COMPUTER SIMULATION AND PARAMETER SENSITIVITY STUDY OF A COMMERCIAL VEHICLE DURING ANTISKID BRAKING

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SUMMARY

A large-scale digital computer model, used for simulating the dynamical response of commercial vehicles, was employed in a parameter sensitivity study which focused on the antiskid braking performance of a heavy truck. Full-scale vehicle test results and laboratory measurements were used in developing a baseline computer representation of the test vehicle. Subsequent to the development of the baseline computer representation, the computer simulation was exercised to study the influence of various vehicle and antiskid system parameters. Basic categories examined in the parameter sensitivity study were: brake system properties, vehicle mass and inertia properties, tire traction characteristics, suspension properties, wheel dynamics, and antiskid system characteristics.

The antiskid braking performance of the examined baseline vehicle was primarily dominated by the adverse effects deriving from rear brake torque imbalances and hysteresis precipitated during wheel cycling. The most noteworthy findings of the parameter sensitivity study for improving the baseline stopping performance, outside of reducing the side-to-side imbalances and rear brake hysteresis, were found to be: (a) reduction in rear brake effectiveness or use of load proportioning valves, (b) modification of antiskid system operation, and (c) modification of rear suspension properties.

1. INTRODUCTION

Promulgation of Federal Motor Vehicle Safety Standard FMVSS 121 in the U.S. has spawned considerable discussion and controversy over the past decade regarding antiskid braking performance of commercial vehicles. Largely out of need to objectively address its own concerns over the impending government regulation, the truck industry members of the Motor Vehicle Manufacturers Association (MVMA) initiated a long-term motor truck braking and handling performance study at the Highway Safety Research Institute (HSRI) of The University of Michigan in 1971. The purpose of this study was to develop computer-based methods, principally time-domain dynamical simulations, for representing and predicting the braking and handling performance of heavy trucks and tractor-trailers. Since that time, a comprehensive set of computer models has evolved under MVMA sponsorship [1-3]. Examples of previous studies which have employed these models are given in References [4 and 5]. The material presented in this paper represents a recent study performed at HSRI to (1) simulate the baseline dynamical response of a specific problem vehicle during straight-line, antiskid braking and (2) perform a subsequent parameter sensitivity study using the baseline vehicle computer representation.

The antiskid braking study discussed here focused on the acquisition and examination of test data obtained from both full-scale vehicle tests and laboratory measurements. The intent of this effort was to gather parameter information for constructing a representative digital simulation of the test vehicle. Upon development of the representative baseline vehicle performance, the digital simulation would then be exercised to study the influence and importance of various vehicle and antiskid system parameters.

2. COMPUTER MODEL

The straight truck vehicle model employed in this study is known as the Phase III MVMA/HSRI braking model [3]. It can be characterized as essentially a pitch-plane "bicycle" model providing pitch and bounce degrees of freedom for the sprung mass, and vertical degrees of freedom for each of the front and rear unsprung masses. Each wheel is allowed its own rotational degree of freedom. Separate side-to-side wheel rotation degrees of freedom are provided for representing the effects due to brake torque imbalances on each axle.

The basic dynamical model is augmented by a set of specialized subprogram models used to represent single- and tandem-axle suspensions [1, 3, 6], antiskid braking systems [3, 7, 8], mechanical friction brake characteristics [1, 3], and longitudinal tire force properties [1, 3]. These subprogram models provide a convenient and flexible means for simulating either simple or more complex operating characteristics exhibited by these individual components.

The block diagram of Figure 1 shows the basic inter-relationships between the major components of the vehicle simulation.

3. TEST VEHICLE/MEASUREMENTS

The test vehicle used in this study was a loaded, 51,000-lb straight truck equipped with a beam axle, single-leaf front suspension, rear "walking-beam" tandem suspension, and an axle-by-axle antiskid brake control system. The front axle was equipped with dual-wedge brakes, each rear axle with S-cam brakes. The center-of-gravity height above ground was 64 in. for the loaded vehicle configuration. The payload weight of 26,000 lbs was positioned 23 in. in front of the center line of the rear suspension.

Full-scale vehicle tests were performed by White Motors, Inc., at the Bendix Automotive Proving Ground. The tests were straight-line braking stops from 60 mph using a full treadle valve application (100 psi). The primary data variables measured in each test are listed in Table 1. "Torque wheels" were used on the right front wheel and right rear-tandem wheel to measure the instantaneous wheel torques.

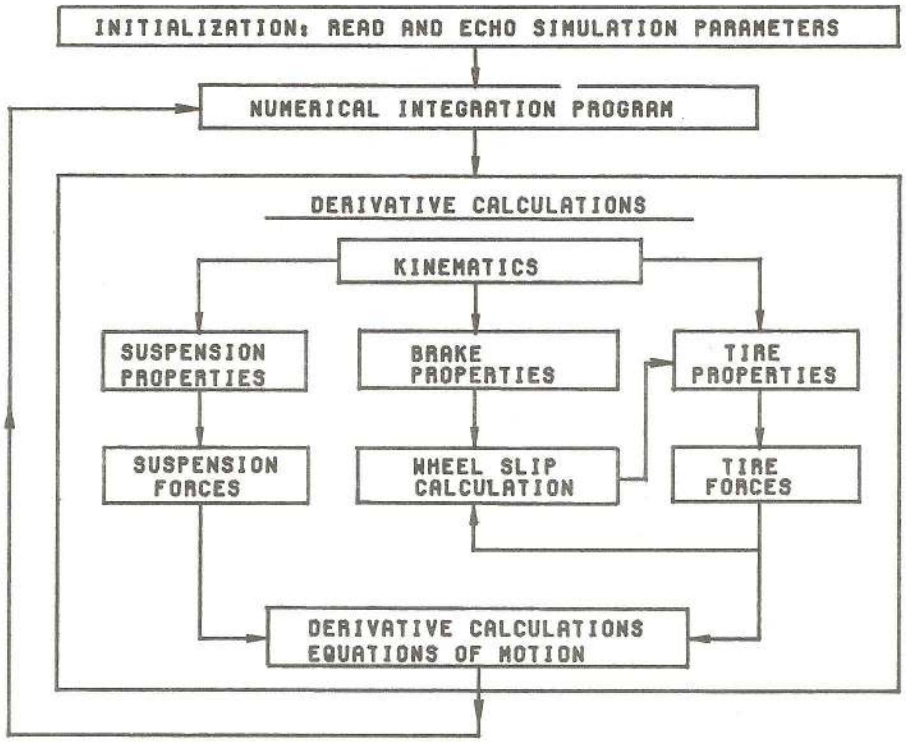

Fig. 1. Vehicle simulation block diagram.

Following the series of full-scale braking tests, the vehicle inertial properties and component system characteristics were measured by HSRI. Table 2 lists the principal vehicle components and their sources of measurement.

The tire force measurements were performed on the same surface used for the full-scale vehicle tests. Data was collected at three speeds (20, 40, and 60 mph) and three vertical loads (rated load $\pm$ 50%). Brake dynamometer data provided initial estimates for the torque effectiveness of each brake. Subsequent examination of the full-scale test records for the front and rear torque wheels revealed significant side-to-side brake imbalances and considerable levels of hysteresis in the rear S-cam actuated brakes. Suspension measurements showed a particularly complex force-deflection relationship for the rear walking-beam suspension. The laboratory analog computer tests of the antiskid system and the full-scale vehicle test records both indicated significant differences in the cycling operation of the front and rear antiskid systems. The antiskid system used on the vehicle employed "worst wheel" (slower running) side-to-side control for each axle and a "pneumatic logic" mechanism within the modulator valve to control the rate of pressure build-up during each cycle.

Table 1. Listing of Data Variables Measured During Full-Scale Vehicle Tests.

Left Front Wheel Speed
Right Front Wheel Speed
Left Middle Wheel Speed
Right Middle Wheel Speed
Left Rear Wheel Speed
Right Rear Wheel Speed
Right Front Wheel Torque
Right Rear Wheel Torque
Front Axle Vertical Position
Rear Axle Vertical Position
Front Axle Brake Pressure
Middle Axle Brake Pressure
Rear Axle Brake Pressure
Driver Treadle Pressure
Vehicle Deceleration
Fifth Wheel Vehicle Velocity
Stopping Distance

Table 2. Sources of Measurement for the Vehicle Parameters and Components.

Tire Force Properties:	Mobile Truck Tire Dynamometer (HSRI)
Brake System:	Brake Dynamometer Data and Full-Scale Vehicle Test Records
Vehicle Mass and Inertia Properties:	Inertial Swing Facility (HSRI)
Suspension Characteristics:	Suspension Measurement Facility (HSRI)
Antiskid System:	Laboratory Analog Computer Tests (HSRI) and Full-Scale Vehicle Test Records

4. COMPUTER MODEL REFINEMENTS

Review of the full-scale vehicle and component tests suggested the need for certain refinements in the brake and suspension subprogram models. Several preliminary computer runs were performed to substantiate and highlight major differences between vehicle test data and the initial computer representation. Rear brake release times which occurred during vehicle tests were much greater than those predicted by the initial simulation runs. Vehicle tests showed average brake release times of approximately 200 milliseconds; simulation release times were less than 100 milliseconds. These differences were reflected in stopping distances—250 feet for the initial computer representation, 300 feet (average) for the vehicle tests. The principal items requiring improved representation within the computer model were found to be:

•Brake pressure-torque hysteresis of the rear S-cam brakes

•Air system transport time lag effects at low pressure levels during antiskid cycling

and •Improved representation of the rear tandem suspension force-deflection characteristics.

Study of the brake pressure, torque wheel, and wheel speed traces from the vehicle tests indicated significant levels of hysteresis (27,000 in-lb) in the rear S-cam brakes during antiskid cycling. A typical hysteresis loop is shown in Figure 2. (No hysteresis was

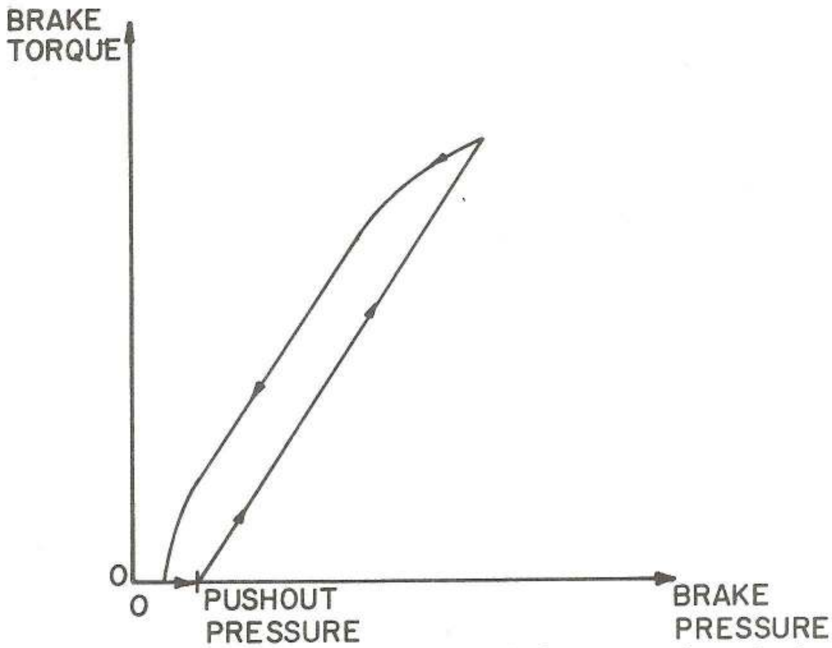

Fig. 2. Typical torque-pressure hysteresis loop; rear S-cam brake.

observed in the front wedge brakes.) The amount of hysteresis present in the rear brakes acted in combination with side-to-side wheel imbalances (caused by asymmetric vehicle loading and/or side-to-side differences in brake effectiveness) to produce significantly greater brake release times than would occur otherwise. Furthermore, the lengthening of brake release times by the hysteresis and imbalance allowed rear brake pressures to decrease to relatively low levels, often below the brake pushout pressure. Hence, additional lags were incurred from transport time delays due to volumetric changes in the air supply system for line pressures less than brake pushout. Such transport time lags in the air supply system were approximately proportional to how far the line pressure dropped during each cycle below the pushout pressure of the brake.

The manner in which the rear brake hysteresis, air supply delays, and side-to-side imbalance all combined to produce a significant stopping performance sensitivity is detailed by the following discussion and reference to Figure 3. During an antiskid pressure

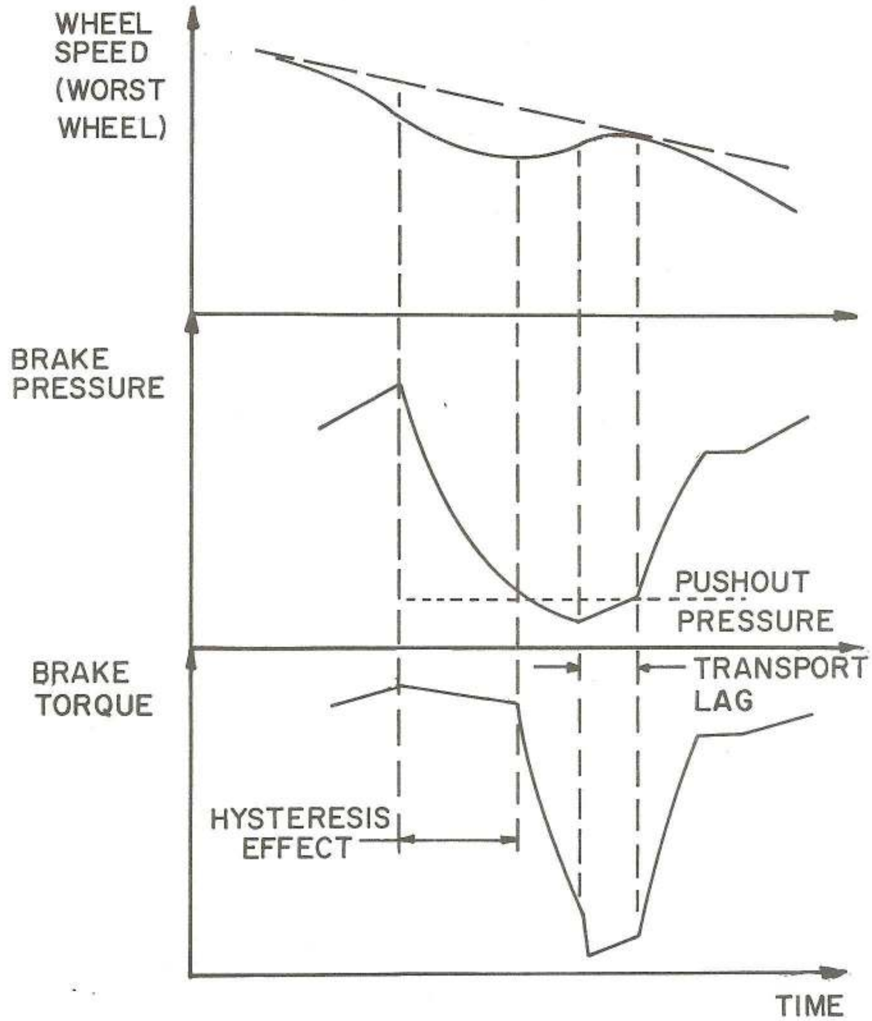

Fig. 3. Representative cycle showing effects of brake hysteresis and air supply transport lag.

application, the "worst" wheel (more effective brake and/or smaller side-to-side vertical load) on an axle is driven toward the peak of the tire-road μ-slip curve with the "best" wheel lagging behind as a result of the side-to-side imbalance. When the antiskid system interrupts the pressure application, the "worst" wheel is generally in the vicinity of the tire force μ-slip peak (15-25% slip). As the brake pressure drops, the brake torque tends to remain high, or lags, due to the hysteresis in the brake. This residual or lagging torque causes the "worst" wheel to remain in a moderate slip regime (25-50% slip) for an additional period of time, forcing the antiskid system to lower the pressure to a level below the pushout pressure of the brake. Meanwhile, the "best" wheel is returning to a relatively low slip condition, providing little braking. With the pressure now decreased to a very low level, the "worst" wheel accelerates back to a low slip condition causing the antiskid system to generate an "ON" command, applying air to the brake again. Depending on how far below pushout the line pressure has fallen in the last release, a transport lag occurs due to the air volume change required in pushing the brake shoes back out to the drum. (The data for this truck indicated a maximum time lag of 40 milliseconds to go from 0 psi to the pushout pressure of 7 psi.) The net effect of the brake hysteresis and air supply delay is to permit free-rolling wheel conditions that otherwise would not occur.

In order to include the hysteresis and air supply delay effects exhibited by this vehicle within the computer model, (1) a small but significant modification was made to the hysteresis computer algorithm in order to represent more accurately the hysteresis-torque relationship at low pressure, and (2) the air supply transport delay effect, as presently represented, was modified and permitted to occur for any line pressure falling below pushout during an antiskid discharge cycle.

The remaining item requiring computer model refinements was the representation of force-deflection characteristics of the tandem suspension. The tandem suspension measurements made at HSRI (see Fig. 4) indicated a somewhat more complex relationship between vertical load and spring deflection than was presently assumed within the computer model. To better represent the suspension characteristics shown in Figure 4, the following equation was developed to approximate the force-deflection characteristics.

$$F_i = F_{ENV_i} + \left(F_{i-1} - F_{ENV_i}\right) e^{-\beta \left|\delta_i - \delta_{i-1}\right|} \qquad (1)$$

where

F_i is the suspension force at the current simulation time step

F_{i-1} is the suspension force at the last simulation time step

δ_i is the suspension deflection at the current simulation time step

δ_{i-1} is the suspension deflection at the last simulation time step

F_{ENV_i} is the force corresponding to the deflection, δ_i, of the outer envelopes of the measured suspension characteristic. F_{ENV} is represented in the simulation by two force vs. deflection tabular functions (upper and lower) input by the program user.

and β is an input parameter used for describing the rate at which the suspension force within an envelope loop approaches the outer envelope, F_{ENV}. Different values of β for increasing vs. decreasing load may be specified.

The representation of suspension force-deflection characteristics by the above method provided a convenient means within the subsequent

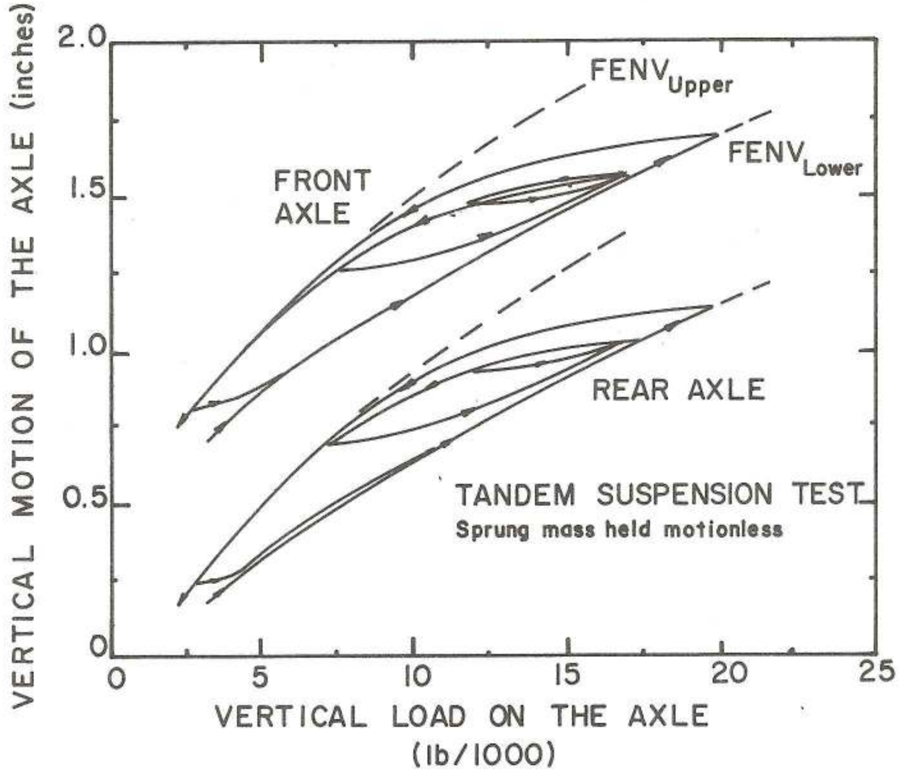

Fig. 4. Rear tandem suspension force-deflection measurement.

parameter sensitivity study for examining realistic variations in the rear suspension properties. For example, reduction in size of the outer envelope loops, F_{ENV} (closer together), corresponds physically to a reduction in the suspension coulomb friction, yet remains consistent with the overall qualitative nature of the measured data.

5. VALIDATION OF THE BASELINE COMPUTER MODEL

Following the program modifications discussed above, several computer runs were performed to observe the influence of these modifications and to determine a final baseline set of parameters representative of the average 60-mph vehicle stop. Vehicle test data showed considerable variation in stopping distances for repeated 60-mph stops (280 ft.-340 ft.), thereby confusing any definition of a baseline performance. However, since the majority of these test repeats did produce stopping distances between 285 ft-310 ft. and displayed similar cycling and dynamic behavior, a representative test result from this majority was selected as the baseline measured response. The final baseline computer representation of the average vehicle response included side-to-side brake torque imbalances, hysteresis in the rear brakes, air system transport lags at low line pressures, and the above-described rear suspension force-deflection characteristics—all selected to closely reflect the measured component characteristics discussed in the previous sections, and to further the agreement between initial simulation attempts and full-scale vehicle test results.

Figure 5 shows a comparison of the baseline measured vehicle response and the final baseline computer model representation for front and rear (leading tandem) axle brake pressures, front and rear axle "worst" wheel speeds, and vehicle velocity and acceleration. As can be seen, the level of agreement between the measured and simulated vehicle response is quite good, particularly in light of the variability displayed by the measured data for repeated tests. The level and frequency of measured brake pressures and wheel speeds during cycling are closely approximated by the simulation results. Furthermore, the baseline computer representation now predicted a stopping distance of 295 feet as compared to 296 feet for the baseline vehicle test.

Figure 6 illustrates the intensity of severe pitching and bouncing that an antiskid braking maneuver can evoke in a commercial vehicle of this class. The variables shown here are from the same baseline computer simulation run as Figure 5. The predicted front axle load on this vehicle routinely exceeds 30,000 lbs. during antiskid cycling with several cycles producing maximum loadings of nearly 40,000 lbs. Likewise, the predicted sprung mass pitch and bounce excursions during cycling have peak-peak amplitudes of approximately 1.5 deg. and 1 in., respectively.

The severe dynamic response which is normally associated with most commercial vehicle antiskid braking is significantly amplified when side-to-side brake torque imbalances and hysteresis effects are also present. Such additional factors, as discussed in the previous section, promote greater opportunities for wheels to free-roll during cycling, thereby magnifying the variations in tire braking forces from maximum levels to near zero. The pulse-like character of the measured and simulated vehicle acceleration traces directly reflects these extreme variations in rear tire braking forces. Figure 7 demonstrates the reduction in pitch and bounce motions predicted by the computer model when hysteresis is removed from the rear brakes. In addition, the predicted stopping distance is significantly decreased from 295 feet to 253 feet.

As will be indicated in the next section, the presence of hysteresis alone, without side-to-side brake torque imbalances and air system transport lags, is not, in itself, sufficient to cause a significant degradation in braking performance. Rather, it is the dynamical interaction amongst these less desirable brake properties during antiskid cycling that precipitates the markedly diminished braking performance in this vehicle.

6. PARAMETER SENSITIVITY STUDY USING THE BASELINE COMPUTER MODEL

Findings of the parameter sensitivity study performed on the baseline computer model are presented and discussed in this section. A summary of selected braking performance and descriptive numerics is provided in tabular form for each parameter variation examined.

The following categories were examined in the sensitivity analysis:

- Brake System Properties
- Tire Traction Characteristics
- Mass and Inertia Properties
- Suspension Properties

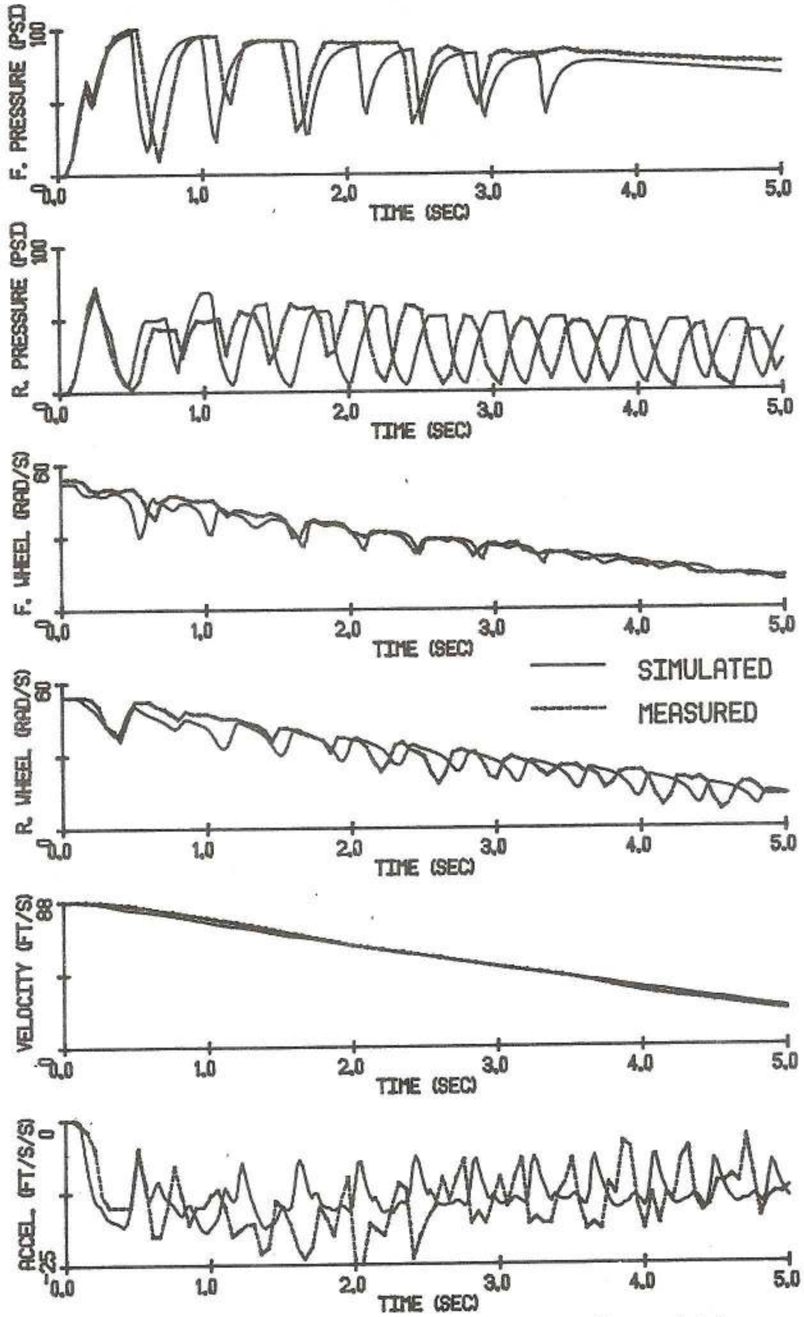

Fig. 5. Comparison of simulated and measured baseline vehicle response.

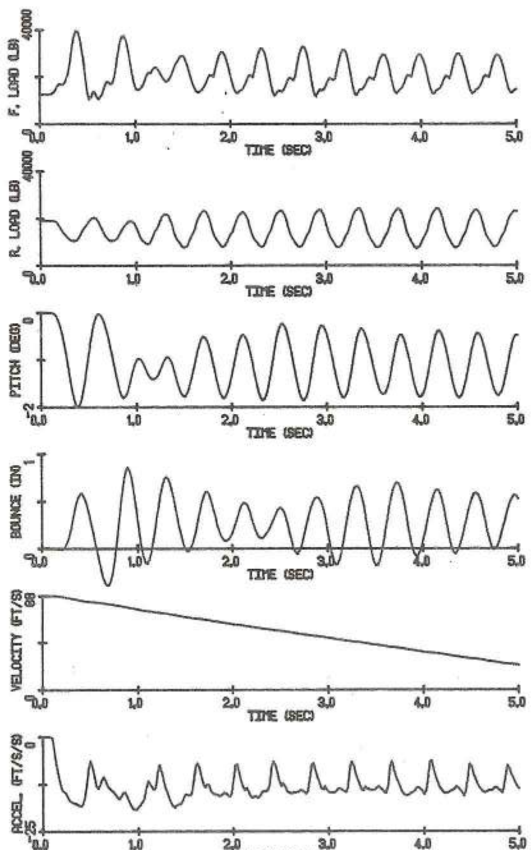

Fig. 6. Severity of simulated baseline vehicle response.

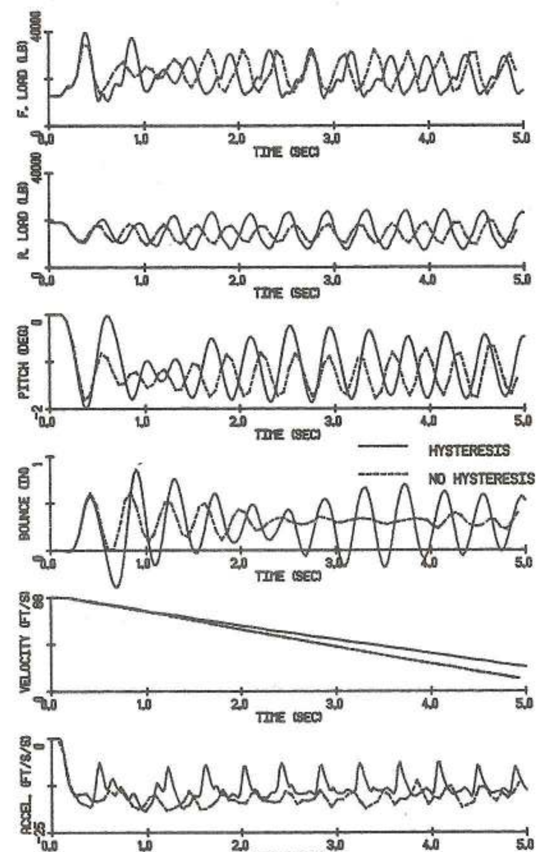

Fig. 7. Effect of removing rear brake hysteresis

•Wheel Dynamics •Antiskid System Characteristics

Table 3 presents the specific parameter variations studied and a summary of results showing predicted stopping distance and time, number of antiskid pressure cycles occurring for each axle, and the maximum vertical load experienced by the front axle. The baseline computer representation results are listed first, followed by the results obtained for each of the described parameter variations from the baseline case.

Brake System Variations - The brake system results shown in Table 3 clearly indicate a strong sensitivity of braking performance to the combined effects of rear brake hysteresis and side-to-side brake torque imbalances. The presence of both these factors were necessary for accurate representation of the dynamic response of the baseline vehicle. As discussed in the previous section, the synergistic mechanism involving rear brake hysteresis, side-to-side imbalances, and air system transport lags, is the principal brake-related reason for the reduced level of braking performance displayed by the baseline vehicle. The parameter sensitivity results further suggest that interruption of this synergism by removal or reduction of either brake hysteresis (Items 1.3, 1.4) or side-to-side brake torque imbalance (Item 1.5) can lead to significantly improved braking performance.

The other brake-related items of interest in Table 3 are the effects of reduced rear brake effectiveness (Item 1.2) and load proportioning (Items 1.11, 1.12). Each of these influences are similar in effect and are advantageous for two reasons: (1) the reduced torque output of the rear brake, whether deriving from a reduction in brake effectiveness or the indicated proportioning valve mechanisms, results in improved brake torque distribution on the vehicle, thereby lessening the redistribution requirements performed by the antiskid system, and (2) the reduced rear torque raises the minimum brake pressures experienced during cycling, thereby reducing the influences of hysteresis and air supply time lags, as discussed above.

The proportioning Items 1.11 and 1.12 demonstrate the effects of including load-sensitive brake proportioning valves, in addition to antiskid, on each axle. Item 1.11 represents an ideal valve which modulated treadle pressure near optimally at each axle in proportion to the prevailing vertical load, with an upper pressure limit determined by the demanded treadle pressure. Item 1.12 was an attempt to emulate the ideal proportioning scheme, 1.11, by use of a physically realistic method which employed suspension deflection as the mechanical means for estimating vertical axle loads.

Finally, Item 1.10 shows the predicted antiskid braking results for this vehicle with an ideal brake system having no hysteresis, imbalance, or air system transport lag effects.

Mass and Inertia Variations - The effects of payload location are demonstrated by Items 2.1 and 2.2 in Table 3. Lowering of the payload produces less pitching and bouncing motion with less fore-aft load transfer and correspondingly fewer antiskid cycles. Forward movement of the payload worsens the vehicle brake torque distribution.

Items 2.3 and 2.4 indicate an improvement in braking performance with vehicle pitch inertia modified from its baseline value. Presumably, the pitch and bounce dynamics associated with the pitch inertia of the baseline vehicle were more susceptible to antiskid cycling frequencies than the pitch and bounce dynamics resulting from increased or decreased values of inertia.

Tire Property Variations - Items 3.1 and 3.2 represent decreases in peak tire-road friction and the total tire-road friction characteristic, respectively. These results are interesting in that they do not demonstrate a proportional increase in stopping distance accompanying the lowered friction. Due to faster brake pressure releases during each antiskid cycle caused by the reduced tire traction, improvement in antiskid cycling efficiency offset much of the loss in braking performance expected from the reduced friction condition.

Variations in the tire vertical stiffness (Item 3.3) and tire traction sensitivity to large vertical loads (Item 3.4) demonstrate little or no effect upon braking performance. The negligible influence played by tire traction sensitivity for large vertical loads occurs because heavy tire loading during cycling corresponds to a low wheel slip condition, thereby minimizing any traction losses/gains obtained from the load sensitive property of the tire.

Suspension Property Variations - Variation of front suspension characteristics demonstrated minimal influence upon stopping performance. However, variation of rear suspension characteristics did have significant impact, particularly Items 4.4 and 4.5, rear Coulomb friction and spring rate changes. Lowering of the rear suspension Coulomb friction level promoted increased pitching and bouncing motion of the vehicle and correspondingly poorer braking performance. The number of front-axle antiskid cycles more than doubled for this variation. Reduction in the stiffness of the rear spring rate altered the pitch and bounce motions in the opposite manner, producing a more controlled stop with fewer antiskid cycles on all axles.

Item 4.6 indicates minimal effects on stopping performance for this vehicle from increasing rear tandem suspension interaxle load transfer from 5% (baseline) to 20%.

Wheel Dynamics - Item 5.1, wheel inertia, was the only wheel variation examined in this category and demonstrated negligible influence upon stopping performance.

Antiskid System Variations - Item 6.1 shows the effects on stopping performance for a faster prediction (brake release) by the antiskid logic module, and Item 6.2 for a faster re-selection (brake re-application) by the logic module.

Items 6.3-6.5 are related to the operating characteristics of the antiskid pressure modulation valve. Item 6.3 shows improvement in stopping performance from increased solenoid valve actuation delays for both release and re-application during each cycle. The small increase in braking performance indicated here is principally a result of deeper wheel slip excursions for the "best" wheel on each axle promoted by the increased valve lags.

Item 6.4 demonstrates the effect of altering the "pneumatic logic" of the modulator valve. During each pressure increase portion of a cycle, the "pneumatic logic" mechanism of the valve causes the rate of pressure increase to abruptly switch to a slow, linear rise rate. This mechanism is active during the modulation of rear axle pressure shown in Figure 5. By lowering the switching point 20% from the baseline operation, a significant improvement in braking performance is

Table 3. Parameter Sensitivity Results

Parameter Variation	Description	Stopping Distance (Ft.)	Stopping Time (Sec.)	Number of Cycles Front	Middle	Rear	Maximum Front Load (lb)
	Baseline	295	6.55	8	14	14	39,600
1.1	Front Brake Effectiveness Increased 25%	279	6.04	13	12	12	40,400
1.2	Rear Brake Effectiveness Decreased 25%	259	5.68	3	7	7	37,600
1.3	Rear Brake Hysteresis Decreased 50%	272	6.08	8	14	14	39,000
1.4	Rear Brake Hysteresis Removed	253	5.72	8	14	14	38,000
1.5	Rear Side-to-Side Brake Imbalances Removed	252	5.49	6	11	11	40,400
1.6	Air System Transport Lag Removed	287	6.34	8	14	14	40,000
1.7	Air System Losses During Cycling Decreased 50%	292	6.54	15	15	15	39,600
1.8	Front Brake Fade Sensitivity Increased 50%	306	6.82	6	14	15	39,600
1.9	Rear Brake Hysteresis Increased 25%	296	6.59	8	14	14	39,600
1.10	Ideal Brakes (No Hysteresis, Imbalances, or Air System Transport Lag)	230	5.12	10	14	14	39,200
1.11	Ideal Vertical Load Proportioning	259	5.68	3	9	9	38,300
1.12	Suspension Deflection Load Proportioning	260	5.64	4	6	6	39,000
2.1	Payload Lowered 2.0 ft.	259	5.72	6	13	13	34,600
2.2	Payload Moved Forward 1.5 ft.	312	6.90	5	17	17	39,000
2.3	Vehicle Pitch Inertia Decreased 25%	269	5.90	8	14	14	38,200
2.4	Vehicle Pitch Inertia Increased 25%	282	6.23	8	13	13	40,600
3.1	Peak Tire/Road Friction Decreased 10%	303	6.62	11	15	15	39,000
3.2	Total Tire/Road Friction Decreased 10%	304	6.65	11	15	15	38,600
3.3	Tire Vertical Stiffness Decreased 20%	300	6.81	9	14	14	39,200
3.4	Load Sensitivity of Tires Removed for Tire Loads Exceeding 10,000 lb	296	6.59	8	14	14	39,600

Table 3 (Cont.)

Parameter Variation	Description	Stopping Distance (Ft.)	Stopping Time (Sec.)	Number of Cycles			Maximum Front Load (lb)
				Front	Middle	Rear	
4.1	Front Suspension Coulomb Friction Decreased 50%	297	6.77	8	15	15	42,200
4.2	Front Suspension Coulomb Friction Increased 50%	273	5.99	8	13	13	38,400
4.3	Front Suspension Spring Rate Doubled	288	6.48	3	14	14	31,200
4.4	Rear Suspension Coulomb Friction Decreased 50%	325	6.90	19	14	14	39,400
4.5	Rear Suspension Spring Rate Decreased 50%	253	5.51	6	11	11	44,000
4.6	Rear Suspension Interaxle Load Transfer Increased from 5% (Baseline) to 20%	287	6.34	8	13	13	39,000
5.1	All Wheel Inertias Decreased 25%	296	6.60	9	14	14	39,800
6.1	Antiskid Logic Module Prediction Level Quickened by ~5% of Wheel Slip	304	6.78	9	15	15	38,000
6.2	Antiskid Logic Module Re-Selection Level Quickened by ~5% of Wheel Slip	285	6.33	8	14	14	39,600
6.3	Antiskid Solenoid Valve Delays Increased 50%	287	6.28	9	13	13	39,000
6.4	Antiskid "Pneumatic Logic" Level Decreased by 20%	263	5.78	4	11	11	39,600
6.5	Antiskid Pressure Rise and Exhaust Rates Reduced by 20%	278	6.28	7	13	13	39,400
6.6	Antiskid "Average Wheel" Option	265	5.80	6	12	12	40,000

predicted. The principal effect of this valve modification was to cause a faster brake release during each cycle, resulting in improved wheel cycling with increased minimum wheel slip levels.

Item 6.5 shows the influence of reducing the pressure discharge and application rates of the modulator valve. The resulting slow-down in pressure release and build-up produced improvement in wheel cycling, attained primarily from higher levels of minimum wheel slip.

Lastly, Item 6.6 demonstrates the improvement in stopping performance which can be expected by employement of "average wheel" axle control on this vehicle. "Average wheel" antiskid systems simply utilize the average of left and right wheel speeds for each axle within the logic module in place of the axle's "worst" wheel speed. While such systems usually provide improved braking from higher average wheel slip conditions, the likelihood of diminished vehicle directional stability is generally enhanced.

Finally, it should be reiterated that the mechanism having primary influence over the stopping performance of this vehicle was the aforementioned interactive effects of side-to-side rear brake imbalance and hysteresis. Many of the other parameter variations which displayed significant influence upon stopping performance, did so indirectly, by modifying or lessening the adverse effects of the baseline brake imbalance-hysteresis mechanism. Primarily for this reason, the results of the parameter sensitivity study presented here may not be applicable to other vehicles, particularly ones not having similar brake system properties.

7. CONCLUSIONS

The antiskid braking performance of the examined baseline vehicle was primarily dominated by the adverse effects deriving from rear brake torque imbalances and hysteresis precipitated during wheel cycling. The most noteworthy findings of the parameter sensitivity study for improving the baseline stopping performance, outside of reducing the side-to-side imbalances and rear brake hysteresis, were found to be: (a) reduction in rear brake effectiveness or use of load proportioning valves, (b) modification of antiskid system operation, and (c) modification of rear suspension properties.

This study has also presented the opportunity to demonstrate the utility of detailed computer models in representing and studying the frequently complicated and inter-active properties of complex physical systems. While it is usually desirable to employ as simple a mathematical model as possible to represent a dynamical system under study, in certain cases the complexity of the dynamical system precludes use of simple mathematical approximations which would fail in representing important detailed mechanisms. Such was the case in this study where relatively obscure brake system properties combined to strongly influence the predicted braking performance during antiskid cycling.

ACKNOWLEDGEMENT

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THE DYNAMICS OF VEHICLES

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