Computer model predictions of the directional response and stability of driver vehicle systems during anti-skid braking

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SYNOPSIS Findings specific to this work and applicable to moderate turning maneuvers during which full brake torque applications occur are: (1) passenger car antiskid systems operating under high friction conditions are effective in assisting drivers maintain directional and path control during full braking demand, only if, a) the rear axle is equipped with a select low ("worst wheel") axle control system, and/or, b) conservative prediction rules are employed in the antiskid logic, particularly at the rear axle location; and (2) under moderate friction conditions (most wet asphalt or concrete pavements), antiskid systems appear to have greater effectiveness in assisting drivers maintain directional and path control of vehicles during limit braking maneuvers. Since the principal mechanism responsible for severe alteration of the directional dynamics of the vehicle during hard braking is the fore/aft load transfer, the level of deceleration and the ability of drivers to adapt quickly to an altered dynamical system, together determine the level of closed-loop braking performance that can be achieved.

NOTATION

- S_p is the wheel slip set point for antiskid prediction
- S_r is the wheel slip set point for antiskid reselection
- T_p is the driver model preview time
- t_L is the driver model transport time delay
- g is the gravitational constant
- LF is left front
- RF is right front
- LR is left rear
- RR is right rear

1 INTRODUCTION

This paper examines the basic issue of driver-vehicle directional control performance during antiskid braking. A comprehensive model of the automobile and a well documented driver steering control model are used to simulate and study the effects of antiskid braking upon the directional control and stability of the combined driver-vehicle system. A general purpose antiskid model, utilized in several prior studies of antiskid braking performance, is employed to represent and simulate basic antiskid system characteristics. Antiskid braking along a circular curve serves as the reference maneuver. Computer model data which provide geometric and dynamical characteristics typical of a current vintage passenger car are used to represent the subject vehicle. Parameters for describing the nominal driver steering control characteristics are selected from previously published data and experience obtained from non-braking driver-vehicle experimental research. Passenger car antiskid systems defined by axle and/or individual wheel control strategies are examined as the primary form of system variation influencing driver-vehicle directional control performance. Other topics examined in the paper include: a) the effects of different driver characteristics (e.g. driver preview time and transport lags) on total system directional stability, b) the influence of varying fundamental vehicle descriptors (tire cornering force characteristics, static load distribution, and inertial properties) on closed-loop directional performance, and c) analysis of cyclical or varying antiskid braking and the predicted effect it has in modifying driver steering control.

2 RELATED WORK

Previous research and studies of driver-vehicle braking performance under conditions of antiskid cycling are not abundant. The few studies that do appear in the literature however are relatively recent and do report findings of both analytical and experimental results. Grunow, Heißing, and Otto (1) examined in a rather comprehensive manner various test procedures for passenger cars with antiskid braking systems. Full scale vehicle tests as well as computer simulation were used to evaluate four proposed test procedures. Grunow, Heißing, and Otto concluded that passenger car antiskid systems can change decisively a vehicle's handling quality during braking. Their evaluation included criteria for steerability, roadholding ability, yaw stability, and deceleration performance. A final recommendation proposed two test procedures: 1) "braking during steady turning with additional steering input," and 2) "straight line braking on a split friction surface."

Mitschke (2) examined, through computer simulation and a driver model, various fore/aft arrangements of antiskid systems while braking on split friction surfaces. Mitschke concluded that the use of rear axle "select low" antiskid systems demonstrated clear superiority over other arrangements insofar as minimizing course deviations and corrective driver steering activity. Furthermore, Mitschke found that use of no antiskid system or an entirely independent wheel control antiskid system rendered the vehicle uncontrollable by the simulated driver.

Satoh and Shiraishi (3) conducted a series of full scale vehicle tests to evaluate a proposed antiskid system especially for front wheel drive passenger cars. Special consideration was given to road surface roughness and its role in exciting antiskid system cycling. Straight line stopping tests were conducted on surfaces with split friction as well as differential roughness. Braking performance was also evaluated during steady turning tests and braking during obstacle avoidance maneuvers. Satoh and Shiraishi concluded that their proposed system, which is characterized by front axle "select high" and rear axle "select low" control logic, offers the best compromise and overall performance for the test procedures used.

Finally, Bisimis (4) in a strictly analytical work argues that it is not sufficient to use antiskid systems simply for prevention of wheel lock itself in order to guarantee directional stability. Bisimis notes that fore/aft wheel slip ratios and their accompanying cornering coefficients are of prime importance in determining whether or not a vehicle will be controllable by drivers during antiskid braking. Correct adaptation of antiskid systems to specific vehicles is also recommended.

In addition to the above references, no substantive work appears in the literature regarding both driver steering and braking control for combined braking and steering maneuvers, particularly at elevated levels of longitudinal deceleration. The recent work of Newcomb (5), which examined the topic of closed-loop braking control by drivers under low and moderate deceleration conditions, should contribute to further understanding of how drivers function not only during braking, but during combined braking and steering maneuvers as well. In relation to the above citations, the work presented in this paper likewise does not address the larger question of combined (simultaneous) braking and steering control by drivers insofar as the presence of an antiskid braking system is presumed to remove from the driver the additional burden of longitudinal control.

3 DESCRIPTION OF MODEL

The overall model employed in this study consists of: 1) a vehicle dynamics model, previously used for representing commercial vehicles, but recently modified to include passenger car representations, 2) a general-purpose antiskid brake system model, and 3) a driver steering control model. The vehicle model by MacAdam et al. (6) is quite comprehensive having six degrees of freedom for the sprung mass, bounce and roll (for beam axles) degrees of freedom for each unsprung mass, wheel rotational degrees of freedom, and steering system compliance. The model used for representing tire force generation within the vehicle model is described in Dugoff, Fancher, and Segel (7). The antiskid system model, while capable of representing relatively complex system characteristics (e.g. Fancher and MacAdam (8), MacAdam (9)), was used in this application to represent simple wheel slip switching conditions and brake pressure-torque characteristics. The driver model by MacAdam (10) is based upon an optimal preview control strategy which calculates steering control values at each instant of time so as to minimize the previewed vehicle-path error. This basic scheme is saddled with a presumed human operator transport lag which reflects the inherent human operator limitation for instantaneous reaction. The driver model has been compared with experimental driver-vehicle results in both the frequency domain and the time domain and shown to be capable of accurately representing driver steering control actions in numerous non-braking applications. The driver model predictions presented in this paper for antiskid braking maneuvers are simply stated, -- predictions. No opportunity has been presented since the development of the model to permit comparison with experimental findings and validation under heavy braking conditions.

4 BASELINE SYSTEM DESCRIPTION

4.1 Baseline Vehicle

The passenger car parameters describing the geometric and inertial details are shown in Table 1 and reflect certain nominal characteristics of current front wheel drive vehicles as seen for example in Riede, Leffert, and Cobb (11). The baseline loading and tire cornering stiffnesses contribute 0.029 radians/g of understeer whereas the steering system compliance adds a small additional amount, bringing the total vehicle understeer to a level of approximately 0.035 radians/g . The brake torque is fixed at a fore/aft proportioning distribution of approximately 65% - 35% .

Table 1 Baseline Vehicle Description

Parameter Description	Value
Wheelbase	2.74 (m)
Front Axle Load	6672 (N)
Rear Axle Load	4448 (N)
Front Tire Cornering Stiffness	38232 (N/rad)
Rear Tire Cornering Stiffness	38232 (N/rad)
Roll Compliance	0.131 (rad/g)
Maximum Front Brake Torque	1017 (N-m)
Maximum Rear Brake Torque	565 (N-m)
Height of sprung mass c.g.	0.635 (m)
Understeer level	0.035 (rad/g)

4.2 Antiskid System Description

The antiskid system behavior to be described was intentionally designed to operate in a simple manner in order to facilitate interpretation of the calculated results. Since the focus here is not on detailed and specific antiskid system characteristics, but rather, on the effect that representative antiskid cycling and wheel slip excursions may have on influencing driver-vehicle directional stability during hard braking, a simplified antiskid control logic is employed. Prediction, or the brake release condition, occurs if the wheel slip becomes greater than some set point, S_p. Reselection, or the brake application condition, occurs if the wheel slip falls below a set point, S_r. Different combinations of S_p values (0.1, 0.2) and S_r values (0.1, 0.2) were studied, the most common being $S_p = 0.2$ and $S_r = 0.1$. Brake torque increases and decreases occur linearly with time. Full brake torque is achievable in 0.5 seconds from zero torque during an application cycle; zero brake torque is achievable from a maximum torque condition in 0.16 seconds. These rates are comparable to those provided by Schurr and Dittner (12) for typical European vehicles.

4.3 Nominal Driver Characteristics

The driver model employs two basic parameters. A driver preview time, T_p , is used to represent "look ahead" capabilities employed by drivers. The driver transport lag, t_L , represents human operator reaction-time limitations. These two parameters in conjunction with an internal model of the basic vehicle dynamics determine the closed-loop steering behavior predicted by the model. Preview time and transport lag values found to be characteristic of most driver-vehicle systems under non-braking conditions lie in the range of 1.0 - 3.0 seconds and 0.2 - 0.3 seconds, respectively. The computer runs performed in this paper used T_p values of either 1.0 or 1.5 seconds and a t_L value of 0.25 seconds to represent the simulated driver.

5 DESCRIPTION OF BASELINE MANEUVER / CONDITIONS

The reference maneuver performed with the driver-vehicle simulation is "braking in a turn." The defined path is a circle of 305 meter radius. Prior to entering the circular curve, the vehicle is started on a 15 meter straight segment to initialize the simulation model. At a time of 3.5 seconds into the maneuver, during steady turning, the demanded brake torque (pedal pressure) is applied and reaches its maximum level at 4.0 seconds. The demanded brake torque is held at its maximum level throughout the remainder of the maneuver. Intervention of the antiskid system to modulate brake torques at individual wheel locations occurs within this time span. The initial vehicle speed for all simulation runs seen here is 26.8 m/sec. High tire/road friction conditions as well as moderate tire/road friction conditions were studied. The high friction condition is defined as having a peak longitudinal tire/road friction coefficient of 1.0 and a slide (locked wheel) value of 0.9 . The moderate friction condition is defined by a peak friction coefficient of 0.6 and a slide value of 0.5 . Both friction conditions achieve peak friction values at a wheel slip ratio of 0.2 .

6 RESULTS

Reference will be made to several example computer runs throughout this section which illustrate various results predicted by the computer simulation study. Table 2 shows nine different configurations of the same baseline vehicle differing in terms of antiskid system characteristics or driver characteristics. Each configuration is identified by a case number followed by a brief description of the fore/aft antiskid arrangement. Independent wheel control, worst wheel axle control (select low), and best wheel axle control (select high) systems were studied and are shown abbreviated in the description column. The next two columns define the antiskid prediction and reselection set points for wheel slip used in each case. Columns 5 and 6 list the driver preview time and transport time delay values. The last column summarizes the tire/road friction condition.

6.1 Results for High Friction Conditions

Cases 1 to 4 seen in Table 2 represent selected runs which exhibited directionally stable (or marginally stable) driver-vehicle systems under the conditions of high friction defined above. By and large, unless the vehicle was equipped with a worst wheel antiskid configuration on its rear axle and/or employed very low values of prediction set point (S_p) at the rear, the driver/vehicle (closed-loop) system was predicted to be unstable for this level of tire/road friction and deceleration demand. Figure 1 shows an unstable system

Table 2 Definition of Cases 1 - 9

Case	Antiskid Description	S_p	S_r	T_p	t_L	Tire-Road Friction
1	Indep-front and rear	0.2	0.1	1.0	0.0	1.0 peak; 0.9 slide
2	Ind-frt; Worst-rear	0.2	0.1	1.0	0.25	1.0 peak; 0.9 slide
3	Worst-front and rear	0.2	0.1	1.0	0.25	1.0 peak; 0.9 slide
4	Ind-frt; Worst-rear	0.1	0.1	1.0	0.25	1.0 peak; 0.9 slide
5	Indep-front and rear	0.2	0.1	1.0	0.25	0.6 peak; 0.5 slide
6	Ind-frt; Worst-rear	0.2	0.1	1.0	0.25	0.6 peak; 0.5 slide
7	Ind-frt; Worst-rear	0.2	0.1	1.5	0.25	0.6 peak; 0.5 slide
8	Best-frt; Worst-rear	0.2	0.1	1.0	0.25	0.6 peak; 0.5 slide
9	Ind-frt; Worst-rear	0.2	0.1	1.0	0.0	0.6 peak; 0.5 slide

response and corresponding time histories of driver steering wheel displacement (limit steering wheel stops are placed at 4 radians), vehicle sideslip, and yaw rate produced during an antiskid braking run. System stability is arbitrarily defined here by the requirement that peak vehicle sideslip levels do not exceed 0.25 radians. In general, most of the high friction runs which reached the defined sideslip condition (0.25 radians) continued to increase and eventually spun out of control. The unstable response seen in Figure 1 corresponds to the same system configuration seen in Case 2 (Table 2) except for the substitution of an independent wheel control system on the rear axle. All runs performed for the high friction condition with no antiskid brake system, but with active driver control, resulted in vehicle trajectories spiraling to the inside of the curve and complete loss of directional stability.

A representative example of a stable closed-loop system response under high friction conditions is seen in Figure 2. The results appearing in Figure 2 correspond to Case 4 which is described in Table 2 as having an independent wheel control system on the front axle, a worst wheel control system on the rear axle, conservative antiskid prediction (S_p = 0.1), and nominal driver characteristics (T_p = 1.0, t_L = 0.25). Note that the vehicle sideslip remains within 0.07 radians of zero throughout the maneuver. At the start of the braking application, the driver steering response is to first countersteer to correct for the initial loss of rear lateral tire force (due to load transfer and increased wheel slip). A correction phase then occurs which stabilizes the vehicle directional response in a slow periodic manner. The brake torque and wheel slip time histories are seen in the second portion of Figure 2. Because of the large fore/aft load transfer due to braking, the heavily loaded front wheels exhibit little antiskid cycling, whereas their counterparts on the rear axle display continuous cycling at a frequency of about 2 Hz.

The importance of conservative values for antiskid prediction set points (S_p) is illustrated by the system response seen in Figure 3. The system configuration seen here corresponds to Case 2 and differs from Case 4 (Figure 2) only in the value of prediction set point (0.2 here, 0.1 for Case 4). The increased rear wheel slip conditions, deriving from the more aggressive wheel slip set point, produces significantly greater vehicle sideslip response and steering control requirements (seen saturating in the corrective phase).

Computer runs performed to examine the influence of modest variations in fore/aft mass center location and tire cornering stiffnesses were largely uneventful. Sensitivities of directional control to such variations were not readily observable because of (a) driver model compensation and (b) large fore/aft load transfer effects due to braking which tend to dominate the system response.

Lastly, Figure 4 presents a summary of results corresonding to cases 1 to 4 for the high friction condition. The intent here is to summarize in a simple manner four different performance measures for the driver/vehicle system relating to (a) directional stability,(b) driver steering control requirements, (c) deceleration performance, and (d) roadholding ability. Performance measures (a) - (d) correspond respectively to the plots of (a) Peak Sideslip, (b) Peak Steer, (c) Average Deceleration, and (d) Path Error. Peak values of vehicle sideslip or steering wheel angle which occurred during the braking maneuver are those recorded. Average deceleration values seen here were calculated from 1 second following full brake torque demand (time = 5 seconds) to the end of the stop. Path error values correspond to the approximate vehicle-path error at the end of the stop (positive values imply vehicle offsets to the inside of the curve).

The matter of driver response time and its importance in permitting drivers to stabilize a vehicle under heavy braking conditions is clearly seen in Figure 4. Case 1 is identical to the unstable system configuration appearing in the time histories of Figure 1 - except for an ideal value of zero seconds for the driver transport lag (t_L). The observed sensitivity of results to a 0.25 second time lag difference in driver steering response is noteworthy. This result would suggest far greater dependence of closed-loop braking performance upon time delay characteristics of human operators than is usually predicted or observed for handling performance.

Comparison of Cases 2 and 4 in Figure 4, in which aggressiveness of the rear axle worst-wheel antiskid system is the only difference, indicates that the less aggressive system (Case 4) suffers slightly in deceleration performance but benefits significantly in terms of reduced sideslip demand and improved roadholding.

6.2 Results for Moderate Friction Conditions

A selection of system configurations for the moderate friction condition is seen in Table 2 (Cases 5-9). In general, this friction condition was less apt to produce an unstable closed-loop system response when compared with the high friction condition and the same antiskid configuration. The primary reason for this observation relates to the reduced level of fore/aft load transfer occurring under lower friction conditions and the diminished effect upon the vehicle directional dynamics due to braking. A representative stable system response is seen in the set of time histories of Figure 5 for Case 6. (The response of the same configuration for the high friction condition is given in Figure 3 and also Case 2 of Figure 4.) The vehicle sideslip and driver steering responses of Figure 5 display similar characteristics seen in the stable high friction cases. The principal difference here lies in the reduced sensitivity of directional stability to antiskid prediction criteria (S_p values) for the moderate friction condition. The brake torque and wheel slip time histories seen in Figure 5 also exhibit greater cycling activity, particularly on the front axle where the reduced tire/road friction coupling is no longer capable of accommodating the maximum brake torque demands.

Figure 6 illustrates the response produced for Case 8 which corresponds to a best wheel control system on the front axle and a worst wheel system on the rear axle. Comparing the steering wheel time history of Figure 6 with its independent system (front axle) counterpart seen in Figure 5, a modest advantage accrues to the best/worst configuration in terms of a reduced driver steering control requirement. Similarly, the deceleration performance of the best/worst configuration is improved while suffering only a small increase in peak value of vehicle sideslip.

Full deceleration runs performed with no antiskid system, but with active driver control, resulted in complete loss of directional stability with all wheels locked, as in the case of the high friction condition. However, the vehicle trajectory, instead of spiraling to the inside of the curve as observed in the high friction cases, drifted outward and away from the circular path.

Finally, Figure 7 summarizes, in the same manner as seen in Section 6.1, the braking performance measures corresponding to Cases 5-9 for the moderate friction condition. Case 5, the all wheel independent system, exhibits poorer directional control properties and requires greater steering control inputs by the driver, when compared with Cases 6-9 which utilize worst wheel systems at the rear axle. Comparison of Cases 6 and 9 indicate a much lower sensitivity to driver transport lag variations for this friction condition than was seen for the high friction condition. Also variations in driver preview time, Cases 6 and 7, suggest only a modest influence as reflected primarily in driver steering control requirements.

7 CONCLUSIONS

The computer simulation results presented in this paper largely support observations and findings of other researchers who have previously investigated different aspects of closed-loop directional stability of passenger cars during antiskid braking. Findings specific to this work and applicable to moderate turning maneuvers during which full brake torque applications occur are:

* Passenger car antiskid systems operating under high friction conditions (most dry asphalt or concrete pavements) are effective in assisting drivers maintain directional and path control during full braking demand, only if, a) the rear axle is equipped with a select low ("worst wheel") axle control system, and/or, b) conservative prediction rules are employed in the antiskid logic, particularly at the rear axle location. Aggressive antiskid control logic, even on rear axle worst wheel systems, should be avoided. Independent wheel control is recommended over worst wheel systems on the front axle in order to improve deceleration performance. For the braking maneuvers studied here, observable changes in directional stability accruing from modest variations in basic vehicle properties tended to be dominated, and for the most part masked, by large fore/aft load transfer effects produced during heavy deceleration and by driver control compensation.

* Under moderate friction conditions (most wet asphalt or concrete pavements), passenger car antiskid systems appear to effectively assist drivers in maintaining directional and path control of vehicles during limit braking maneuvers. Furthermore, certain closed-loop braking performance measures, as described in Section 6.1, exhibit lower sensitivity to

antiskid braking system variations under conditions of moderate friction than under conditions of high friction.

Finally, the principal mechanism responsible for severe alteration of the directional dynamics of the vehicle during hard braking is the fore/aft load transfer. As such, the level of deceleration and the ability of drivers to adapt quickly to an altered dynamical system, together determine the level of closed-loop braking performance that can be achieved. Accordingly, it is recommended that experimental tests be encouraged to help address, in a systematic fashion, questions pertaining to driver-vehicle interactions during braking and combined braking and steering maneuvers.

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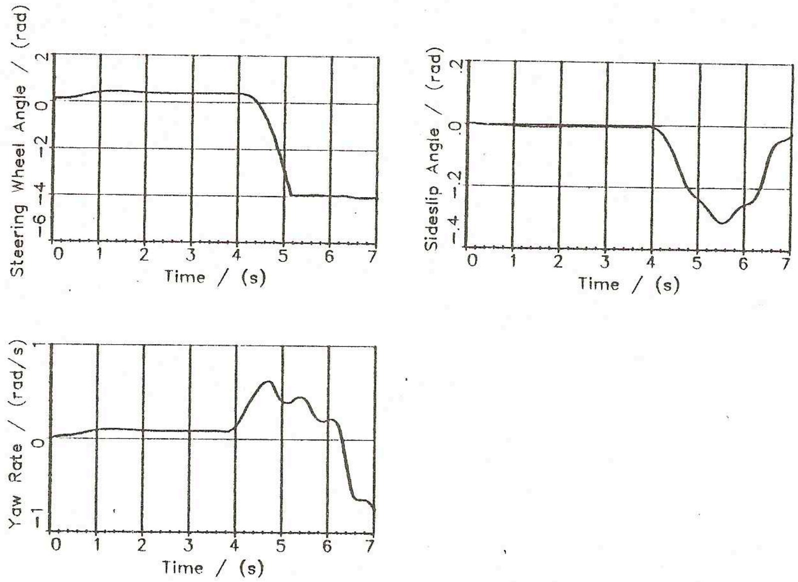

Fig 1   Unstable system response — high friction condition

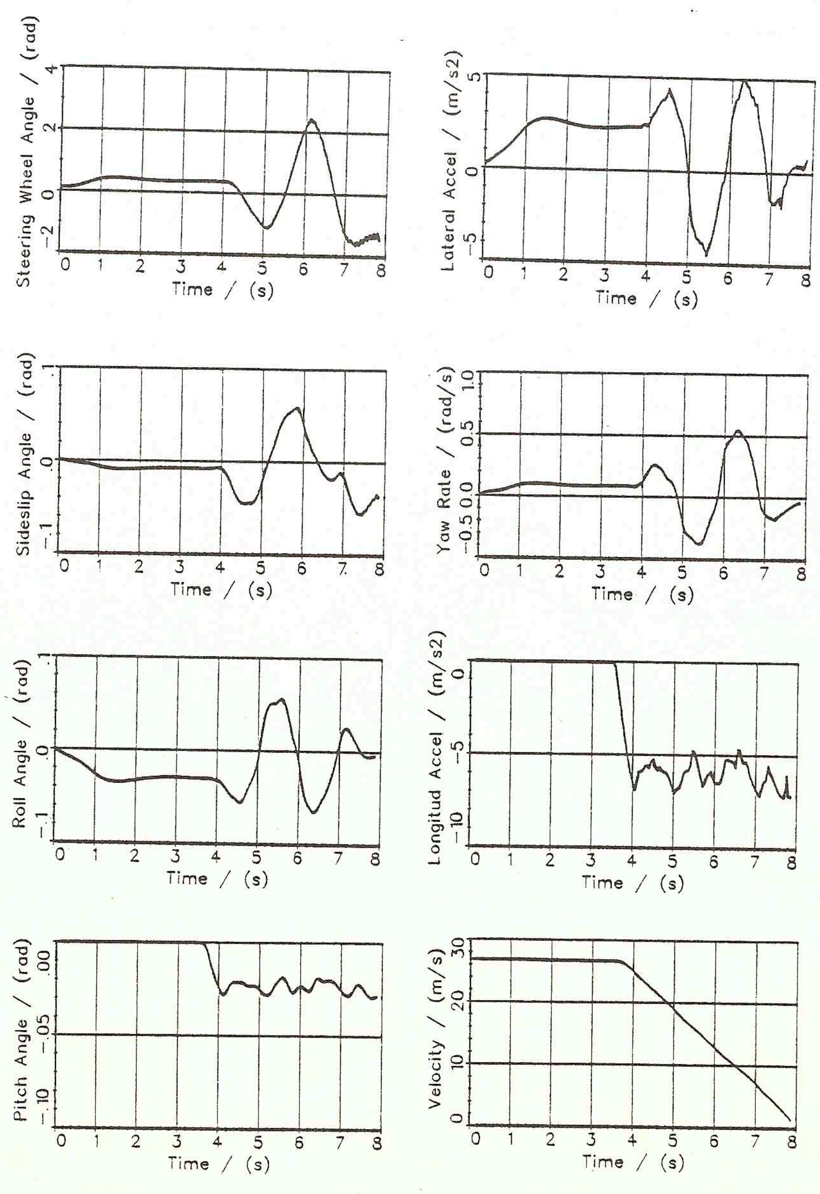

Fig 2a   Stable system response — high friction condition

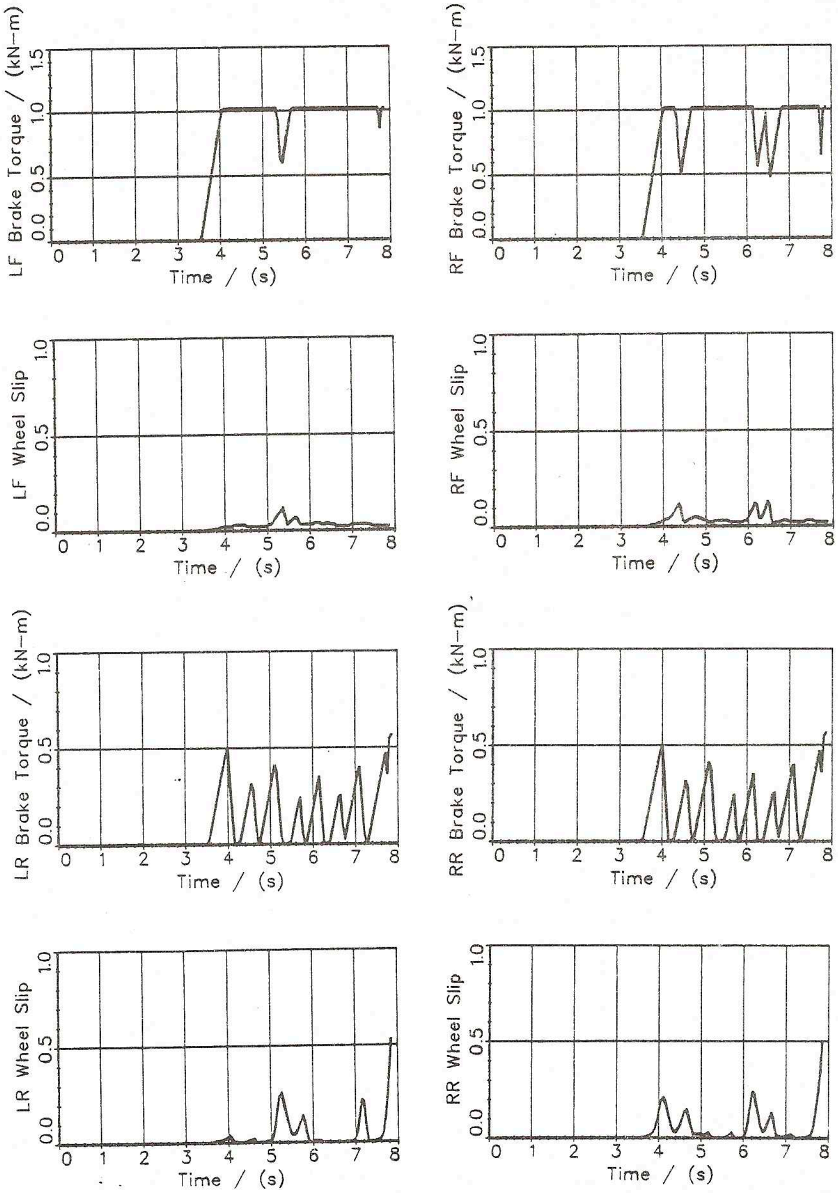

Fig 2b Stable system response — high friction condition

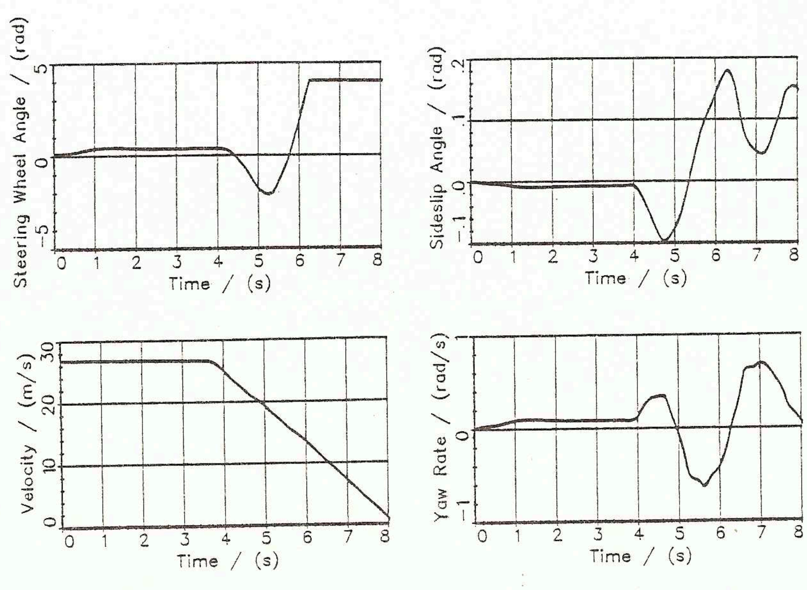

Fig 3 Rear anti-skid system prediction value S_p = 0.2 (case 2)

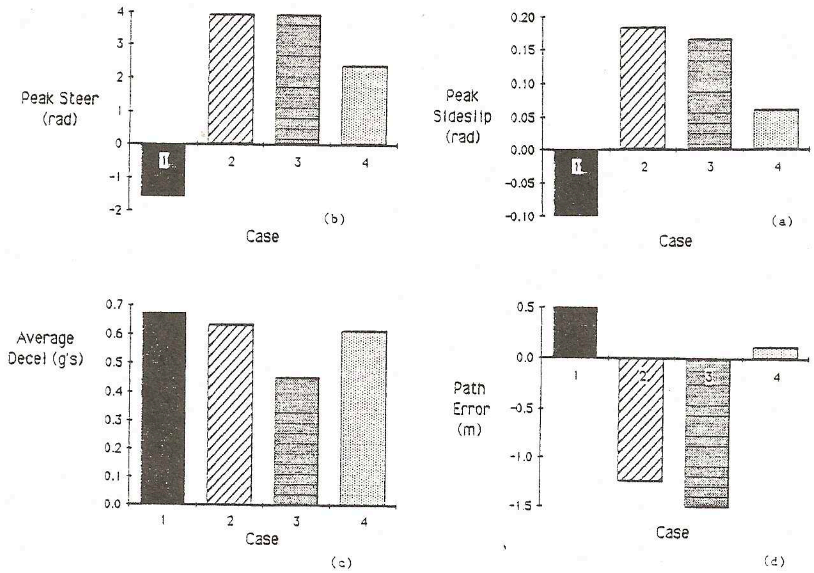

Fig 4 Summary of selected braking performance measures – high friction condition

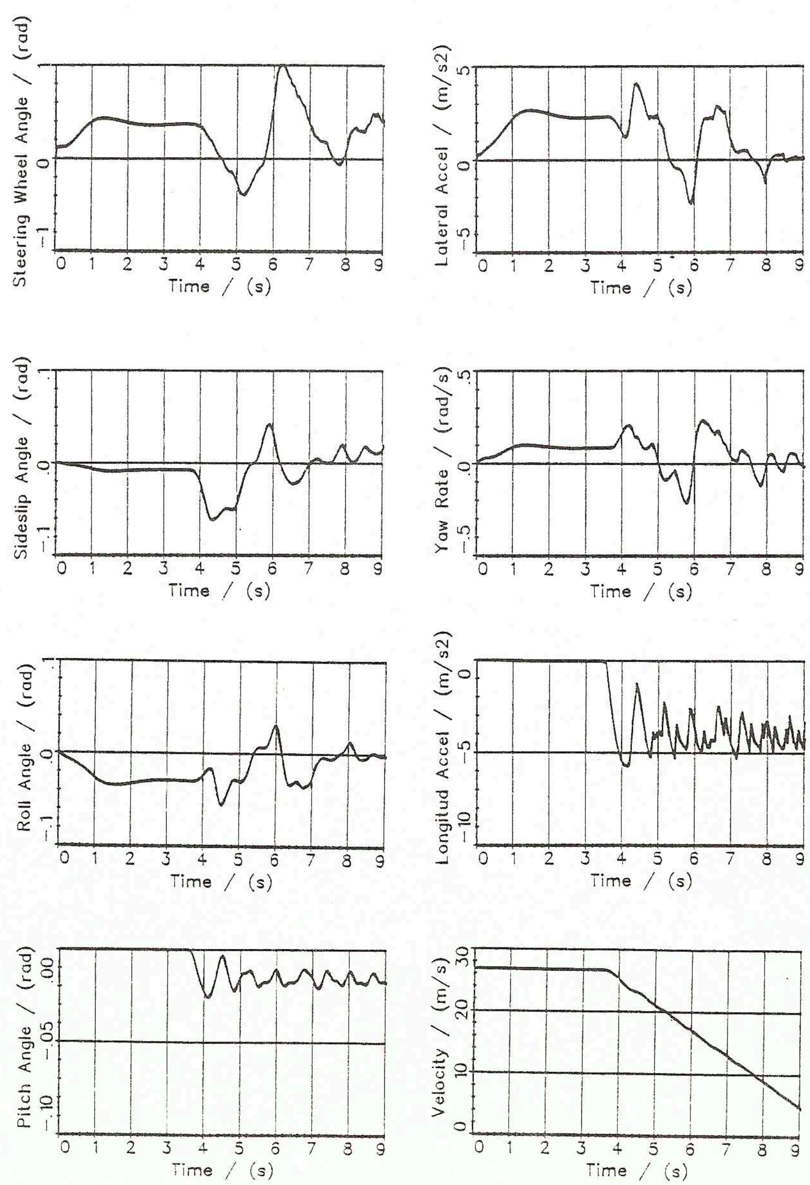

Fig 5a Stable system response – moderate friction condition (case 6)

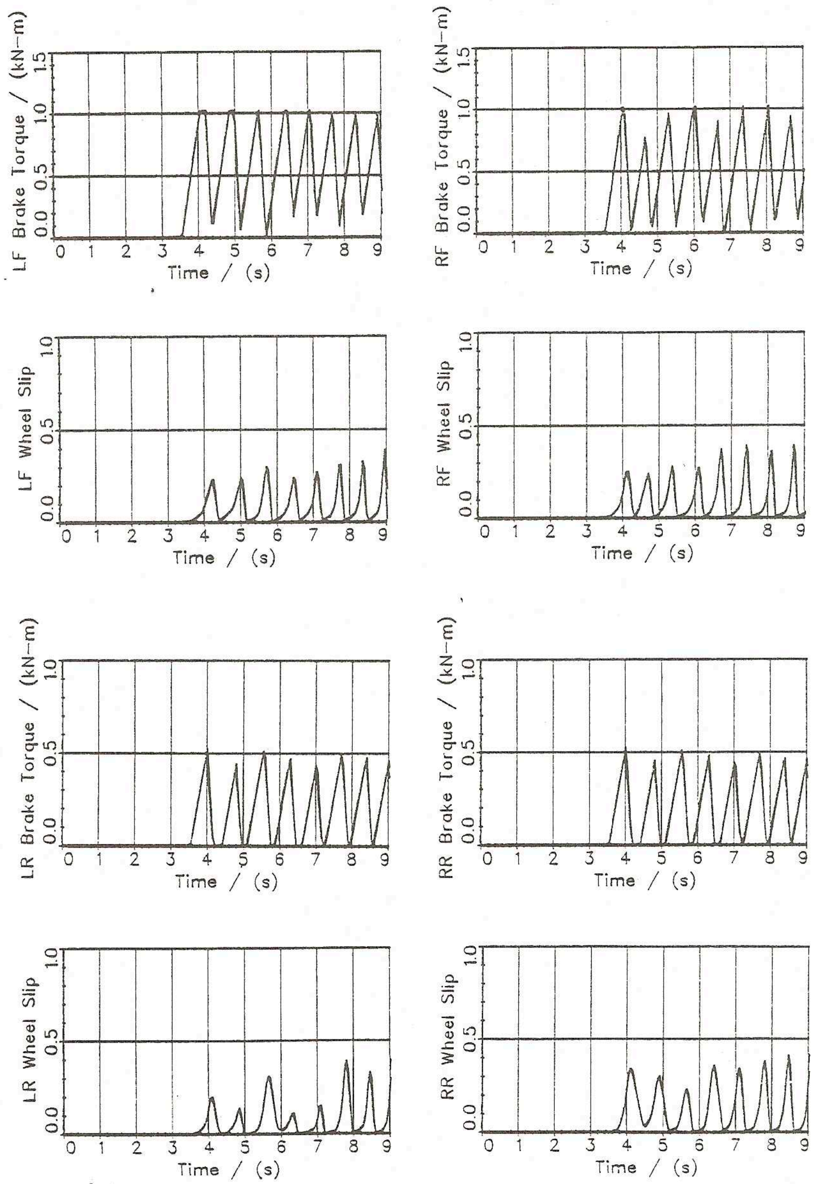

Fig 5b Stable system response — moderate friction condition (case 6)

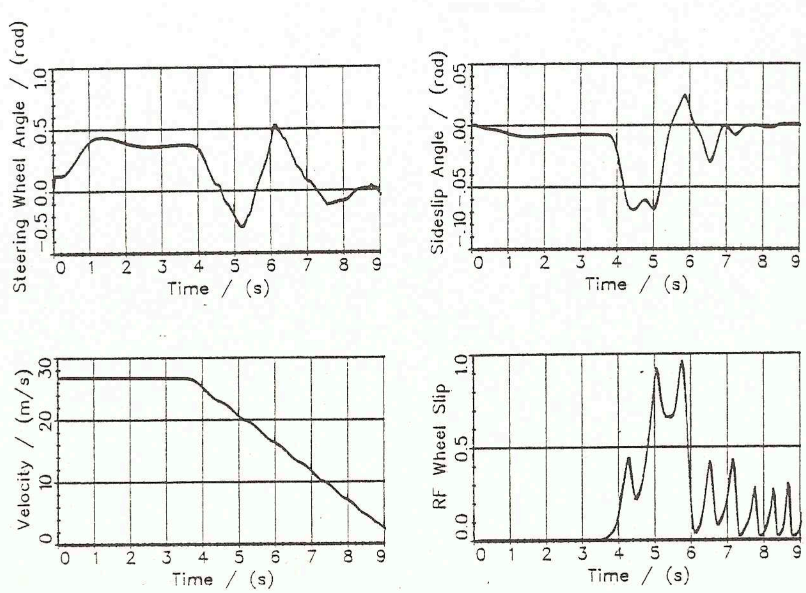

Fig 6 Best wheel/worst wheel result (case 8)

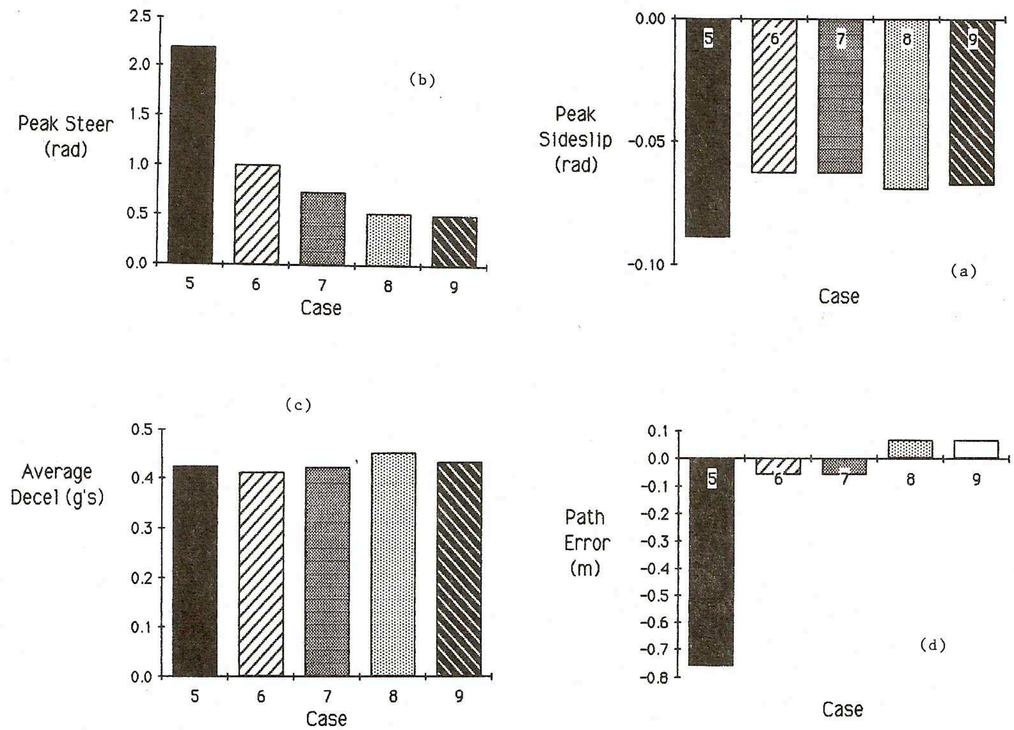

Fig 7 Summary of selected braking performance measures — moderate friction condition

ANTI-LOCK BRAKING SYSTEMS FOR ROAD VEHICLES

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anti-lock braking systems for road vehicles

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The ever-increasing volume of traffic on the roads requires a high level of operational safety in all types of vehicle. Reliable braking is fundamental to safety, but braking in certain circumstances can cause the wheels to lock, with consequent loss of control and the possibility of accident and injury. Anti-lock devices have been available for some time and they are now fitted to many commercial vehicles as standard. There is also an increasing demand for anti-lock systems to be fitted to private cars and motorcycles. In the light of the current interest in this aspect of braking, a conference on the subject was organized by the Institution of Mechanical Engineers in September 1985. More than twenty papers were presented and they are now published in this volume. The topics discussed covered the design and performance of anti-lock systems, their integration with other vehicle systems, testing and instrumentation, reliability and relevant legislation.

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