is arrangement of priorities between targeted performance criteria depending on the actual manoeuvre. A variant of corresponding system coordination is introduced in [54], where an agent-based approach is applied to the integration of the AFS and ARS, the wheel slip control (WSC), the brake control, and the active driveline. In accordance with Fig. 12, the integration procedure is realized by the coordination object, which defines the control priorities for the individual systems (controller agents) and computes the combined control outputs for the steering angle δ_{comb} and wheel torque T_{comb}. The coordination from [54] uses the fuzzy logic and the weighting factors to formulate the outputs in accordance with four criteria: reference yaw rate, reference lateral acceleration, maximum allowable side slip angle, and desired longitudinal acceleration. The key part by the conflict resolution is assigned to the weighting factors. For example, a potential conflict can take place when the brake-based control has to track simultaneously the reference yaw rate and demanded longitudinal acceleration. In such a situation the proposed IMC will minimize the following performance function:

$$P = \sqrt{\begin{array}{l} w_\psi\left(\left(\dot{\psi}_{ref}-\dot{\psi}\right)/E_1\right)^2 \\ \quad + w_a\left(\left(a_x - a_{xref}\right)/E_2\right)^2 + w_f\left(f_{inst}/E_3\right)^2, \end{array}} \tag{11}$$

where w_i are the weighting factors, E_i are upper bounds of the tracking errors, f_{inst} is the instability factor taking into account the actual side-slip angle and lateral acceleration. The condition like introduced in Eq. (10) is then used for composing the coordination rules. The simulation of the cornering (with and without acceleration and deceleration) on different surfaces has indicated that this approach provides a stable vehicle motion with sufficient handling performance and reduction of involvement of brake actuators [54].

Analysis of other relevant studies in the IMC for over-actuated vehicles under the X+Y integration class points to the commonly encountered use of torque-based control systems, when the performance criteria of longitudinal dynamics are of importance also at steady-state and transient cornering manoeuvres. For instance, a corresponding IMC variant has been investigated by Magna Steyr as the concept of Global Chassis Control [55], [56], where the integrated controller was advanced with the torque vectoring. The road tests of the car equipped with the Global Chassis Control have demonstrated the increase of average velocity during the lane change and slalom manoeuvres on 4.8…6.6% [56]. Another IMC example, proposed by Mitsubishi Motors and covering the brake-based and torque-based YMC, the ABS and the active centre differential, has allowed to reduce the lap time on 1,5 sec for the vehicle tested on the 2,4-km handling circuit [57].

The implementation of torque-based systems for the IMC under the X+Y integration class is also reasonable in the case of over-actuated electric vehicles, especially in the case of AWD powertrain with individually controlled electric motors. In particular, the work [58] describes the integration of the brake-based direct yaw moment control, the active front steering and the torque-based yaw control through in-wheel electric motors on the basis of a non-linear method. This method combines steady-state-like control, feedforward control considering reference vehicle dynamics parameters, and state-dependent error feedback control. Another combination of the feed-forward and feed-back control with the yaw moment observer has been introduced in [59], where test results are presented for small electric car with electric motors and the AFS/ARS. As reported in [59], this control method provides equalization of workload for each wheel and quick response in terms of yaw rate correction. Promising results are also obtained for the relevant IMC that includes the genetic fuzzy active steering control and the torque vectoring control, which functions are being performed by in-wheel motors [60]. In the proposed method, the integration should follow the Gaussian activation function calculated as

$$\chi_{ATVC}\left(\delta_{corr}\right) = \begin{cases} e^{\left[-\frac{\left(\delta_{corr}-\delta_{corr\_max}\right)^2}{2\sigma^2}\right]} \cdot 100\%, & \left|\delta_{corr}\right| \le 3^\circ \\ 100\%, & \left|\delta_{corr}\right| > 3^\circ, \end{cases} \tag{12}$$

where χ_{AVTC} is the percentage of torque vectoring actuation, δ_{corr} is the steering angle required to correct the yaw dynamics in given driving conditions, δ_{corr_max} is the actuator range limit of the active steering controller. It can be seen that the condition (11) ensures gradual inclusion of TV into the control process. This solution avoids the influence of TV on driving comfort (provided by the active steering) as long as possible from viewpoint of vehicle stability. The validation of the proposed approach on various vehicle manoeuvres (double lane change, step-steer response, braking in a turn and on split-μ surfaces) has confirmed the developed IMC system not only guarantees required stability performance but also provides additional benefits for the longitudinal dynamics, e.g. less losses in vehicle velocity in cornering manoeuvres or noticeable reduction of stopping distance at braking, as compared with the operation of stand-alone active steering or torque vectoring control.

In closing it may be remarked that the system integration for over-actuated vehicles is primary attributed to the lateral dynamics control in the most of analyzed studies. The improvement of the longitudinal dynamic performance is being considered as a collateral task. However, importance of the IMC under the X+Y integration class is increasing for electric vehicles. In this case an appropriate combination of active systems is also beneficial for overall improvement of energy efficiency, when the integration strategy is setting up reasonable priorities for energy-saving operation of actuators and increase of energy regeneration.

VII. X+Z-INTEGRATION CLASS

The X+Z-integration class is rarely discussed in research literature. Among few relevant systems, the integrated ABS control and electric motor management, aimed at simultaneous optimization of the brake performance, energy efficiency and pitch dynamics of an electric vehicle, can be mentioned [61]. The most important task for the Z-domain - rollover stability - is mainly discussed by the Y+Z- and X+Y+Z-integration classes that will be shown in next sections.

VIII. Y+Z-INTEGRATION CLASS

The IMC under Y+Z-integration class aims principally at ensuring of lateral and roll stability of the vehicle. Concurrent improvement of the ride comfort at cornering manoeuvres is also beneficial. Though these tasks are relevant to individual vehicle dynamics control systems like ESC, their performance can be insufficient in some complex manoeuvres, especially by low-friction road surface conditions. This fact is confirmed in particular with extensive road tests of vehicles equipped with non-integrated and integrated variants of industrial vehicle dynamics control systems [62]. The reasonable integration of active chassis/powertrain systems under Y+Z class will be further discussed for several characteristic cases.

Some studies are investigating the possibility to correct the yaw and roll vehicle dynamics without brake-based or torque-based yaw stability control systems. But the use of alternative control systems like the active steering can be insufficient in many operational conditions. A possible solution variant is discussed in [63], where the active front steering is integrated with the active suspension. The proposed scheme has been studied for two integration methods: the state-based stochastic sub-optimal control technique and the hierarchical control, Fig. 13. In the first case the integration is performed in accordance with the optimization considering stability, handling, and ride comfort. In the second case the integration is based on the redistribution of the torque demand, required for the correction of vehicle motion, in accordance with the following rule:

$$\begin{cases} T_{AFS} = n_1 T_w + (1 - n_1)\, T_p \\ T_{AS} = n_2 T_p + (1 - n_2)\, T_w, \end{cases} \tag{13}$$

where T_w is the wheel torque, T_p is the pitch torque, n_1 and n_2 are the weighting coefficients. Both approaches have been investigated using the vehicle-in-the-loop testing procedures for the step steering input and double lane change. Particular effects are achieved for the hierarchical integration, where peak values during the double lane change were reduced on 25.1 % for the side slip angle and on 30,1 % for the vertical acceleration as compared to the vehicle with a non-integrated control.

A major part of the IMC systems belonging to the Y+Z class includes an integration of the active suspension with the brake control/brake-based yaw dynamics control [64]–[68]. Analysis of published studies shows that different methods are being proposed for such integration. For instance, the work [64] has presented a concept of the active suspension/brakes integration using the Linear Parameter Varying design and fault-tolerant control. The share of individual systems in formulation of the control demand is defined on the basis of three weighting functions for the lateral acceleration, the heave acceleration, and the suspension deflection. The control priorities are set up so that the brakes are in the operation mainly in the case of loss effectiveness of the suspension. This system has been investigated using simulation of cornering manoeuvres. The results confirmed applicability of this approach in supporting the lateral and roll stability in rough surface conditions. Another approach is discussed in [66] for the integration of the AS and the ESC. The active suspension is based on the linear quadratic optimal control with the optimal static output feedback methodology. The simulation results of fish-hook manoeuvre at various vehicle velocities have confirmed feasibility of the developed control approach but indicated that the active suspension has to be activated under rollover situations only. Otherwise it can deteriorate the driving comfort. This integration method has been modified in [68], in particular, by offering the objective function as

$$J = \int_0^\infty \left\{ \begin{array}{c} \rho_1 \left(\dot{\psi} - \dot{\psi}_{ref}\right)^2 + \rho_2 a_y^2 + \rho_3 \phi^2 + \rho_4 \dot{\phi}^2 \\ + \rho_5 M_{\psi br}^2 + \rho_6 M_{\psi sl}^2 + \rho_7 M_{\psi sr}^2 \end{array} \right\} dt, \tag{14}$$

where ρ_i are the weighting factors, ϕ is the roll angle, $M_{\psi br}$, $M_{\psi sl}$, $M_{\psi sr}$ are the control yaw moments by differential braking, and left-side and right-side active suspension forces correspondingly. In addition, the yaw moment distribution is realized by the weighted least square procedure. This IMC option has been also simulated for different fish-hook tests, which confirmed a more robust system operation accompanied with rollover speed increase on 36% in comparison to tests without the integrated control.

Another optimization approach is realized in the IMC system proposed in [65] and uniting the ESC and the active

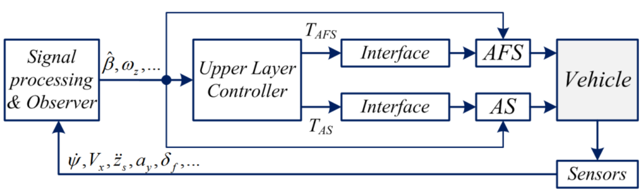

FIGURE 13. Integrated active steering and suspension system (adapted from [63]).

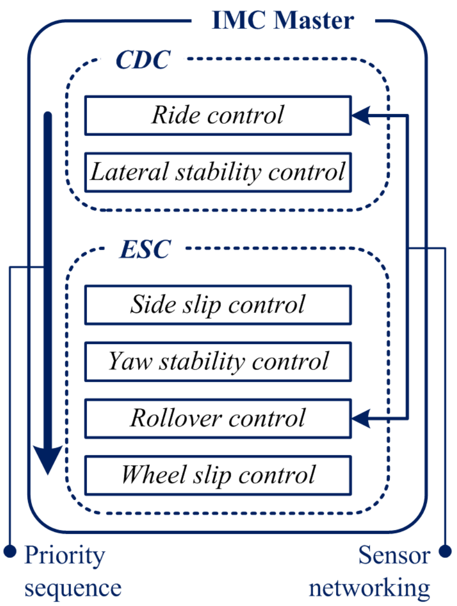

FIGURE 14. Master strategy of IMC (adapted from [65]).

suspension in the form of the continuous damping control (CDC). The integrated controller includes a set of functions, Fig. 14, in accordance with an appointed priority sequence. The sequence depends on the importance of the control sub-tasks in terms of vehicle safety. The reference vehicle states are calculated using a sliding mode control algorithm. It is proposed in [65] that the optimization of the integrated motion control law uses the procedure of the simulation-based worst-case dynamic evaluation (WCDE). The discussed study has reported that this method allows to find failure modes of the system operation (e.g. rollover and spin-out) already on the simulation stage, without carrying out of the standard tests on real vehicles. It can be considered as beneficial for the IMC design process.

Many recent studies consider the Y+Z-integration in context of over-actuated vehicles. A corresponding example is described in [69], where the combined operation of AS, the AFS and the WSC (active brakes) serves to control the yaw and roll stability, Fig. 15. The integration is organized using the rule-based approach. The yaw motion control is firstly realized through the AFS and WSC. Here the yaw moment ΔM_ψ to be corrected is defined from feed-forward $M_{\psi-ff}$ and feed-back $M_{\psi-fb}$ components from the reference model. The IMC is also observing the dynamic stability index DSI

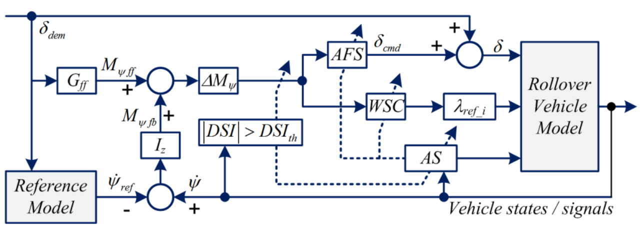

FIGURE 15. IMC of yaw and roll stability (adapted from [69]).

calculated as

$$DSI = \frac{2h_g}{B}\left(\frac{a_y}{g} + \frac{I\ddot{\rho}}{m_b g h_g}\right), \tag{15}$$

where h_g is the height of vehicle mass centre, B is the track, I is the moment of vehicle mass inertia, m_b is the unsprung mass of the vehicle, ρ is the roll angle. When DSI exceeds a certain threshold value DSI_{th}, the active suspension activates the rollover prevention control mode. The simulation of NHTSA fish-hook manoeuvres for the vehicle equipped with the proposed system architecture has showed that the IMC is able to reduce the roll angle by 26,3% as compared with the stand-alone operation of the active suspension only. Certain improvement has been also observed for root mean square of side slip angle and yaw rate - reduction on 47,1 % and on 8,2 % respectively as compared with the brake-based wheel slip control.

Another example of the IMC of the Y+Z-integration class is introduced in [70], where the all-wheel drive hybrid electric vehicle has the integration of the AFS, the ESC and the individual wheel torque control and, in addition, is supplemented with the anti-roll control system (ARCS). The proposed integration is based on advancement of the optimal control technique introduced above for the works [66] and [68]. The study has performed benchmarking of different control configurations by example of the simulation of a sine-with-dwell manoeuvre from 80 km/h. In particular, it can be estimated from the results of [70] for this particular manoeuvre that the inclusion of the ARCS into the IMC process allows to reduce the maximum roll angle from 4,09 to 1,44 deg and the maximum roll velocity from 20,15 to 10,02 deg/s. However, this inclusion did not improve the system performance in terms of the yaw stability criteria (maximum yaw rate and maximum side slip angle).

The roll control described in previous examples can be realized not only with active suspension elements. In particular, the studies [71], [72] have investigated the active stabilizer bar as element of the IMC. The work [73] has proposed the integrated yaw and roll stabilization with the combination of the ESC and aerodynamic actuators (active wing and rudder devices).

Several summarizing points can be mentioned after analysis of research publications dedicated to the IMC systems that are attributed to the Y+Z-integration class:

- Most of analyzed approaches are validated only in simulation that complicates an objective evaluation of overall system performance;
- Though the qualitative effect from the IMC in enhancement of lateral/yaw and roll dynamics has been observed in almost all cases, every so often the related quantitative improvement of corresponding performance indicators (maximum roll angle, tracking error of reference yaw rate, maximum side slip angle et al.) are marginal;
- Nevertheless, the systems of the Y+Z-integration classes have to be considered as an important technique from viewpoint of fail-safe and robust operation.

IX. X+Y+Z-INTEGRATION CLASS

The IMC covering all domains of vehicle dynamics has more comprehensive control logic in comparison to previously considered integration classes. Systems under the X+Y+Z-integration class have to properly arrange priorities between concurrent control tasks and to find an optimal vehicle performance at any safety- and efficiency-critical manoeuvres. The relevant IMC technique is discussed in next sections.

As for over-actuated/multi-actuated vehicles, the system architecture for the IMC attributed to the X+Y+Z-integration class is not essentially different from the X+Y-, X+Z- or Y+Z-classes in many cases. But distinguishing features can be found in the control logic. For example, the work [74] has used the integration of the AFS, the AS and the WSC. In the proposed system the active suspension has three control modes: (i) ride comfort (applied in normal control conditions), (ii) safety assistance (applied at emergency braking), (iii) stability assistance (applied at cornering manoeuvres). The wheel slip control has two modes: ABS and brake-based ESC. Depending on the vehicle motion parameters, the integrated controller identifies seven states and selects corresponding system combination, Table 1. The proposed rule-based approach can be easily implemented and ensures a certain level of robustness that was confirmed in [74] with simulation results of braking, single-lane and double-lane change manoeuvres at different road conditions. In particular, for the emergency braking, the system provided faster wheel speed response and better reference slip tracking due to the normal force variation by the active suspension.

TABLE 1. State-based IMC (adapted from [74]).

Vehicle motion parameters and conditions	Priority control task	System integration
Normal driving $\lvert\delta\rvert<\delta_{th} \wedge \lvert d\psi/dt\rvert\leq(d\psi/dt)_{th} \wedge \lvert a_x\rvert<a_{xth}$	Comfort	AS (ride comfort mode)
Emergency braking $\lvert\delta\rvert<\delta_{th} \wedge \lvert d\psi/dt\rvert\leq(d\psi/dt)_{th} \wedge \lvert a_x\rvert\geq a_{xth}$	Braking performance	WSC (ABS mode) + AS (safety mode)
Neutral or understeer motion $\lvert\delta\rvert<\delta_{th} \wedge \lvert d\psi/dt\rvert>(d\psi/dt)_{th}$	Handling	AFS + AS (stability mode)
Neutral or oversteer motion $\lvert\delta\rvert\geq\delta_{th} \wedge V_{ch}^2>0 \wedge V_{ch}^2>V^2$; $(1<1+V^2/V_{ch}^2\leq 2)$		
Split-μ braking $\lvert\delta\rvert\geq\delta_{th} \wedge V_{ch}^2>0 \wedge 0\leq V_{ch}^2< V^2$; $(2<1+V^2/V_{ch}^2)$	Stability	WSC (ESC mode) + AFS + AS (stability mode)
Unstable motion with high understeer $\lvert\delta\rvert\geq\delta_{th} \wedge V_{ch}^2<0 \wedge \lvert V_{ch}^2\rvert\geq V^2$; $(0<1+V^2/V_{ch}^2\leq 1)$		
Unstable motion with high oversteer $\lvert\delta\rvert\geq\delta_{th} \wedge V_{ch}^2<0 \wedge \lvert V_{ch}^2\rvert< V^2$; $(1+V^2/V_{ch}^2\leq 0)$		
δ_{th} is the steering angle threshold, V_{ch} is the characteristic vehicle velocity computed according to the Hurwitz stability criterion, a_{xth} is the longitudinal acceleration threshold, $(d\psi/dt)_{th}$ is the yaw rate threshold		

For cornering manoeuvres, the risk of critical situations in terms of the roll and yaw stability has been also reduced as compared with non-integrated system configurations.

The X+Y+Z-integration class can be also represented with the system architecture, when only two systems are responsible for the vehicle dynamics control in all domains (in such a case the vehicle has to be considered as multi-actuated, not over-actuated). This case can be referred, for example, to the work [75], where the active suspension is integrated with the brake control in accordance with the following scheme, Fig. 16. The control on the vehicle motion is realized through brake torques T_{brc} and suspension forces F_{zc}. In addition, the controller is synthesized using linear parameter varying (LPV)/H$_\infty$ method to address in particular a disturbance attenuation problem. The work [75] has investigated this approach by way of the double lane change simulation with variation of road surface conditions. It was observed that the proposed integration not only ensure required stability and handling but also improve efficiency of the actuator use for both active systems.

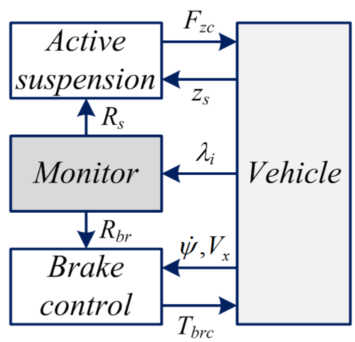

FIGURE 16. Integration variant of active suspension and brake control (adapted from [75]).

The combined use of the LPV/ H$_\infty$ technique for the complex IMC is also discussed in [76] as applied to three active systems: suspension, brakes, and steering. The work [77] has advanced the IMC logic and used the LPV/H$_\infty$ for the vertical dynamics controller and the flatness based method for the longitudinal/lateral controller. In both cases the integration procedures was realized through monitoring of several indexes, as for previous case from [75]. The simulation of the vehicle double lane change manoeuvre with the proposed IMC indicated less oscillations of longitudinal vehicle velocity as well as improvement of the load transfer and stability indexes. Hence, the contribution to the performance enhancement for three domains of the vehicle dynamics has been achieved.

It should be mentioned that several other complex control tools are investigated in various published studies dedicated to the system integration for over-actuated/multi-actuated vehicles. In particular, the article [78] has proposed reinforcement learning control. The works [79]–[81] have discussed diverse control allocation approaches. However, lack of (i) results experimentally validated in real driving conditions and (ii) benchmarking procedures for competitive control technique does not allow to favour one or another method.

Analysis of other studies points out that the further development of the IMC under discussion can be related to X-by-wire technologies and electric vehicles. Here, essential progress was achieved in the middle of 2000s with the first prototypes and concepts of vehicles with the powertrain composed from two, three or four individual electric motors. In particular, introduction of an in-wheel motor was a key step in transition from already existing integrated chassis management to integrated wheel corners. The first and best known technologies in this line were proposed by Siemens VDO, Michelin and Volvo. The Autonomous Corner Module by Volvo has been investigated in different constructive variants with special attention given to the design aspects of suspension and embedding of planetary gear into the wheel hub unit [9], [82]. eCorner of Siemens VDO integrated electric propulsion with active brake, damping and steering systems [83], [84]. A particular feature of the eCorner concept was the use of fully electro-mechanical brake-by-wire systems as the electronic wedge brake. An alternative wheel corner solution has been developed as Michelin Active Wheel [85]. This system had the same set of active systems like eCorner but was supplemented with specific tyre design. These and other wheel corner concepts are strong examples of innovative engineering ideas, however their wide application on the vehicles is still limited due to number of technological limitations.

X. ANALYSIS OF INTEGRATION METHODS

IMC variants analyzed in previous sections have diverse principles of the integration of individual active systems. A selection of the integration architecture depends on many factors and has to take into account response speed and energy consumption of actuators, limitations of individual systems in terms of corrective torques and forces, et al. Of particular importance is also the fact that for some system configurations the simultaneous tracking and control on vehicle dynamics parameters from different domains can be even considered as generically unsolvable problem [86]. In this regard the widespread integration methods merit consideration here. An extended classification of available and potential integration architecture for the vehicle control has been introduced in [87]. Following this classification, the IMC can be organized as (i) centralised, (ii) supervisory, (iii) hierarchical and (iv) coordinated control. It should be noted that there are no detailed recommendations in the analyzed studies about the most appropriate variant of the integration architecture for one or another combination of active chassis and powertrain systems.

Within the framework of the presented paper, a simplified classification of integration methods is proposed. Two classes - *single-criterion* IMC and *multi-criterion* IMC - are conditionally designated. In the first case, the actuation or inclusion of individual systems into the control process is selected on the basis of a certain specific parameter of the vehicle dynamics. In the second case, the operation share of all participating individual systems is defined by several concurrent parameters characterizing different vehicle properties. Several most relevant examples for both cases are introduced next with particular attention given to vehicle dynamics parameters included into the decision-making process.

A. SINGLE-CRITERION IMC

A simple approach for the single-criterion IMC is to apply a rule-based coordination of individual systems, where the rules relate to one or more performance indicators of vehicle dynamics. A corresponding example has been introduced in [88]. This study investigated the active steering, the active differential and the brake control under the Y-integration class. The control is based on the estimation of the side slip dynamics and uses β-$d\beta/dt$ phase plan to form the integration rules, Fig. 17. It can be seen here that firstly only active steering is included into the integrated control chain. Then subsequent activations of the active driveline and the active braking take place, when an actual driving situation becomes more critical,. Results of steer input simulations confirmed that the proposed order of the systems intervention provides a reasonable trade-off between stability and steerability and reduces influence on the longitudinal vehicle dynamics [88]. The discussed method has been extended in [89] with the inclusion of active suspension. However, subsequent simulation of double-lane change manoeuvres with this extended configuration detected only minor improvements in the yaw rate and side slip dynamics.

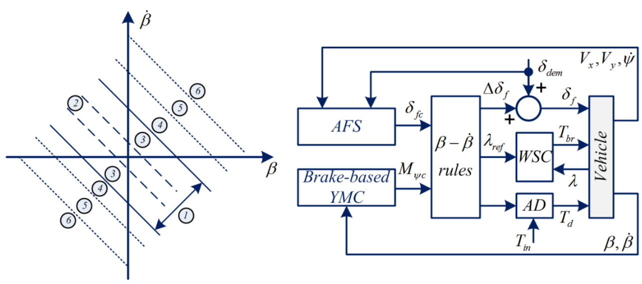

FIGURE 17. Different regions in β-$d\beta/dt$ phase plan for the rule based integration scheme (adapted from [88]); 1 - reference region for control design, 2 - active steering inclusion for steerability improvement, 3 - transitions between control tasks, 4 - active steering inclusion for stability improvement, 5 - active driveline inclusion, 6 - active braking inclusion.

The rule-based β-$d\beta/dt$ phase plan method has been also subjected to benchmarking with the Characteristic Locus (CL) method in [90]. The CL approach estimates firstly the stability of the IMC as a multi-input multi-output feedback system using characteristic locus, and then proposes the design of a multivariable compensator from the spectral decomposition of the transfer function matrix describing the system. The application example from [90] concerns the integration of only two systems, AFS and ESC. For this system combination, the simulation of cornering manoeuvres pointed out that the CL method has a better performance in comparison with the rule-based approach in terms of less operation time of the brake-based ESC (benefits from viewpoint of energy consumption by actuators and tyre energy dissipation).

Other studies propose more practical and simpler IMC methods, when the integration engages limited number of systems. In particular, the work [91] has investigated the integration of the brake-based and torque-based yaw dynamics control (AD, ESC, TV) and used the state-based method. (i) If the requested correcting yaw torque M_z^{req} can be generated by the active differential only, then ESC and TV systems are deactivated. (ii) If the requested correcting yaw torque M_z^{req} can not be fully covered by the active differential, then the active differential still generates the maximum torque $M_{z,diff}^{max}$ and the rest torque $(M_z^{req} - M_{z,diff}^{max})$ is equally generated by the ESC and TV (share of 50% for each system) under condition that the vehicle velocity should not be changed. (iii) If it is not possible to keep the vehicle velocity and to correct yaw dynamics simultaneously, the share of ESC control is increased to generate required yaw torque in spite of possible velocity reduction. Hence, such an approach considers both criterion of vehicle performance (velocity) and safety (yaw dynamics) during the control process and tries to minimize the losses in performance (velocity reduction) by simultaneous ensuring the vehicle stability.

To improve the IMC performance, the actuators characteristics has to be also included into consideration by defining the integration principle. In such a case the tools that are differ from the rule-based technique could be useful. For example, the study [92] is described the general yaw moment control (Y-integration class) and proposed to use the control allocation in the following form. The global yaw moment to be realized by N actuators available in the integrated system is

$$\begin{aligned} T_{global}(t) &= T_{Act1}(t) + T_{Act2}(t) + \ldots + T_{Actn-1}(t) \\ &\quad + T_{Actn}(t) = \sum_{i=1}^{N} T_{Acti}(t), \end{aligned} \tag{16}$$

The yaw torque request for particular actuator can be written as

$$T_{Acti} = T_{\max i}\left(a_y\right) u_i(t), \tag{17}$$

where T_{max} is the maximum yaw torque that could be generated by the actuator for given maneuver conditions characterized by the lateral acceleration a_y, $u \in [0;1]$ is a multiplication factor. Hence, the integration task can be reduced to the optimization problem

$$\min(\|u\|) \text{ subject to } \underline{T} \cdot \underline{u} = T_{global}. \tag{18}$$

Such an approach allows to eliminate actuators with less potential from the control process, when required. However, optimization procedures can be more resource-consuming by real-time processing in an on-board vehicle controller. Nevertheless, the control allocation is becoming now more and more attention, especially in context of the multi-criterion IMC discussed in next section.

Some summarized remarks can be done for the single-criterion IMC principle:

- Most of relevant solutions are using the rule-based methods (including fuzzy methods) that is favourably for easier real-time implementation on the vehicle;
- Analyzed single-criterion integration examples are generally considered the balance between two vehicles properties, in particular, stability vs. performance and stability vs. driving comfort; Inclusion of more vehicle attributes into the integration strategy can requires more complex integration architecture.

B. MULTI-CRITERION IMC

Analysis of the research publications allows to note that the multicriterion IMC systems have mainly a multi-layer structure. To give an example, Fig. 18 displays the architecture of the multi-layer vehicle dynamics control system proposed in [93] for an electric vehicle with two electric motors. Here the layer of supervisory controller defines the reference vehicle dynamics. The upper level controller computes the actual parameters like vehicle forces and yaw moment as inputs for the lower level controller, which determines commands for the actuators. The activation of systems is essentially depended on actuator constraints, which are set up by the lower level controller. For instance, as applied to the case study from [93], the supervisory controller defines firstly the control mode in accordance with Fig. 18. But then the lower level controller performs the optimization of control actions by criteria of driving efficiency, minimization of allocation error, and wheel slip limits. After the optimization, the actuator commands are distributed individually to each actuator - electric motors and individual wheel brakes in the considered case. This approach has been validated by simulation of a complex manoeuvre including constant turning and several double-lane changes and confirmed better performance in terms of rollover index, minimization of the side slip and yaw rate error tracking as compared with the event-based ESC.

Many multi-criterion IMC systems are based on the control allocation methods [9], [79], [80], [94], [95]. The control allocation requires optimization procedures to define the individual contribution of each individual system (actuator) to the control process. The different parameters of vehicle dynamics can be used in this regard. In particular, the study [9] has investigated a concept of the electric vehicle equipped with the motor torque control, the active steering, and the active suspension elements. The IMC under discussion belongs to the X+Y+Z-integration class. The operation of individual actuators is defined using the optimization problem with the following particular objectives: to minimize the tyre wear by minimizing the tyre slip angles α_i, to minimize the energy consumption by minimizing the longitudinal tyre forces F_{xi} subjected to the tyre slip angles and wheel slip λ_i, and to minimize the utilization of the tyre

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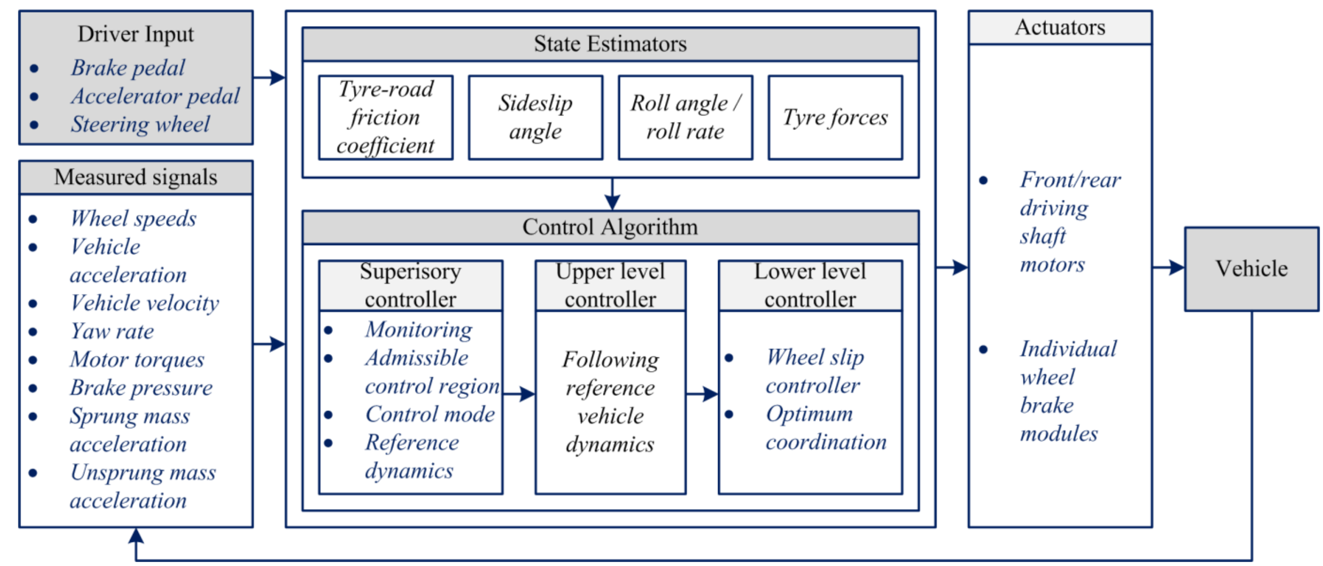

FIGURE 18. Multi-layer IMC architecture and corresponding admissible control region (adapted from [93]).

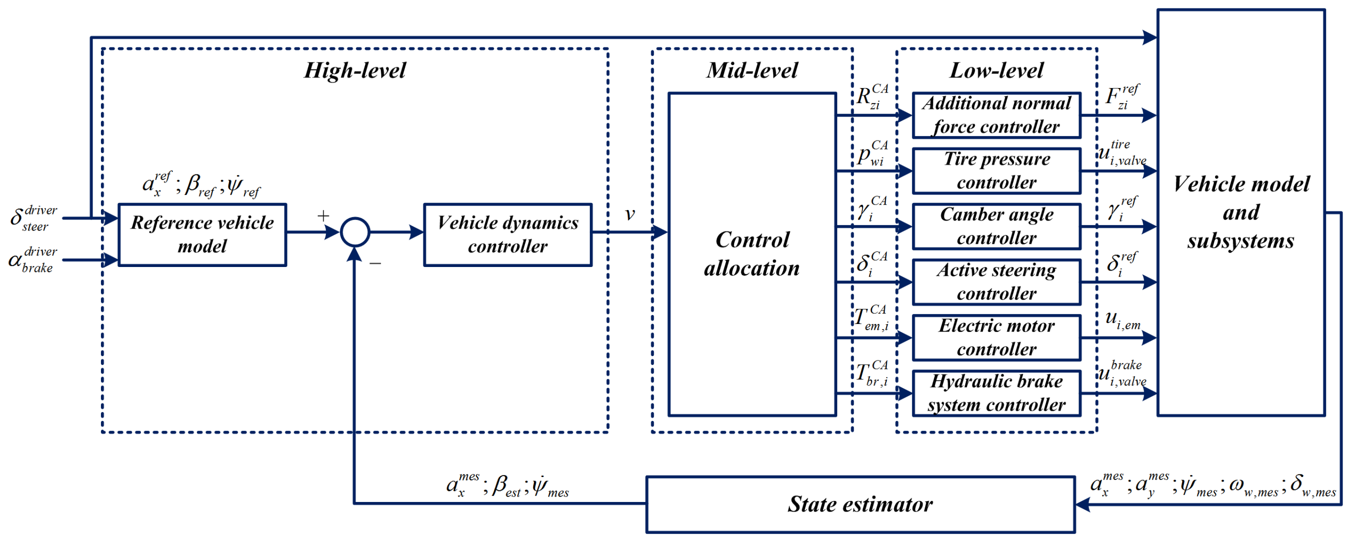

FIGURE 19. Block diagram of integrated active safety system for electric vehicle (adapted from [98]).

friction potential:

$$\min_{\alpha_i}\left(f_1 = \sum_{i=1\ldots4}|\alpha_i|\right); \quad \min_{\alpha_i,\lambda_i}\left(f_2 = \sum_{i=1\ldots4}F_{xi}\right);$$
$$\min_{\alpha_i,\lambda_i}\left(f_3 = \max_{i=1\ldots4}\sum_{i=1\ldots4}\frac{\sqrt{F_{xi}^2+F_{yi}^2}}{\mu\cdot F_{zi}}\right). \tag{19}$$

Then the corresponding multi-objective optimization problem can be solved as a weighted sum of three mentioned objectives with weighting factors g_k:

$$\min_{\alpha_i,\lambda_i}\left(f = \sum_{k=1\ldots3}(g_k\cdot f_k)\right). \tag{20}$$

The simulation of slalom and acceleration tests for the small electric vehicle utilizing the described IMC has confirmed the feasibility of the described methods and indicated in addition its benefits for disturbances rejection.

An advanced approach has been successively investigated in [96]–[99]. The developed concept of the integrated system concerns the electric vehicle with individually controlled in-wheel motors and a set of active subsystems for brakes, steering, camber angle, and tyre inflation pressure, Fig. 19. Taking into account the operation of electric motors in driving and braking modes, the discussed integrated control operates six subsystems in total. In accordance with Fig. 19, the proposed control strategy requires (i) vehicle state estimator based on extended Kalman filter, (ii) high-level controller of vehicle motion, (iii) middle-level control allocation, and (iv) lower-level individual controllers of each subsystem. The high-level control consists of reference vehicle model and vehicle dynamics controller. Generalized longitudinal and yaw torque are determined according to control errors. The reference longitudinal acceleration is calculated based on the measured pedal force. The middle-level control allocation takes into account that relationship between generalized forces and yaw torque from control demand and control inputs is nonlinear. The feedback linearization by cancelling of nonlinear term is used in the middle-level block:

$$\nu * = \nu - f(x, u_{k-1}) \approx B_{u(k-1)} u_k \tag{21}$$

Since six vehicle subsystems are considered, and electric motors can develop positive (traction) and negative (brake) torques, the control input vector is given as:

$$u^{CA} = \begin{bmatrix} \delta_{afs}^{CA}, \delta_{ars}^{CA}, M_{em,i}^{CA,pos}, M_{em,i}^{CA,neg}, \\ M_{br,i}^{CA}, \gamma_i^{CA}, \Delta p_{wi}^{CA} \end{bmatrix}^T. \tag{22}$$

The control effectiveness matrix Bu is equal to:

$$B_u = \left[B_{steer}, B_{emotor}^{pos}, B_{emotor}^{neg}, B_{brake}, B_{\gamma}, B_{pw} \right] \tag{23}$$

The aim of control allocation is to minimize allocation error $(B_u u^{CA} - \nu^*)$ and control energy, taking into account actuator constraints. The final constraints for optimization include also tyre constraints, actuator position and rates limits. The lower-level actuator control should (i) guarantee a precise tracking of reference control signals obtained from the middle level of the controller and (ii) estimate boundary conditions for control allocation taking into account subsystem dynamics. The described concept has been tested on the HIL test rig for the complex cornering manoeuvre and confirmed that the developed control system demonstrates good tracking of reference yaw rate with RMSE below 3.0 deg/s. The vehicle sideslip angle does not exceed 5.0 deg. The chosen control allocation strategy has also allowed to optimize energy consumption of electric motors and to keep the tyre dissipation on the low level.

Summarizing, it can be mentioned for the multi-criterion IMC with the control allocation that the complexity of problems of optimal distribution algorithms and definition of actuator constraints depends strongly from number of control systems (actuators) and objectives to be reached through the integrated control (cost functions for the optimization). It allows to optimize the vehicle dynamics in an efficient way not only from viewpoint of safety and performance but also with taking into account of ride comfort, driveability and other vehicle properties. However, an increased complexity of the multi-criterion IMC can be also considered as a certain obstacle for the real-time controller implementation.

XI. SUMMARY

The presented analysis of different integrated motion control systems allows to conclude that they can be classified in accordance with longitudinal, lateral and vertical domains of vehicle dynamics. For the proposed classes each, various configurations of active chassis and powertrain systems are investigated in the research literature. However, despite this variety, only a few IMC systems have been validated and verified on full-scale vehicles - most of published results are based on the simulation or the HIL tests only.

Another critical point for current research in the integrated vehicle dynamics control is the lack of clear benchmarking criteria to identify optimal or the most efficient combinations of active systems. In particular, the reviewed IMC systems have demonstrated certain effects in improvement of one or another vehicle characteristic as stability, braking performance, tyre friction utilization et al., but an analysis of corresponding energy consumption due to operation of several systems for the same control tasks is often missing. It point clearly to the demand for the development of corresponding value engineering methods to be used in the IMC design.

Further investigations are also required for the establishment of common standard procedures to evaluate IMC performance in complex motion cases. Available test standards are defining particular situations from corresponding domains of vehicle dynamics, for instance, braking, lane change maneuvers et al. The advancement or development of new test procedures with simultaneous criteria of safety, driving comfort, energy efficiency is strongly needed.

The listed aspects have to be considered by the development of future IMC systems.

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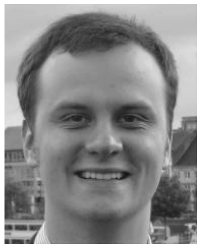

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