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Emulating the Lateral Dynamics of a Range of Vehicles Using a Four-Wheel-Steering Vehicle

Allan Y. Lee
California institute of Technology

Abstract

The concept of a Variable Dynamic Testbed Vehicle (VDTV) has been proposed as a tool to evaluate collision avoidance systems and to perform driving-related human factors research, among others. The goal of this study is to analytically investigate to what extent a VDTV with four-wheel-steering can emulate the lateral dynamics of a broad range of vehicle models. Using a particular mid-sized vehicle as a baseline, our study indicated that this mid-sized vehicle modelled with a closed-loop four-wheel-steering system can be controlled to emulate the lateral response characteristics of a range of vehicles, from "small" to "full-size," reasonably well over a speed range of interest. A novel steering control configuration that has the potential to improve further the "degree of emulation" has also been proposed.

Introduction

To study the correlation between vehicle response characteristics and driver commands relative to crash avoidance, the National Highway Traffic Safety Administration's Office of Crash Avoidance Research (OCAR) has at its disposal a comprehensive set of tools and facilities. These include the Vehicle Research and Test Center, and the (currently being developed) National Advanced Driving Simulator. To augment these tools and facilities, OCAR has defined its concept of a Variable Dynamics Testbed Vehicle (VDTV) [l]. This vehicle would be capable of emulating a broad range of automobile dynamic characteristics, allowing it to be used in crash avoidance systems development and in driving-related human factors research, among others. A similar but more limited experimental vehicle, called Variable Response Vehicle was developed in the 70's by the General Motors Corporation for vehicle handling research [2]. This vehicle has independent electro-hydraulic controlled front and rear steering and steering force systems, enabling it to emulate a variety of vehicle directional control characteristics.

To emulate the dynamics of a broad range of vehicles, the steering, suspension, and braking sub-systems of the VDTV must be "programmable." To accomplish this goal, "active" elements such as a fully-active suspension system, a four-wheel-steering system, and an anti-lock braking system are added to the vehicle. Software changes made to the algorithms that control these active sub-systems can then effect significant changes in the vehicle's lateral, longitudinal, and braking dynamics. The scope of the present study is limited to an analytical investigation of the extent to which a VDTV with four-wheel-steering can emulate the lateral response characteristics of a range of vehicles.

Vehicle Dynamic Model

A vehicle handling model that the author had developed, VEHDYN, is used in this study. The lateral dynamics of a vehicle are modeled in VEHDYN using the approach suggested in Ref. 3.

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This model includes both the vehicle's yaw, roll, and lateral degrees of freedom. Since the pitch degree of freedom does not significantly affect handling, it was not included in this model. Hence, the "states" of this vehicle model are: yaw rate, side-slip angle, roll rate, and roll angle.

For simplicity, VEHDYN uses a linear tire model. Lateral forces and aligning torques generated by the tires are computed as functions of the tires' slip and camber angles. This tire model also includes effects that the vehicle's roll angle has on both the camber and tire angles. Results obtained with VEHDYN are accurate up to approximately 0.3 g's of lateral acceleration. Beyond that, models which include both the tire saturation effects and suspension nonlinearities must be employed.

In this study, VEHDYN is augmented with the following steering actuator dynamic models:

$$\tau_f \dot{\delta}_f + \delta_f = \delta_{fc}, \quad (1)$$

$$\tau_r \dot{\delta}_r + \delta_r = \delta_{rc}. \quad (2)$$

Here, δ_f and δ_r are the front and rear tire angles, while δ_{fc} and δ_{rc} are commands sent to the front and rear steering actuators, respectively. The front tire command δ_{fc} is related to the driver steering wheel angle δ_{SW}: $\delta_{fc} = \delta_{SW}/N_S$, where N_S is the steering ratio. For two-wheel-steering vehicles, there is no rear tire command (i.e., $\delta_{rc} = 0$). For four-wheel-steering (4WS) vehicles, the rear tire command is determined by a control algorithm typically implemented using a microprocessor. The time constants of the front and rear steering actuators are τ_f and τ_r, respectively. In this study, the bandwidths of these actuators are both assumed to be 4 Hz.

Estimated values of vehicle parameters used in VEHDYN, for a range of passenger sedans are summarized in Table 1. In that table, the symbols S, C, M, and F are used to denote "small", "compact", "mid-sized", and "full-sized" vehicles, respectively. Parameter values in that table are estimated using data given in, among others, Ref. 4. Linearized tire parameters are estimated using data given in Ref. 5, and are summarized in Table 2.

Steady-state and Transient Characteristics of Vehicle Models

Using VEHDYN, and the estimates of vehicle and tire parameters given in Tables 1 and 2, both the steady-state and transient characteristics of the four selected vehicle models were analytically computed. Results obtained are depicted in Figure 1.

The steady-state vehicle handling qualities can be characterized by its control sensitivity. The control sensitivity of a vehicle at a given forward speed is defined by its steady-state lateral acceleration per 100 degrees of steering wheel excursion. It is also called the steering sensitivity or lateral acceleration gain of a vehicle. As depicted in Fig. 1, this gain generally increases with the vehicle's forward speed.

The transient characteristics of the vehicles are compared using the "90% rise times" and "percent overshoots" of their lateral acceleration responses. The 90% rise time is a measure of the vehicle's "speed of response" when it is subjected to a "step" steering wheel command. Since a true "step" is physically impossible, the steering command is ramped to its steady-state value in, say, 0.15 seconds. The 90% rise time is then defined as the time it takes the vehicle's lateral acceleration to reach 90% of its steady-state value, measured from the time the steering command reaches 50% of its steady-state value. The percent overshoot is related to the amount of "damping" in the vehicle's transient responses. It is defined as the peak acceleration measured from the steady-state acceleration level, as a percent of that steady-state value. As depicted in Fig. 1, the 90% rise time decreases while the percent overshoot increases with increasing forward speed. That is, a vehicle has less lateral stability at a high forward speed.

The lateral dynamics of a vehicle can also be measured using frequency-domain performance metrics. The vehicle's yaw rate bandwidth is the frequency at which the magnitude of the transfer function, from the steering wheel to the vehicle's yaw rate, has dropped below 70.7% of its steady-state value. As depicted in Fig. 1, vehicle bandwidth generally increases with increasing vehicle speed.

As expected, both the steady-state and transient characteristics of the four selected vehicle models differ from one another. From Fig. 1, we note that Vehicle-M has relatively better handling qualities: shorter rise time and smaller percent overshoot in its acceleration responses, together with a larger yaw rate bandwidth. Accordingly, it is selected as our "baseline" vehicle. The main objective of our study is to analytically investigate how well Vehicle-M with a four-wheel-steering system can emulate the lateral response characteristics of other vehicle models that are either "smaller" or "larger" than itself.

Four-Wheel-Steering Control Algorithms

The lateral dynamics of a vehicle can be substantially altered by steering its rear wheels in conjunction with those at the front. For example, the control sensitivity of a four-wheel-steering vehicle at a given forward speed can be increased/decreased by steering the rear wheels out-of-phase/in-phase (respectively) with the front wheels. The closed-loop 4WS control algorithm used in this study has the following structure:

$$\delta_{rc} = K_\delta(U)\,(\frac{1+\tau_1 s}{1+\tau_2 s})\delta_{fc} - K_r(U)r\ . \qquad (3)$$

Here, the variables δ_{fc} and r are the front steering command and the filtered yaw rate of the vehicle, respectively. The speed-dependent feed-forward gain $K_\delta(U)$ can approximated by: $K_\delta(U) = a_0 + a_1 U + a_2 U^2$. Here, U is the vehicle's forward speed, which varies from 80 to 170 km/h (about 50 to 100 mph). Similarly, the speed-dependent feedback gain $K_r(U)$ is given by: $K_r(U) = b_0 + b_1 U + b_2 U^2$. For simplicity, the parameters τ_1 and τ_2 of the lead-lag compensator are not function of the vehicle speed. The feed-forward gain $K_\delta(U)$ is used mainly to alter the vehicle's steady-state responses while the lead-lag compensator is used to vary the vehicle's transient characteristics. The feedback gain $K_r(U)$ affects both the steady-state and transient characteristics of the vehicle. Appropriately selected, these control parameters will allow us to vary the lateral dynamics of Vehicle-M so as to approximate those of Vehicle-S, Vehicle-C, and Vehicle-F. Other controller designs could also be used [6].

The addition of a rear steering actuator (with mass m_R and at a distance l_R behind the rear axle) produces the following effects on the vehicle: the total mass of the vehicle is increased to $M + m_R$, where M is the total mass of the "nominal" vehicle (cf. Table 1). Also, the nominal location of the vehicle's c.g. is shifted rearward by Δ:

$$\frac{\Delta}{L} = (\frac{m_R/M}{1+m_R/M})(\frac{b+l_R}{L}), \qquad (4)$$

where the parameters a, b, and L are defined in Table 1. The new vehicle's c.g. is now located at $(\bar{a}, \bar{b})$ where $\bar{a} = a + \Delta$, and $\bar{b} = b - \Delta$. The increased yaw moment of inertia of the sprung mass about the new vehicle's c.g. is: $I_{zz} + m_R(l_R+\bar{b})^2 + M\Delta^2$. The added weight also altered the loadings on the front and rear tires. Both the cornering stiffnesses (C_α, in N/deg) and camber stiffnesses (C_γ, in N/deg) of the tires change with their respective loadings (F_Z, in kg. wt.) [5]:

$$C_\alpha = -51.97 + 4.536F_Z - 0.00465F_Z^2\ , \qquad (5)$$

$$C_\gamma = 0.217F_Z - 0.00022F_Z^2\ . \qquad (6)$$

Assuming $m_R = 15$ kg.wt. and $l_R = 0$ meters, all these effects have been taken into account in our study.

Results and Discussions

Via a trial-and-error process, control parameters, including a_i and b_i, $i = 0,...,2$, as well as τ_1 and τ_2 are iteratively determined so that the lateral response characteristics of the 4WS-controlled Vehicle-M closely approximate those of Vehicle-S, Vehicle-C, and Vehicle-F, over a speed range of interest. Results obtained are given in Table 3. Alternatively, an optimization algorithm such as that described in Ref. 7 could be used to determine these control parameters in a more systematic way. This is a topic for future research.

Graphical comparisons between the lateral characteristics of Vehicle-S and the 4WScontrolled Vehicle-M are depicted in Fig. 2. Those for Vehicle-C and Vehicle-F are given in Figs. 3 and 4, respectively. Mean emulation errors in the vehicle's control sensitivity, 90% rise time, percent overshoot, and yaw rate bandwidth, over the speed range of interest, are summarized in Table 4. These results indicate that a 4WS-controlled Vehicle-M can emulate the lateral response characteristics of Vehicle-S rather well. On the other hand, it is rather difficult for the 4WS Vehicle-M to simultaneously approximate both the steady-state and transient characteristics of Vehicle-C very well. The same is true with Vehicle-F. The mean emulation errors associated with the 90% rise times for these vehicles are high.

The levels of emulation can potentially be improved if the VDTV has both the four-wheel-steering and steer-by-wire features depicted in Fig. 5. With this steering configuration, the controller architecture given in (3) can be modified to:

$$\delta_{fc} = \delta_{SW}/N_S - G_f r \,, \qquad (7)$$

$$\delta_{fr} = G_{ff}\delta_{SW} - G_r r \,. \qquad (8)$$

Here, the variable r is the filtered yaw rate of the vehicle. Other measured vehicle quantities such as lateral acceleration could also be used instead of the yaw rate. The term G_{ff} is a feed-forward controller similar to the lead-lag compensator design used in (3), while G_f and G_r are feedback controllers for both the front and rear controllers, respectively. The "feedforward" and "feedback" components of (7-8) are similar, respectively, to the "Command Augmentation System" and "Stability Augmentation System" proposed in Ref. 6. The added emulation benefits that one can derive with this controller architecture are to be confirmed in future study. But experimental results obtained from a vehicle fitted with a similar steering system have shown promises [8].

Yet another approach is to add "dummy" weight (or weights) to the above described 4WS vehicle. Dummy weights can be placed in the trunk of the vehicle but we must ensure that structural integrity of the vehicle chassis is not compromised. Both the masses and the longitudinal locations of these dummy weights are now additional "design parameters" that we can vary to achieve better emulation results. Results obtained with this approach, not given here, are encouraging.

Concluding Remarks

The concept of a Variable Dynamic Testbed Vehicle has been proposed as a tool to evaluate collision avoidance systems, to perform driving-related human factors research, among others. The goal of this study was to analytically investigate to what extent a VDTV with four-wheel-steering can emulate the lateral dynamics of a broad range of vehicle models. Using a mid-sized vehicle as a baseline, our study indicated that the "base-line" vehicle fitted with a closed-loop four-wheel-steering control system can emulate the lateral response characteristics of a "small" vehicle, over a speed range of interest, very well. The degrees to which a "compact" and a "full-size"vehicle can be emulated by the four-wheel-steering baseline vehicle are poorer. The "level of emulation" can potentially be improved if the VDTV has both the four-wheel-steering and steer-by-wire features depicted in Fig. 5. The added "emulation" benefits that one can derive with such a steering system must be confirmed in future study.

Results given in this paper were obtained using "linear" vehicle and tire models. However, collision avoidance maneuvers made by drivers are typically "high-g," and are at the performance limits of vehicle and tires. To assess the "emulabilty" of various vehicle models by an actively controlled base-line vehicle in these maneuvers, "nonlinear" vehicle models (see, for example, Ref. 9) must be employed. This is a difficult but important topic for future study.

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Disclaimer

The discussion in the text of this paper reflects the opinion and findings of the author, and not necessarily those of the reviewers or the National Highway Traffic Safety Adminstration.

Table 1 Estimated Values of Vehicle Parameters

Vehicle Parameters	Vehicle S	Vehicle C	Vehicle M	Vehicle F
Class	small	compact	mid-size	full-size
Wheel base [L] (m)	2.39	2.62	2.69	2.90
Track width (m)	1.40	1.40	1.55	1.58
Weight [M] (kg.wt.)	1007	1262	1419	1750
Weight front (%)	65.3	64.0	64.6	57.0
c.g. distance from front axle [a] (m)	0.83	0.94	0.95	1.22
c.g. height (m)	0.51	0.54	0.56	0.57
Roll inertia (kg-m^2)	328	431	573	717
Pitch inertia (kg-m^2)	1535	2032	2553	3848
Yaw inertia [I_{zz}] (kg-m^2)	1545	2082	2687	3907
Front tire excursion (deg)	$\pm$32.8	$\pm$31.7	$\pm$26.5	$\pm$30.0
Front/Rear roll stiffnesses (Nm/deg)	684 490	828 381	1206 935	865 294
Front and Rear roll damping (Nm-sec/deg)	42.7	53.5	60.1	74.2
Steering ratio (-)	17.0	17.6	17.0	18.0

Table 2 Estimated Tires Characteristics

Vehicle	Vehicle S	Vehicle C	Vehicle M	Vehicle F
Tire	P185 /60R14	P185 /75R14	P205 /65R15	P215 /70R15
Front/Rear Loading (kg.wt.)	658 349	808 454	917 502	998 752
Front/Rear Cornering Stiffness (N/deg, per wheel)	633.2 433.4	704.7 509.2	1051.0 794.0	1092.4 954.3
Front/Rear Aligning Torque Stiffness (Nm/deg, per wheel)	11.77 6.26	14.46 8.13	16.41 8.99	17.86 13.47
Front/Rear Camber Stiffness (N/deg, per wheel)	20.95 9.60	27.43 13.16	54.50 41.03	87.98 71.69
Front/Rear Aligning Torque per Camber (Nm/deg, per wheel)	1.18 0.63	1.45 0.81	1.64 0.90	1.79 1.35

Table 3 Selected Values of Control Parameters

Control Parameters	Unit	Vehicle S	Vehicle C	Vehicle F
$a_0 \times 10$	–	0.522	-0.856	-2.153
$a_1 \times 10^3$	$(km/h)^{-1}$	-1.613	3.520	6.524
$a_2 \times 10^5$	$(km/h)^{-2}$	0.543	-1.026	-2.097
$b_0 \times 10$	sec	0.760	1.867	2.217
$b_1 \times 10^3$	sec $(km/h)^{-1}$	0.000	0.350	0.554
$b_2 \times 10^5$	sec $(km/h)^{-2}$	0.000	-0.250	-0.416
T_1	sec	.-0.27	+0.21	+0.20
T_2 x 10	sec	0.48	0.29	4.0

Table 4 Mean Emulation Errors

Mean Emulation Errors	Vehicle S	Vehicle C	Vehicle F
control sensitivity (%)	2.96	2.38	5.06
90% rise time (%)	0.49	15.81	10.05
percent overshoot (%)	1.92	1.52	2.46
yaw rate-based BW (%)	2.12	2.39	1.37

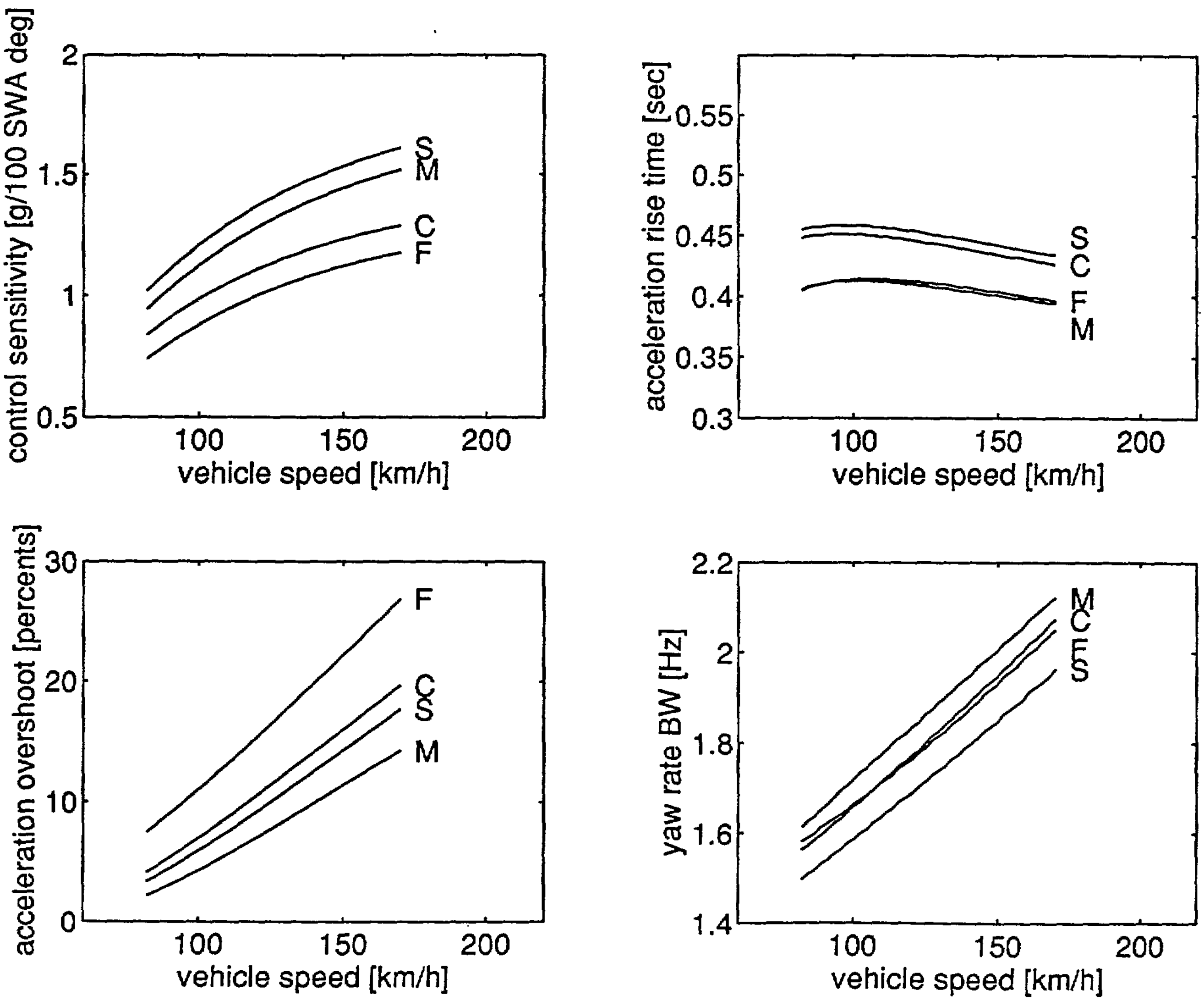

Fig. 1 Lateral characteristics of four vehicle models

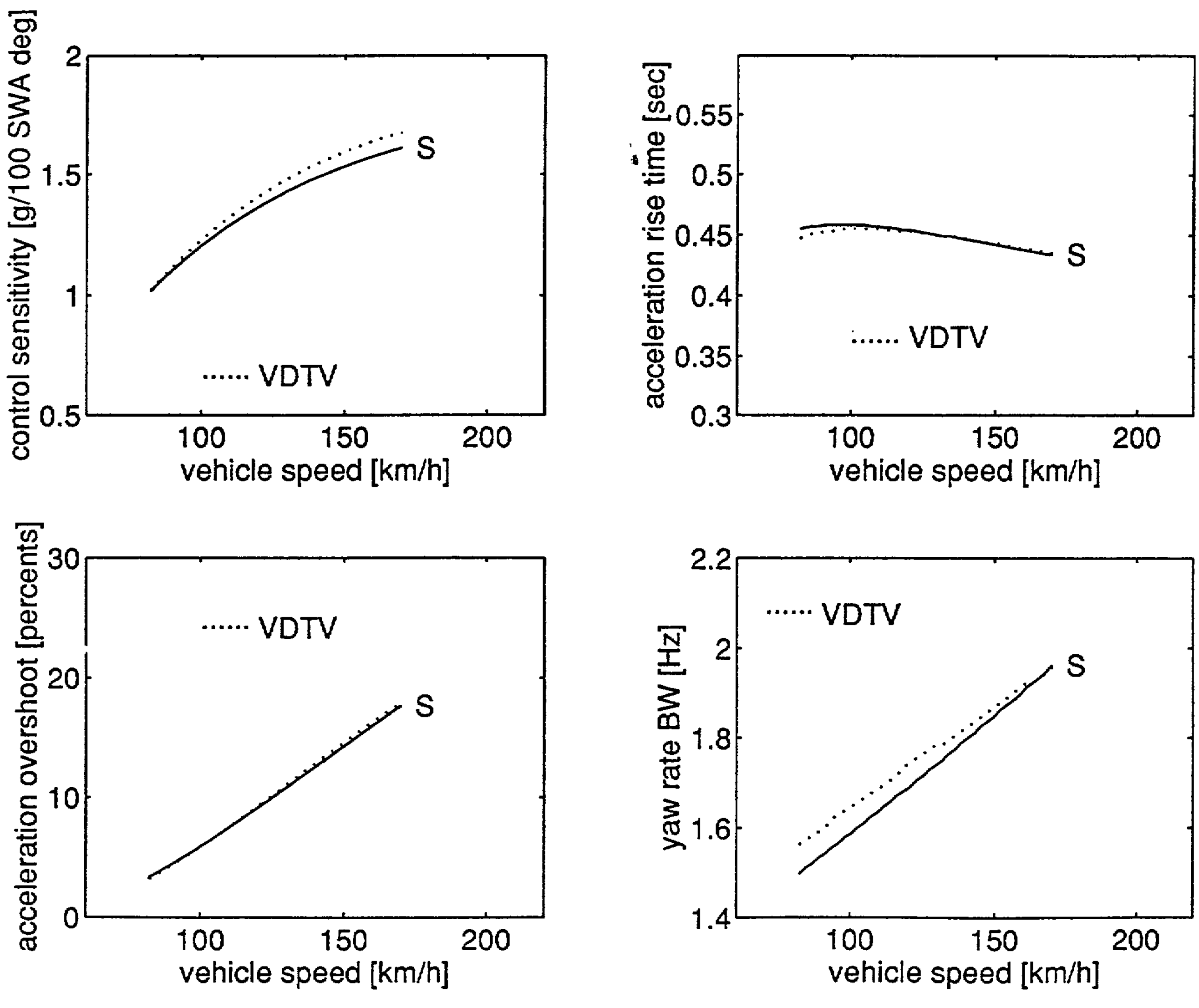

Fig. 2 Emulating the lateral characteristics of Vehicle-S using a VDTV

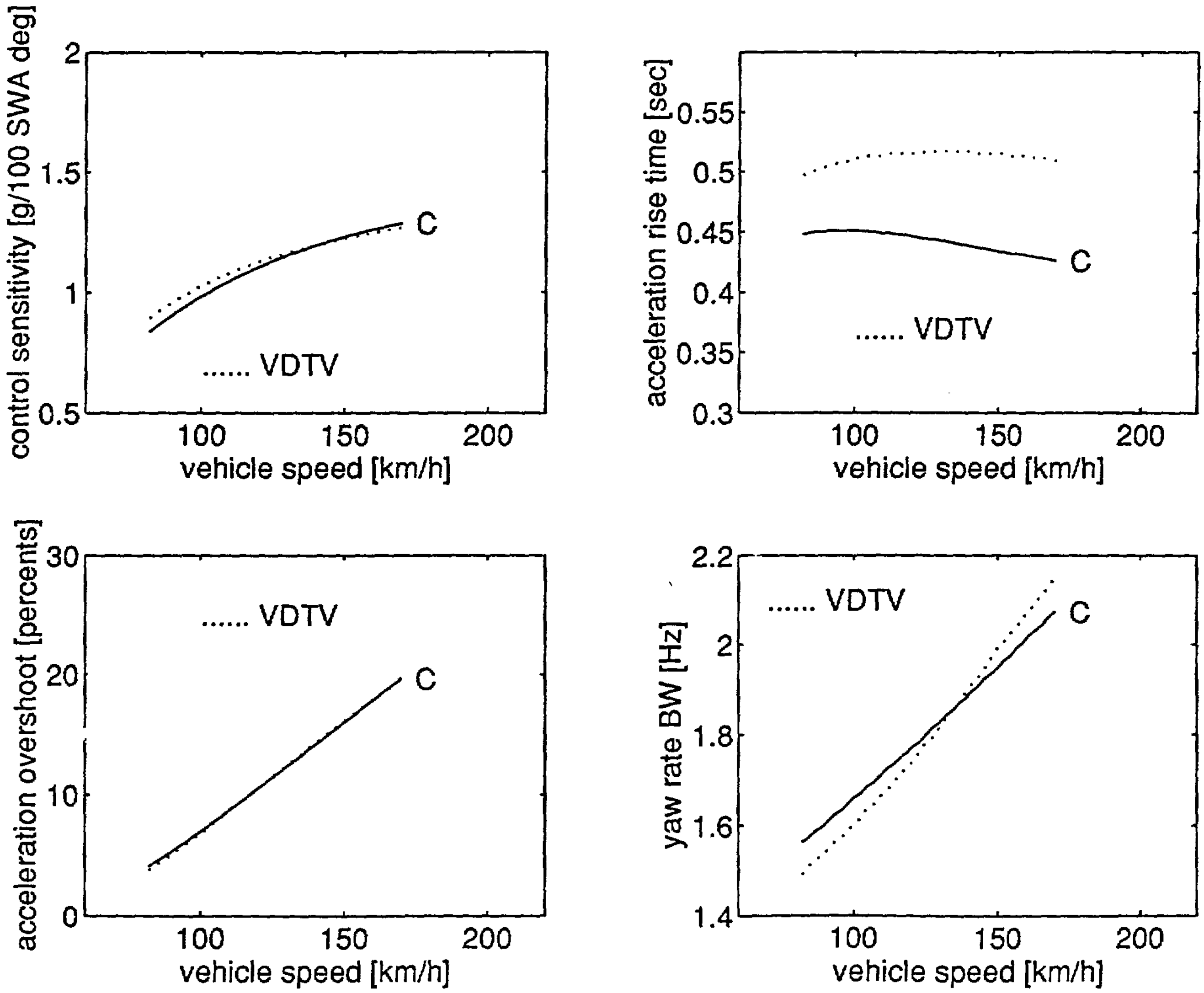

Fig. 3 Emulating the lateral characteristics of Vehicle-C using a VDTV

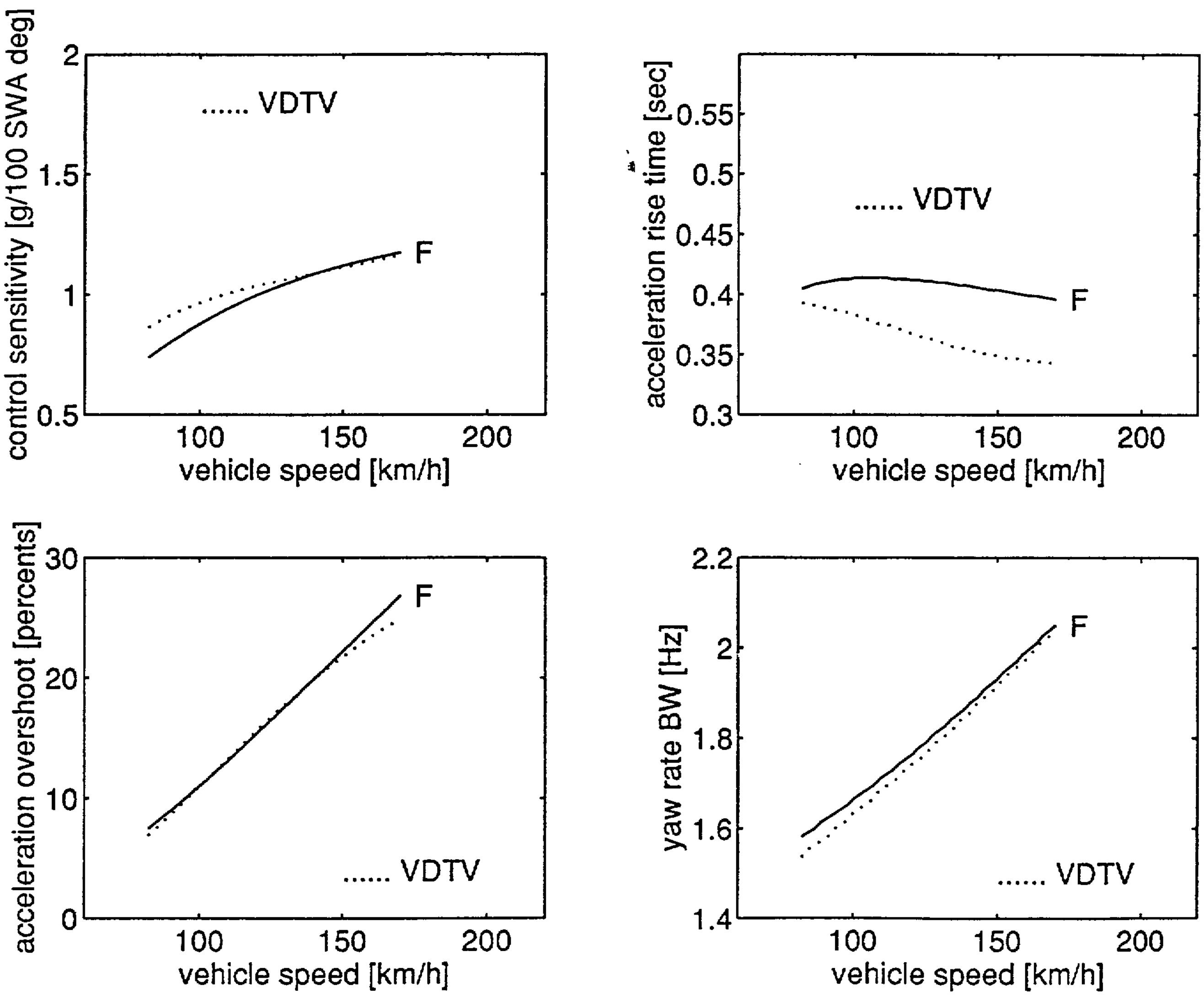

Fig. 4 Emulating the lateral characteristics of Vehicle-F using a VDTV

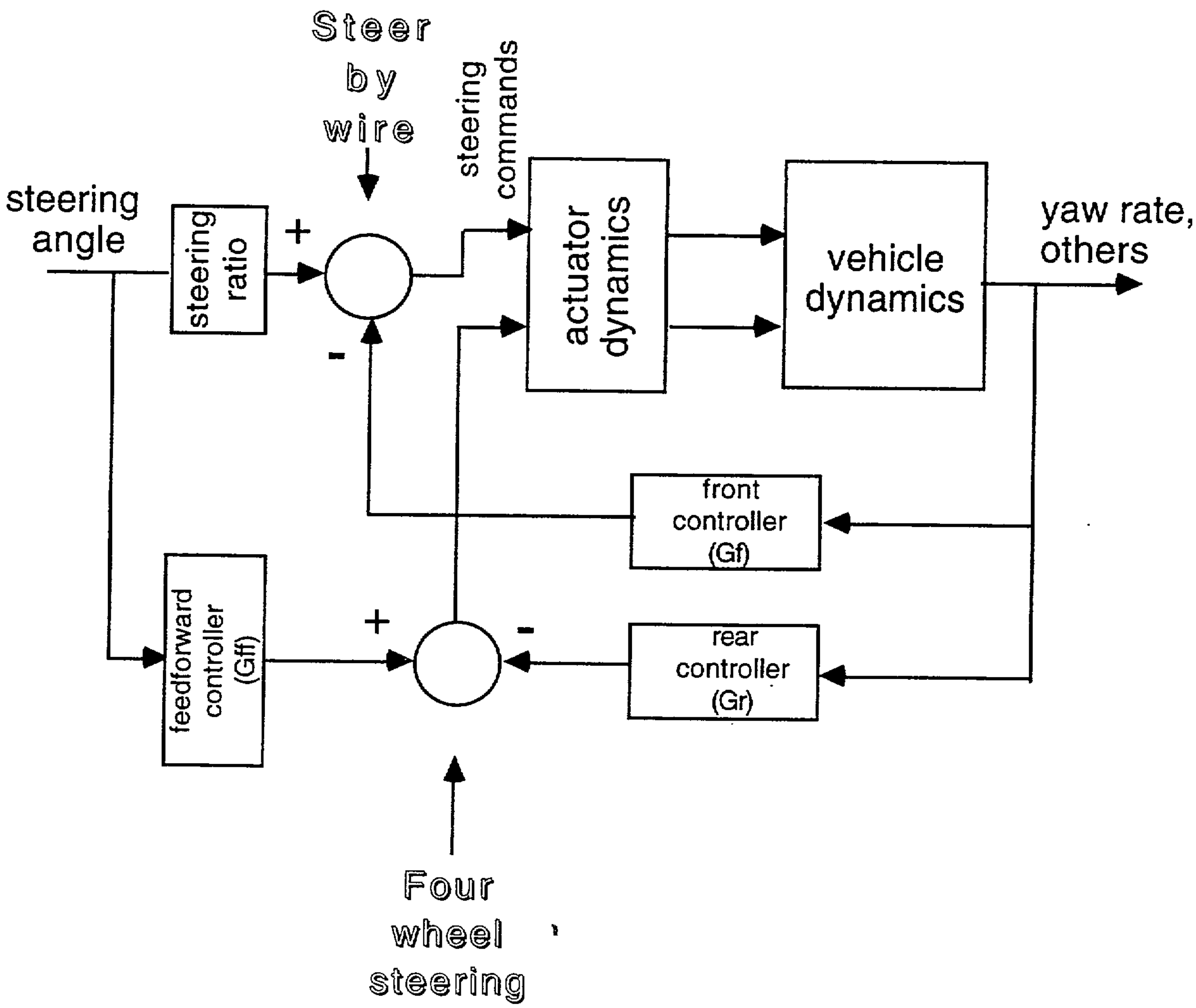

Fig. 5 A VDTV with steer-by-wire and four-wheel-steering