BRAKE MODELING ANTI-LOCK SIMULATION

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This paper describes how the torque characteristics of commercial vehicle brakes can be modeled and made part of a hybrid computer simulation of an antilock system. The braking system of a commercial vehicle is an open loop pneumatic-mechanical system with feedback controlled by the vehicle operator. To achieve optimum performance, i.e. short stopping distance and vehicle control, the vehicle operator must be aided by a brake control system which senses wheel velocity and acceleration, and then reduces excessive brake pressure and reapplies controlled brake pressure. Most systems in use today consist of an electronic computer, a solenoid controlled air valve, and wheel speed sensors. Design and analysis of this complex system require a combination of engineering simulation and experimental testing. In the simulation of a commercial vehicle antilock system, three areas are difficult to model; the air valve system, the tire-road interface, and the brake drum/brake shoe interface. Their simulation is discussed along with techniques for verifying the accuracy of the models being used. Elements that make the study of the air brake stopping problem difficult are a surface friction range of 8 to 1, load variations of 10 to 1, and transitions in brake torques of 8 to 1. In addition to these factors, the effects of variable pneumatic delays and steering torques must be considered along with the desire to produce a low cost, highly dependable, fail-safe/self-test unit.

Technical Approach

The ultimate abjective of this engineering effort was to design an anti-lock system which not only met the performance of Motor Vehicle Safety Standard 121 (1), but also the requirements of reasonable cost, high reliability, maintainability, immunity to environmental influences, smooth stops, etc.

To meet these objectives, a parallel effort of simulation, engineering and instrumented road testing was initiated. The road testing would determine the real world performance and the simulation would be the place to evaluate new components and control concepts and predict their performance.

Those that showed promise could then be road tested. Thousands of candidate control schemes could be tried in the Simulation Laboratory in the time it would take to road test just a few, shortening the total development time and reducing the cost.

The equipment used included analog Computer with hybrid interface, electronic mode control, and digital logic, and a Digital Computer (See Figure 1).

The analog computer was used to model the vehicle dynamics, brakes, and tire road interface and to control the simulation. The digital computer was used to simulate the anti-lock controller logic (See Figure 2).

The simulation, as shown in the block diagram, Figure 3, consists of several sub-blocks and mechanizes these models either on the analog/hybrid computer or in some cases, use the actual hardware to "close the loop."

Simulation runs or solutions were made for a variety of conditions, i.e., road surface, brake unbalance, etc., and compared to road test data. The sub-blocks were then modified and/or changed to agree with the road test data. These were modified as better and more complete test data was taken.

In the development process, two different types of simulation were performed. One was a two axle model which used two and a half analog computer consoles, i.e., about 230 operational amplifiers and associated equipment; and two general purpose digital computers, one for each anti-lock controller. The simulation model included the variable spring and damping coefficients for both the suspension and tires, the weight shift as a function of wheel base, center of gravity, and axle loading. This two axle simulation was used to study the interaction effects of two anti-lock systems, the weight shift front to rear, and vehicle pitch, deflection and vibration. This data determined the "roughness of ride" and suspension system efforts.

The other type of simulation which was and still is being used for anti-lock development and customer demonstration, is a single axle model with mass transfer. For the most part, this has proven sufficient for present development effort. This single axle model verifies operation with split coefficient, i.e., one wheel on a different surface from the other, all degrees of brakes unbalance, and weight shift side to side. The simulation model provides independent control of the brake gain, slip characteristics, vehicle loading, rate and force of brake application, axle loading, center of gravity, weight

Figure 1. Simulation laboratory.

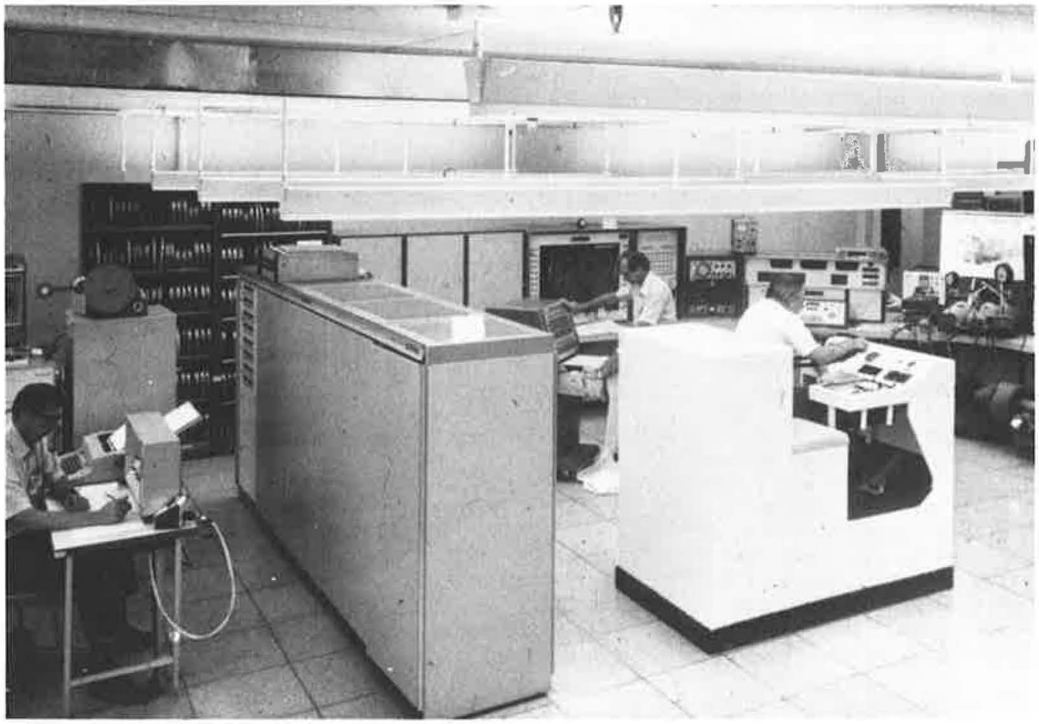
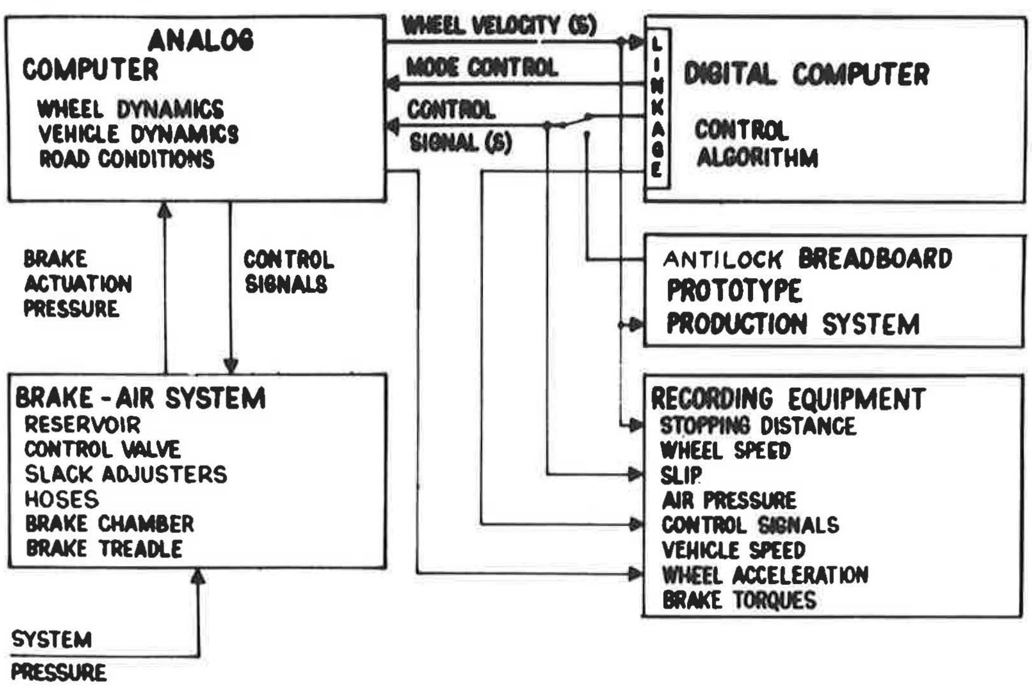


Figure 2. Simulation block diagram.



shift, etc.

The parameters that are usually recorded on one or more strip chart recorders are vehicle speed and acceleration, wheel speed and acceleration, slip and road torque, brake chamber air pressure, treadle pressure, reservoir pressure, and brake torque. Stopping distance and time of stop are displayed on digital meters in feet and seconds, respectively.

The simulation setup described in this paper is for one axle. The two axle simulation is similar.

Simulation Setup

The equations (or models) of motion for a commercial vehicle are fairly well known. The actual parameters or function must be obtained through searches in technical literature, experimental testing and measurement and equipment specifications. The modeling of a pneumatic air brake system is complicated by the compressibility and other non-linearities associated with the air system. The brake chambers change volume when they are pressurized. It was decided early in the program to use the actual reservoir, hoses, relay valve, solenoid valve, slack adjusters, brakes, etc., instead of modeling them. Both wedge and cam brakes were used. This made it much easier to study the effects of hose size and length.

The input to the braking system is driver input, i.e., the rate of change of brake treadle force which in turn produces the treadle pressure P_t. Mathematically this is:

$$P_t = f\ (t) \qquad 1$$

where P_t is the treadle pressure. In the laboratory simulation of this, two methods were used (See Figure 4). One method provided a simulated truck cab with a brake treadle for an operator to apply the pressure desired. The other and most often used method, had a solenoid valve in series with a pressure regulator and a needle valve to provide a repeatable input (command) pressure. The regulator simulated the treadle pressure and the needle valve provided the proper rate of pressure change.

Two different air reservoirs were used to provide FMVSS-121 minimum requirements for trucks and trailers. This pressure, P_s, was routed to the brake control valve. Typical brake hoses were then routed to either a wedge brake or cam brake axle (See Figure 4). A pressure transducer on the respective air chamber was used to obtain the chamber pressure, P_B. The air supply to the reservoir was cut off once the stop was started so that air consumption for the stop could be determined.

The next problem was "knowing the chamber pressure, how is the brake torque calculated?" Early in the program, a linear (straight line) function with a typical slope was used. This completed the simulation loop, but did not produce good correlation with vehicle test results. As the program progressed, slope changes were added to the original straight line function. Hysteresis and the effects of vehicle velocity were also added. The current brake model uses data obtained on Rockwell's brake test dynamometer (See Figure 5) which also closely matches the data obtained by the Highway Safety Research Institute, (1). Mathematically this is:

$$T_B = f\ (P_B,\ V_t) \qquad 2$$

where T_B is the brake torque, P_B is the air pressure in the brake chamber measured with a pressure transducer and V_t is the vehicle velocity.

Reference (3) suggests that the rate of energy input to the brake is also a pertinent factor. As yet, this had not been included in the simulation.

In addition, the brake torque T_B, by definition, must be equal to or less than the road torque T_R, when the wheel velocity $V_w = 0$. Otherwise, the wheel would have a rotational velocity opposite the vehicle, i.e., would be going backwards. The net torque then is

$$\triangle T = T_R - T_B \geq 0 \qquad 3$$

where T is the net torque, T_R is the road torque generated by the slip and V_w is the rotational velocity of the wheel. The velocity is calculated by integration of the wheel acceleration:

$$V_w = \int \triangle T\ (R/I)dt \qquad 4$$

where R is the radius of the wheel in feet, and I is the polar moment of inertia of the wheel.

If the vehicle velocity is known, the wheel slip can be calculated:

$$SLIP = 1 - (V_w/V_t) \qquad 5$$

where slip is unitless ranging from zero (free rolling) to one (lockup). V_w and V_t are the velocities of the wheel and vehicle, respectively. If the instantaneous slip is known, the coefficient of friction can be calculated. The first order approximation of u, (mu), the tire-road coefficient of friction or the normalized longitudinal force, F_x/F_z, is that it is a function of slip:

$$u = F_x/F_z = f(SLIP) \qquad 6$$

where F_x is the longitudinal shear force and F_z is the vertical load. The value of u at slip = 0 is usually assumed to be zero and u = .75, .3, and .1 with slip = 1.0 for dry, wet, and icy surfaces, respectively. Values of slip in between are not easily determined. A first approximation is that u increases linearally to a peak about slip = .15 to .20 and then decreases to the slip = 1.0 value (See Figure 6). The vehicle lateral stability decreases with slip; hence, a stop with all wheels with slip = .2 (wheel speed 80 percent of vehicle speed) would result in a short controlled vehicle stop.

Recent studies (2) indicate that the friction is also a function of vertical load, wheel velocity, inflation pressure, and tire wear condition. As yet, these effects have not been included in the simulation .

The road torque is calculated from the friction:

$$T_R = R\ \ W_E\ \ u \qquad 7$$

where T_R is the road torque, R is the wheel radius and W_E is the effective weight on the wheel, i.e.,

Figure 3. Simulation flow diagram.

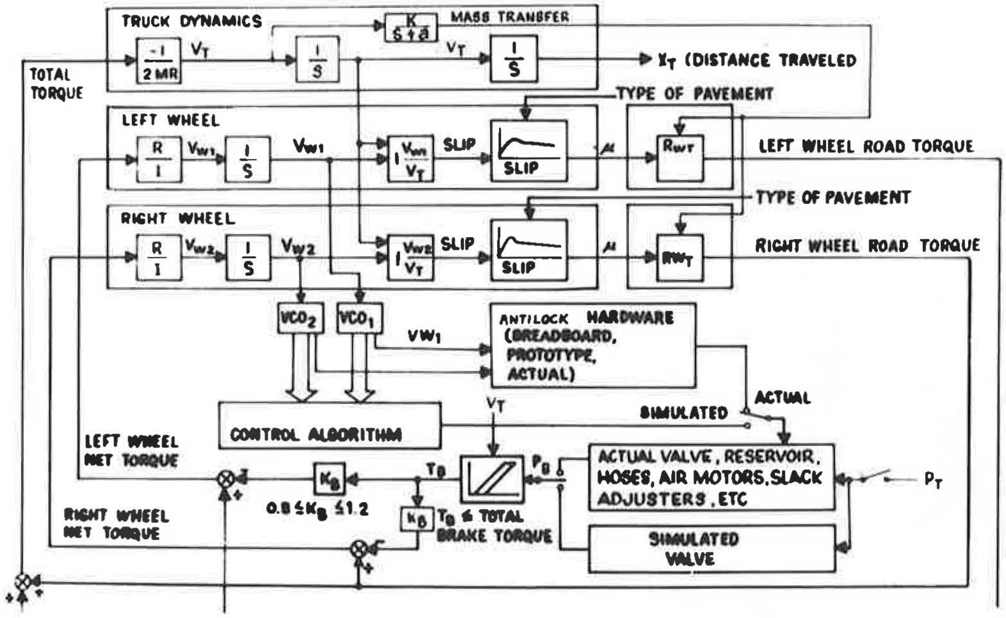


Figure 4. Air block diagram.

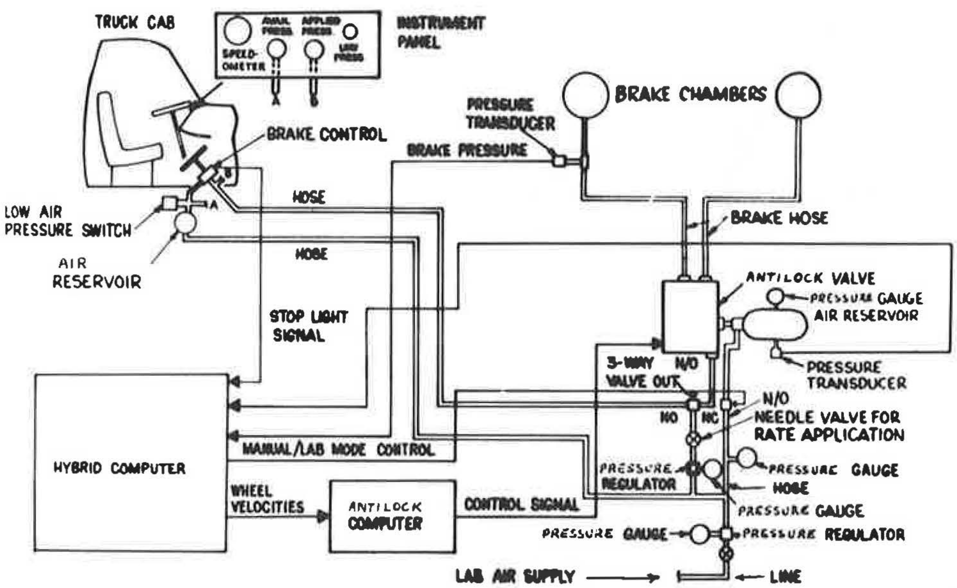


Figure 5. Brake test dynamometer.

Figure 6. Road torque and lateral stability.

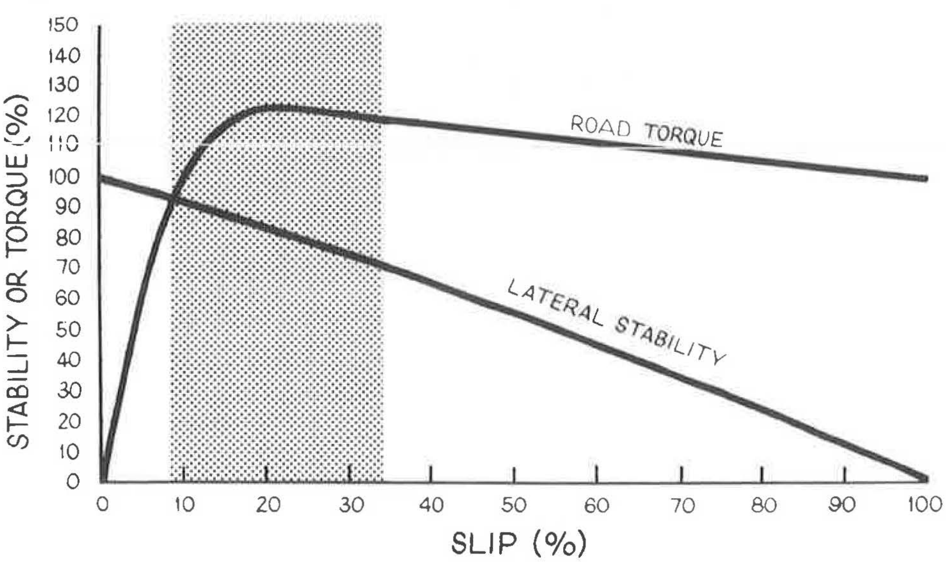


Figure 7. Excess pressure.

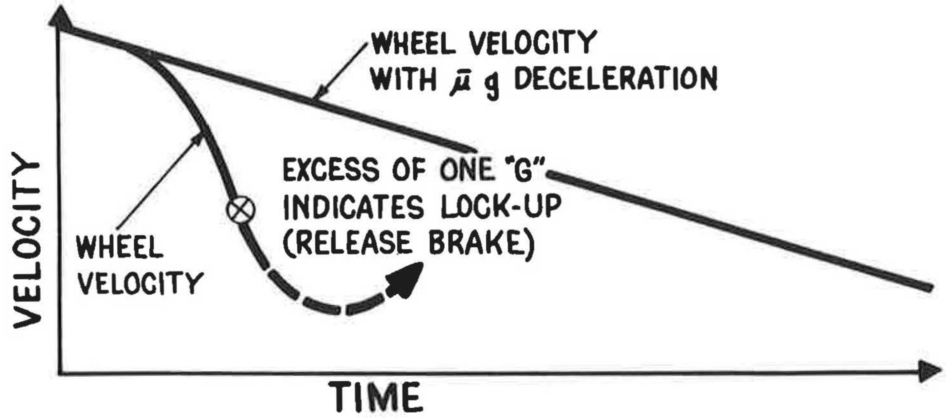


Figure 8. Low pressure.

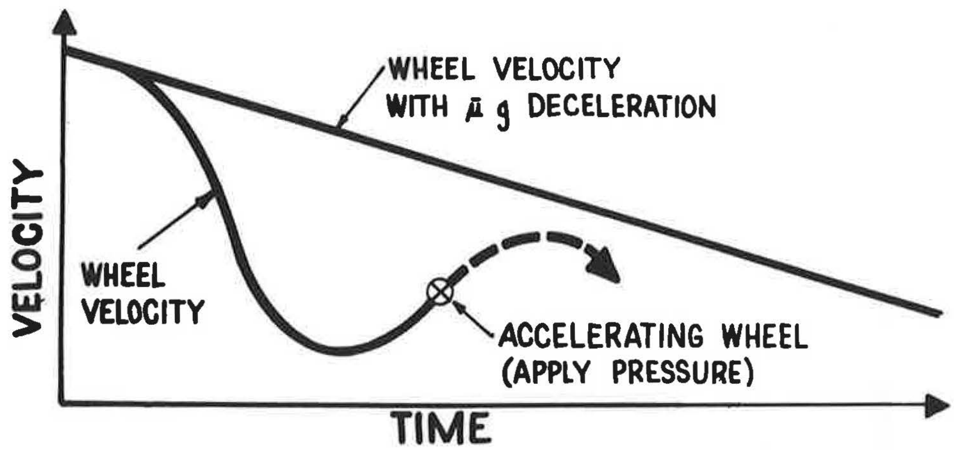


Figure 9. Low road torque.

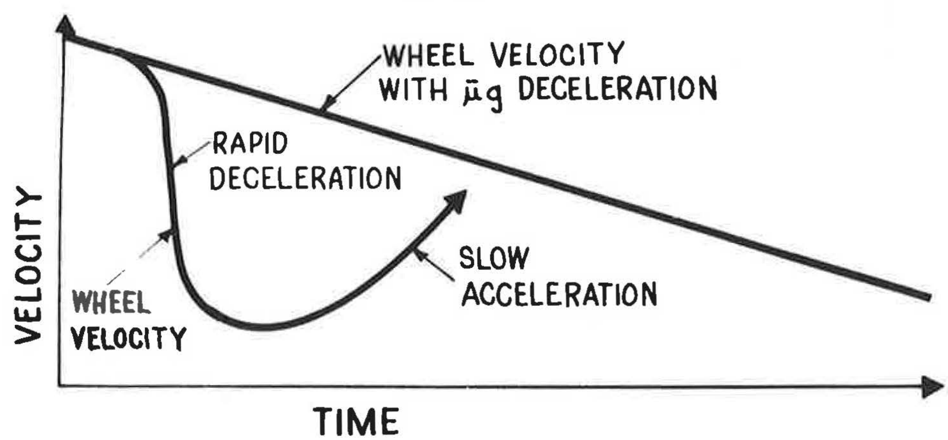
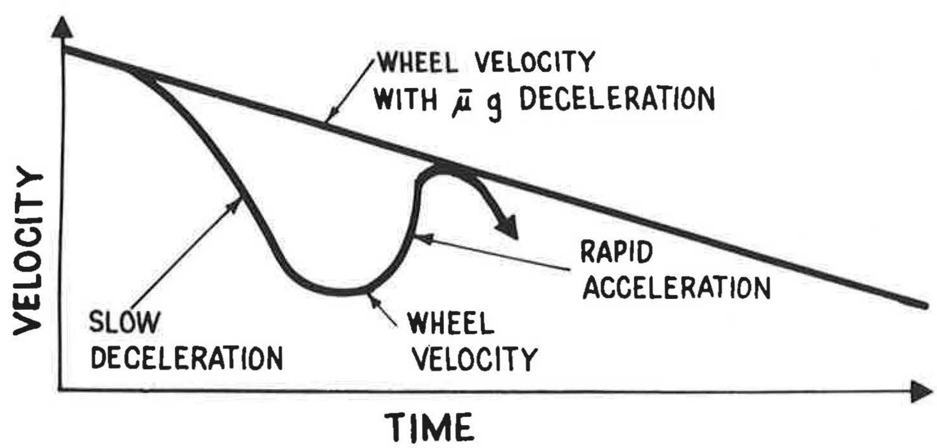


Figure 10. High road torque.



the static weight, plus or minus the weight transferred. The single axle simulation did not include the effects of the suspension system.

If a 1 of the individual road torques are summed to create a total road torque, vehicle deceleration can be computed:

$$\dot{V}_T = \frac{F_E - F_D - \sum (T_R/R)}{M_T} \qquad 8$$

where $\dot{V}_T$ is the truck deceleration. T_R is the sum of all the road torques, M_T is the total mass of the truck (gross weight), R is the wheel radius, F_E and F_D are the engine and aerodynamic drag forces, respectively, which are assumed to be independent of the braking simulation (equal to zero). The dot indicates a time derivative.

The weight transferred was modeled as:

$$W_E = W_S \pm \frac{W_T\,(C/L)\;(\dot{V}_T/g)}{2n\,(S + a)} \qquad 9$$

where W_E is the effective weight on one wheel, W_S is the static weight on one wheel, W_T is the total vehicle weight, C is the center of gravity height, L is the wheel base, g is acceleration due to gravity, n is the number of axles in front or behind the center of gravity, and (S+a) is a first order lag which approximates the pitching time constant of the vehicle typically about .5 seconds.

The two wheel velocity signals, for the simulated axle, are fed into voltage controlled oscillators which simulate the wheel sensors. They are programmed such that the voltage is proportional to wheel velocity and can be fed into either a general purpose digital computer for control algorithm development, breadboard, prototype, or production hardware.

System Design

In the system design phase of the engineering effort, all of the design concepts, DOT specifications, system environmental considerations, and cost and reliability factors are used as inputs. The output of the simulation of the wheel anti-lock is the system definition, logic control equations, and an understanding of the effects of parameter changes on system performance.

The stopping of a vehicle is normally caused by an operator (truck driver) pushing on the brake treadle (an air pressure regulator with output proportional to applied foot pressure). If this pressure is large enough, it will cause the wheel to decelerate in excess of one "g" (32 ft/sec^2). Therefore, a good wheel anti-lock system should release (at least partially) some of the brake torque, most commonly done by releasing some of the brake air pressure (Figure 7). If too much pressure is released (the instantaneous ideal pressure would keep the wheel rotation about 80 percent of the vehicle velocity for a minimum distance stop) the wheel will tend to accelerate (Figure 8).

By using logic and measuring the rate of wheel acceleration and deceleration, information can be derived. If the deceleration is rapid and the acceleration is slow, Figure 9, this indicates a large excess of brake torque caused by a low u, (mu), unloaded condition, weight shift, or high air pressure. If the wheel deceleration is slow and the acceleration is rapid, Figure 10, this indicates low excess brake torque caused by high mu, loaded vehicle, weight shift onto the axle or low air pressure.

This information can be used to determine what the control scheme should be to reduce any excess brake pressure and thereby prevent uncontrolled wheel lock-up and the potential consequences. All types of candidate systems can be studied and evaluated by use of the simulator. The poorer ones are discarded and the better ones are refined and used for candidate breadboard, prototype, and production hardware.

System Verification and Test

Once a candidate system has been selected by use of simulation and good systems engineering judgment and practices, it is necessary to evaluate the design.

Again, simulation can be used to verify and test the device. The major difference is that the actual breadboard, prototype, or production system is substituted for the simulated controller. As with the design phase, the system performance was compared with customer and DOT specifications and requirements over the range of environmental conditions. These included dry, icy, and wet road surfaces, loaded and unloaded vehicles, wedge and cam brakes, brake unbalance, various brake materials, effects of weight transfer, split road surface conditions, road surface transitions, various rates of brake application, variations in battery supply voltage, and with the use of an environmental chamber, high and low temperature operation. The objective was to verify and test the system under a wide variety of conditions.

Why Simulate?

Simulation is used because it provides the following advantages over on-vehicle testing, alone.

1. Instantaneous results for a whole spectrum of environmental conditions from ice to concrete, all loads, all conditions of brakes, or any other set of conditions, a near optimum can be designed.
2. These results are in real time with real hardware.
3. The test conditions are controllable and repeatable.
4. All tests are completely instrumented.
5. When the simulation results are correlated with truck tests, truck testing efficiency is increased.

We found simulation to be the most practical way to test with so many variables. In the more than five years of simulation of anti-lock systems, we made several hundreds of thousands of simulated vehicle stops. Some results of this are typified in Figures 11 and 12. Figure 11 pictures a typical tractor-trailer vehicle in a panic stop situation with all wheels locked up. Steering control was lost, the vehicle went out of the twelve foot lane and "jack-knifed." In Figure 12, the same vehicle, same conditions, except with an anti-lock system installed. Results were a smooth, short, and safe stop.

Figure 11. Typical tractor-trailer without anti-lock system.

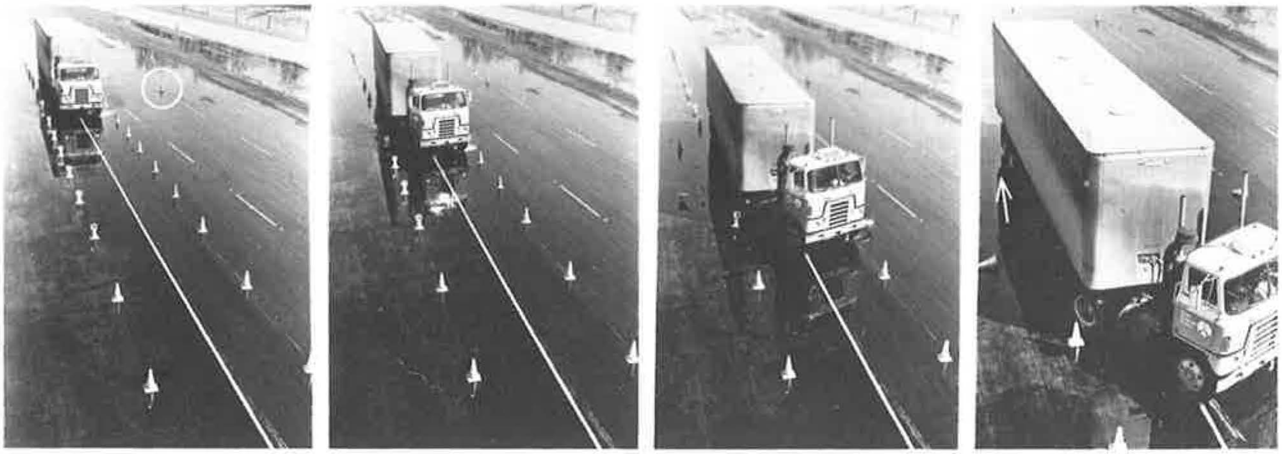


Figure 12. Typical tractor-trailer with anti-lock.

1. Federal Motor Vehicle Safety Standard No. 121, Air Brake Systems: 49CFR 571.121.
2. Robert D. Ervine and Pual S. Fancher, "Preliminary Measurement of Longitudinal Properties of Truck Tires." Paper 741139, presented at SAE National Truck Meeting, Troy, Michigan, November 1974.
3. T.M. Post, P.S. Fancher, and J.E. Bernard, "Torque Characteristics of Commercial Vehicle Brakes." Paper 750210, presented at SAE Automotive Engineering Congress and Exposition, Detroit, Mich., Feb. 1975.