Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility

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Published online: 28 Jan 2009.

To cite this article: Daofei Li, Shangqian Du & Fan Yu (2008) Integrated vehicle chassis control based on direct yaw moment, active steering and active stabiliser, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 46:S1, 341-351, DOI: 10.1080/00423110801939204

To link to this article: http://dx.doi.org/10.1080/00423110801939204

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Integrated vehicle chassis control based on direct yaw moment, active steering and active stabiliser

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(Received 31 July 2007; final version received 23 January 2008)

In order to coordinate steering, braking/traction and active stabiliser, an integrated vehicle chassis controller is proposed. The controller is realised via a main/servo-loop structure: in the main-loop, stabilising forces/moments are first calculated using robust sliding mode controller; then in the servo-loop, the stabilising forces/moments are optimally distributed to tyre control inputs, i.e. active steering and wheel torques. To reduce body roll angle and assist to track the desired yaw rate, active stabilisers are further integrated. Through open-loop and closed-loop simulations using Matlab/Simulink® and MSC CarSim®, it is verified that the controller can significantly improve vehicle handling performances and ride in comfort in the meantime.

Keywords: vehicle dynamics; tyre nonlinearities; active steering; direct yaw moment control; active stabiliser

1. Introduction

Under critical driving circumstances, e.g. emergency cornering, it is normally difficult for a driver to stabilise the vehicle, and accidents could occur in such critical driving situations. To ensure vehicle active safety performance, many advanced active chassis control systems, e.g. active front steering, direct yaw moment control (DYC) and active roll control (ARC) system, have been developed and brought into the market.

By applying an additional steering angle to front/rear wheels, active steering can improve the lateral vehicle dynamics performances by directly regulating tyre slip angles and thus lateral tyre forces. Unfortunately, as many previous researches have shown that when lateral tyre force is near saturation, active steering becomes less effective. On the other hand, braking-based DYC is considered to be an effective solution to improve vehicle active safety in emergency situations. However, since the essential principle of DYC is to regulate the tyre longitudinal force differences in left and right wheels instead of using tyre lateral force directly, it may not be sufficiently effective in controlling vehicle sideslip angle, which is a crucial state variable for vehicle performance.

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ISSN 0042-3114 print/ISSN 1744-5159 online
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DOI: 10.1080/00423110801939204
http://www.informaworld.com
Therefore, in order to compensate the drawbacks described above and to improve vehicle handling and stability performances further, many researchers investigated the potentials of integrated vehicle chassis control (IVCC) system during the past few years [1]. Particularly, extensive researches have been carried out on the integration of DYC and active steering, e.g. [2–6]. However, based on the linear tyre model, most of these pioneer works focused on how to calculate the desired active yaw moment $M_{zd}$, while few of them studied the detailed distribution of stabilising forces/moments to tyre inputs. Some of them adopted relatively simple methods, e.g. to distribute $M_{zd}$ only to braking forces in one single axle. Very recently, some researchers began to focus on detailed forces/moments distribution. For instance, based on different cost functions (e.g. tyre workload and control inputs), different optimisation methods are proposed to distribute the stabilising forces [7–10]. As a persisting problem, it is still quite necessary to further investigate on how to ‘optimally’ distribute the stabilising forces/moments to each subsystem.

On the other hand, as a cost-effective active suspension, ARC system can improve ride comfort and roll stability during emergency cornering. In addition to reducing body roll angle, ARC can also affect vehicle handling stability through active control of front/rear roll stiffness distribution ratio [11–13]. It is based on the fact that when tyre vertical load is changed, tyre cornering stiffness will change accordingly but nonlinearly. The re-distribution of roll moment can affect the front/rear cornering stiffness through varying tyre normal loads. However, there are still few research works about integrating active suspension systems (particularly ARC) with other subsystems [14–18].

In this paper, an integrated controller, coordinating DYC, active four-wheel steering (4WS) and ARC, is developed for a ‘future’ vehicle in which its four wheels can be independently steered/braked/driven, and its front/rear axles are also equipped with active stabilisers. The proposed chassis controller will be evaluated through the co-simulation of Matlab/Simulink® and MSC CarSim®, the latter of which is a professional software for vehicle dynamics.

2. Vehicle modelling

2.1. Nonlinear tyre modelling

A simple tyre model of Burckhardt [19] is adopted to describe tyre combined slip characteristics with consideration of nonlinearity. Along with the full vehicle model, tyre forces and relevant angles are also indicated in Figure 1.

The longitudinal slip $s_L$ and lateral slip $s_S$ are defined as

$$s_L = \begin{cases} \frac{(v_R \cos \alpha - v_W)}{v_W}, & v_R \cos \alpha \leq v_W \\ \frac{(v_R \cos \alpha - v_W)}{(v_R \cos \alpha)}, & v_R \cos \alpha > v_W \end{cases} \quad (1)$$

$$s_S = \begin{cases} (1 + s_L) \tan \alpha, & v_R \cos \alpha \leq v_W \\ \tan \alpha, & v_R \cos \alpha > v_W \end{cases} \quad (2)$$

In Equations (1) and (2), $\alpha$ is tyre slip angle, $v_R$ and $v_W$ are free rolling speed and centre forward speed of wheel, respectively. The resultant slip is $s_{Res} = \sqrt{s_L^2 + s_S^2}$. Given the three factors ($c_1$, $c_2$, and $c_3$), which correspond to road friction conditions, the friction coefficient $\mu_{Res}$ and tyre forces can be calculated as Equations (3) and (4),

$$\mu_{Res}(s_{Res}) = c_1(1 - e^{-c_2s_{Res}}) - c_3s_{Res} \quad (3)$$

$$F_L = \mu_{L}F_z = s_LF_z\mu_{Res}/s_{Res}$$

$$F_S = \mu_{S}F_z = s_SF_z\mu_{Res}/s_{Res} \quad (4)$$
As indicated in Figure 2, the calculated lateral forces under varying vertical loads are compared with test data, implying that the adopted Burckhardt tyre model can describe tyre nonlinearities with acceptable accuracy.

2.2. Simplified vehicle model

For main-loop controller design, the full vehicle model, shown in Figure 1, has four degrees of freedom, i.e. longitudinal, lateral, yaw and roll motion. In horizontal plane, the simplified
vehicle dynamics can be represented as follows.

\[
\begin{bmatrix}
    m_v (\dot{v}_x - v_y r) \\
    m_v (\dot{v}_y + v_x r) \\
    I_{zz} \dot{r}
\end{bmatrix}
= \begin{bmatrix}
    F_x \\
    F_y \\
    M_z
\end{bmatrix} = F_u
\]

(5)

In Equation (5), the generalised force \( F_u \) for vehicle motion control can be described as

\[
F_u = [F_x \ F_y \ F_z]^T = M_f [F_{Li} \ F_{Si}]^T = M_f F_t
\]

(6)

where \( M_f \) is a conversion matrix which is determined by vehicle parameters and dynamics state variables.

For body roll motion, the simplified dynamics can be written as

\[
I_{xx} \ddot{\phi} = m_b h_0 a_y - C_\phi \dot{\phi} - (K_{\phi f} + K_{\phi r} - m_b g h_0) \phi
\]

(7)

where \( C_\phi \) is the total roll damping, \( K_{\phi f} \) and \( K_{\phi r} \) are the roll stiffness of front and rear axles (including the stiffness of passive suspensions and active stabilisers), i.e. \( K_{\phi f} = K_{\phi sf} + K_{af} \), \( K_{\phi r} = K_{\phi sr} + K_{ar} \).

3. Integrated vehicle chassis controller design

3.1. Control structure

As shown in Figure 3, the total control task is functionally realised via two loops, i.e. the main-loop and the servo-loop. In the main-loop, driver intention (cornering or braking) is first expressed into a reference model. In order to follow this reference model, a sliding mode controller is adopted to compute the desired stabilising force \( F_{ud} \), i.e. \([F_{xd} \ F_{yd} \ M_{zd}]^T \) in the horizontal plane. Also, an active roll torque \( T_{xa} \) is computed in the main-loop to reduce body roll angle.

Accordingly, the servo-loop is to further distribute \( F_{ud} \) to desired variables of four tyres, i.e. tyre slip ratios and slip angles. And a roll moment distribution controller will determine the active roll torques to be produced by front and rear stabilisers. By subsystem regulators, the desired tyre variables and roll torques will be converted to final actions of actuators. For instance, a PID-based slip controller will convert the desired wheel slip ratios \( s_{Li} \) to wheel control torques \( T_{wi} \).
3.2. Main-loop controller design

3.2.1. Reference model

The desired longitudinal velocity \( v_{xd} \) can be calculated from driver acceleration or deceleration command \( a_{xd} \). For stability consideration, vehicle sideslip motion should be suppressed to the minimum level. Assuming zero desired understeer gradient, the desired yaw rate \( r_d \) can be derived from the driver steering angle \( \delta_D \) and longitudinal velocity \( v_x \). Furthermore, it is also necessary to limit the maximum value of \( r_d \) to avoid large lateral acceleration that exceeds tyre cornering capability. Therefore, the reference model can be summarised as follows:

\[
\begin{align*}
v_{xd} &= v_{x0} + \int_{0}^{t} a_{xd}(\tau) d\tau \\
v_{yd} &= 0 \\
r_d &= \frac{v_x}{L} \delta_D, |r_d| \leq \mu_{peak} \cdot g/v_x \quad (8)
\end{align*}
\]

where \( v_{x0} \) is the initial longitudinal velocity, \( L \) is the wheel base, and \( \mu_{peak} \) is the peak adhesion coefficient.

3.2.2. Sliding mode controller

The designed controller should be robust enough to adapt to many uncertainties, e.g. variation of vehicle parameters and sensor noises. Therefore, in the main-loop, a robust sliding mode method is employed to control this nonlinear vehicle system with coupling [20, 21]. The simplified vehicle dynamics in Equation 5 can be rewritten as follows:

\[ \dot{X}_i = f_i(X, t) + g_i(X, t) \cdot U_i, \quad (9) \]

where \( X = [v_x, v_y, r]^T \), and the control inputs are \( U = F_{ud} = [F_{xd} F_{yd} M_{zd}]^T \).

In order to suppress the tracking error of motion variables, the sliding function is chosen as

\[ S_i = e_i + \Lambda_i \xi_i, \quad (10) \]

where \( e_i \) is the motion state error, i.e. \( e_i = X_i - X_{id} \); \( \xi_i \) is the error integration, i.e. \( \xi_i = \int_{0}^{t} e_i(\tau) d\tau \); and \( \Lambda_i \) is a positive coefficient.

Then the control law that satisfies the sliding condition can be obtained as Equation (11). By straightforward derivation [22], it can be verified that with the designed control law, the sliding surfaces defined above are reachable.

\[ U_i = \hat{g}_i^{-1}[\hat{u}_i - k_{1i}S_i - k_{2i}\text{sgn}(S_i)] \quad (11) \]

In Equation (11), \( \hat{f}_i \) and \( \hat{g}_i \) are the nominal values of \( f_i \) and \( g_i \), respectively; \( \hat{u}_i = -\hat{f}_i + \hat{X}_{id} - \Lambda_i e_i; k_{1i} \) and \( k_{2i} \) are control parameters. To avoid chattering of control inputs, the sign function \( \text{sgn}(S) \) is replaced by a saturation function defined as

\[ \text{sat}(S_i/\Phi_i) = \begin{cases} 
\text{sgn}(S_i), & |S_i| \geq \Phi_i, \\
S_i/\Phi_i, & |S_i| < \Phi_i.
\end{cases} \]

Finally, the desired force \( F_{ud} \) for vehicle motion control can be summarised as Equation (12):

\[
F_{ud} = \begin{bmatrix}
m_v(v_{xd} - v_y r - \Lambda_1 e_1 - k_{11}S_1 - k_{21}\text{sat}(S_1/\Phi_1)) \\
m_v(v_x r - \Lambda_2 e_2 - k_{12}S_2 - k_{22}\text{sat}(S_2/\Phi_2)) \\
I_{zz}(-\Lambda_3 e_3 - k_{13}S_3 - k_{23}\text{sat}(S_3/\Phi_3))
\end{bmatrix}.
\]

(12)
3.2.3. Roll angle controller

Since body roll motion is generated by the lateral acceleration of sprung mass, the active stabilising roll torque is calculated by a simple proportional controller to reduce roll angle, i.e.

\[ T_{xa} = k_R a_y, \]  

where \( k_R \) is a tuning parameter of controller.

3.3. Servo-loop controller design

3.3.1. Horizontal forces/moments distribution

Basically, the distribution of \( F_{ud} \) to four tyres is a nonlinearly constrained optimisation problem, where the constraints include tyre–road adhesion constraints (‘friction circle’), rate/magnitude limits of active steering and wheel torque. In this paper, aiming to stabilise the vehicle by optimal actuator actions, sequential quadratic programming (SQP) approach [7] is adopted to deal with these constraints. The original constrained optimisation problem is converted to an unconstrained programming problem, in which the changing rates and the magnitudes of actuator inputs are suppressed to the minimum level.

Firstly a weighted cost function \( J_{SQP} \) is established as

\[ J_{SQP} = E^T W_E E + \Delta u_c^T W_{\Delta u_c} \Delta u_c + u_c^T W_{u_c} u_c, \]  

where the force tracking errors \( E \) are defined as the difference between desired value and actual value of vehicle control forces, i.e. \( E = F_{ud} - F_u \); \( u_c \) are the tyre variable magnitudes, i.e. tyre longitudinal slip ratios and slip angles; and \( \Delta u_c \) are defined as tyre variable increments that need to be regulated, i.e. \( \Delta u_c = [\Delta s_{L1}, \Delta s_{L2}, \Delta s_{L3}, \Delta s_{L4}, \Delta \alpha_l, \Delta \alpha_r]^T \); while \( W_E, W_{\Delta u_c} \) and \( W_{u_c} \) are their corresponding weighting matrices. Note that in selecting appropriate weighting matrices, both force tracking performance and actuator limits should be taken into consideration.

At the \((k + 1)\)th time step, there exist the following two equations:

\[ u_c(k + 1) = u_c(k) + \Delta u_c, \]  

\[ F_u(k + 1) \approx F_u(k) + J_{cob} \cdot \Delta u_c, \]  

where \( J_{cob} \) is the corresponding Jacobian matrix, defined as \( J_{cob} = \partial F_u / \partial u_c \). By referring to Section 2, \( J_{cob} \) can be expressed as \( J_{cob} = M_{fz,s} \cdot M_{BF,s} = M_f \cdot (\partial F_l / \partial u_c) \). In each time step, the elements of \( J_{cob} \) are determined by vehicle parameters and feedback information from sensors or estimators, e.g. tyre vertical loads \( F_{zj} \), and tyre variables etc. In fact, it is \( J_{cob} \) that reflects the actual control capability of different tyre variables.

Secondly, since the cost function \( J_{SQP} \) should be minimised with respect to the control input increments \( \Delta u_c \), the optimal trade-off among force tracking errors \( E \), control increments \( \Delta u_c \) and control magnitudes \( u_c \), can be achieved via Equation (17):

\[ \partial J_{SQP} / \partial \Delta u_c = 0. \]  

By combining Equations (15) and (16), the desired increments of tyre variables can be solved from Equation (18) as follows:

\[ \Delta u_c^d = (W_{u_c} + W_{\Delta u_c} + J_{cob}^T W_E J_{cob})^{-1} J_{cob}^T W_E \tilde{E} - W_{u_c} u_c(k), \]  

where \( \tilde{E} \) is the difference between the desired and the current values of stabilising forces, i.e. \( \tilde{E} = F_{ud} - F_u(k) \).
Finally, through wheel slip controller and steering regulator [22], the calculated optimal
tyre variables from Equation (18) will be further converted into tyre control inputs, i.e. wheel
torques and active steering angles.

3.3.2. Roll moment distribution control

The active roll torque $T_{xa}$ given in the main-loop should be properly distributed to front and rear
axles. Unlike tyre slip angle and slip ratio, roll moment distribution ratio affects vehicle han-
dling only indirectly via load transfer. Besides, as shown in Equation (18), tyre vertical loads $F_{zi}$ can also be directly considered through $J_{cob}$. Therefore, here active stabilisers only play
an auxiliary role in controlling yaw motion, while their main task is to reduce body roll angle.

Let $\lambda_f$ be the ratio of front roll stiffness to total roll stiffness, i.e. $\lambda_f = K_{\phi f}/(K_{\phi f} + K_{\phi r})$; then a proportional-integral control law is adopted to track the desired yaw rate $r_d$.

$$\lambda_f = k_{PR}(r_d - r) + k_{IR}\int(r_d - r)dt + \lambda_{f0}, \lambda_f \in [\lambda_{f\min} \lambda_{f\max}], \quad (19)$$

where control parameters $k_{PR}$ and $k_{IR}$ are the proportional and the integral gains, respectively,
and $\lambda_{f0}$ is the initial ratio of stabilisers. The range of distribution ratio is determined based on
the mechanical limits of stabiliser.

4. Simulation and discussion

As shown in Figure 3, the proposed IVCC is evaluated via co-simulation of Matlab/Simulink®
and MSC CarSim®. The controller designed in the Simulink® module will output steering
angles and wheel torques to the CarSim® module; while in the CarSim® module, the vehicle
model with fully nonlinear independent suspensions and nonlinear tyres is established as a
virtual vehicle.

The simulations include open-loop step steer manoeuvre and closed-loop double-lane-
change manoeuvre. The responses of vehicles with different configurations are compared
with each other. Note that both individual controllers, i.e. DYC and 4WS, have the same control structure as designed in Section 3.3.1, i.e. using main/servo-loop, while the difference
consists in that they have only wheel torque or active steering angle as control inputs. The
presented simulation results are based on the following condition: the initial speed is 120 km/h
and the road surface is dry asphalt.

4.1. Open-loop step steer manoeuvre

In this simulation, the vehicles are subjected to a step steer input at 1.0 sec, with a front wheel
steering angle of 2°. The vehicle dynamic responses are shown in Table 1 and Figures 4–6.

From Figures 4–6, it can be seen that since the tyres have run into saturation region ($a_{y\max} =
0.7 g$), the conventional vehicle cannot generate enough lateral forces for cornering. Its steady

Table 1. RMS of yaw rate error and CG sideslip angle.

<table>
<thead>
<tr>
<th>Vehicle configuration</th>
<th>Yaw rate error $r_d - r$</th>
<th>CG sideslip $\beta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>4WS</td>
<td>1.1158°/sec</td>
<td>0.1883°</td>
</tr>
<tr>
<td>DYC</td>
<td>1.0574°/sec</td>
<td>3.0848°</td>
</tr>
<tr>
<td>4WS+DYC</td>
<td>1.0060°/sec</td>
<td>0.1248°</td>
</tr>
<tr>
<td>4WS+DYC+ARC</td>
<td>1.0027°/sec</td>
<td>0.1315°</td>
</tr>
</tbody>
</table>
Figure 4. Yaw rate responses in step steer manoeuvre.

Figure 5. Vehicle CG sideslip angle responses in step steer manoeuvre.

Figure 6. Roll angle responses in step steer manoeuvre.

The yaw rate is only 12°/sec, while sideslip angle and body roll angle are as large as 2.6° and 2.7°, respectively.

As shown in Figure 5, individual 4WS vehicle can effectively suppress the sideslip motion within 0.5°. Since 4WS can only regulate tyre lateral forces, its longitudinal speed gradually reduces from 120 to 93 km/h, and accordingly its desired yaw rate $r_d$ increases from 13.6°/sec to 17°/sec; see Equation (8).

Although DYC controller can track the desired yaw rate through differential tyre longitudinal forces, it cannot suppress the sideslip angle well, with its RMS value up to 3.0848°, as shown in Table 1. Compared to the conventional vehicle, its lateral acceleration is relatively higher, $(a_{y\max\text{DYC}} = 0.84$ g$)$, and thus its steady CG sideslip angle is 1.4° bigger than that of conventional vehicle.
On the other hand, as for vehicles with both integrated controllers, DYC+4WS and DYC+4WS+ARC, the desired yaw and sideslip motion can be satisfactorily tracked, which shows significant improvements in vehicle stability. As can be seen from Figure 6, by adding ARC system, the body roll angle can be significantly reduced (from originally $3.2^\circ$ to $1^\circ$ in the end of simulation). Besides, since ARC can assist to control yaw rate, the RMS value of yaw rate error in DYC+4WS+ARC case is slightly smaller than that of DYC+4WS case; see also Table 1. Note that for comparison among different controllers, the corresponding elements of weighting matrices in Equation (14) are kept the same for all controllers. If these weightings are further tuned, hopefully the assistant effect of ARC on yaw control can be more remarkable.

4.2. **Closed-loop double-lane-change manoeuvre**

To further evaluate control benefits, double-lane-change manoeuvre is performed as a kind of driver-based handling test. For all vehicles with different controllers, the same steer controller [23] in CarSim® is used as the driver model. The simulation results are shown in Figures 7–9.

As can be seen in Figure 7, both the conventional and DYC vehicles show large lateral deviations (4.07 m and 3.99 m, respectively), though DYC vehicle finally settles to the target path at the end of simulation. Figure 8 also suggests that these two vehicles, particularly the
Figure 9. Body roll angle responses in double-lane-change manoeuvre.

conventional vehicle, are difficult to control for the driver model with unacceptably large steering wheel angle demands.

As shown in Figure 7, it can be seen that compared to the conventional and DYC vehicles, with the help of active steering, all the other three vehicles can follow the target path much better. Particularly, both DYC+4WS and DYC+4WS+ARC vehicles show almost zero path deviation in the changed lane, and the according driver steering demands are also quite satisfactory; see Figure 8. As for the fully integrated controller, DYC+4WS+ARC, the body roll angle can be reduced by about 0.5° as compared to that of DYC+4WS case (see Figure 9). With this reduction of roll angle, it can be said that the driver confidence can also be improved by the integrated DYC+4WS+ARC controller.

5. Conclusion

In this paper, an IVCC is proposed to coordinate steering, braking/traction and active stabiliser. The controller is realised via a main/servo-loop structure: the main-loop calculates the stabilising force $F_{ud}$ using sliding mode control, while the servo-loop optimally distributes $F_{ud}$ to each tyre, using SQP approach. Both open-loop and closed-loop simulations show that the integrated 4WS+DYC controller can effectively coordinate active steering and active wheel torque, and handling performances can be significantly improved, compared to individual 4WS or DYC controllers.

Obviously, by introducing ARC system, body roll angle can be reduced greatly by the integrated 4WS+DYC+ARC controller. Since active stabilisers can indirectly affect vehicle handling, they are also beneficial to yaw control, though the improvement is not so significant. But by further tuning the controller weightings in the cost function, the stabilisers are, hopefully, to show more remarkable effect in yaw rate tracking.

In future work, constraints of steering/braking/traction actuators will be directly considered, e.g. rate and magnitude limits of actuators can be expressed as upper and lower bounds of tyre variables. The distribution of $F_{ud}$ to actuator inputs will be formulated as a constrained optimisation problem, which will be solved by control allocation approaches, e.g. quadratic or linear programming.

Acknowledgements

The authors would like to thank the National Natural Science Foundation of China (NSFC Grant No. 50575141) for its great support.
References
