ARTICULATED VEHICLE SIMULATION: A FRESH APPROACH TO SOME RECURRING PROBLEMS

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ABSTRACT

Current and pending goverment standards have encouraged an expanding interest in the braking and handling of articulated vehicles. Vehicle dynamicists are increasingly turning to simulation as an aid in their analysis. Two facets of the simulation of these vehicles which are recurring stumbling blocks will be discussed here, namely (1) the digital simulation of suspensions exhibiting large amounts of coulomb friction, and (2) the limitations of previous representations of the hitch and a more comprehensive hitch model. Some details of the simulation for which this work was done and results from the validation effort are also given.

I. INTRODUCTION

The problems of vehicle handling appeared in the literature as long ago as 1925 when the pioneering analysis of Broulhiet (1) was published. Subsequent investigators developed linearized equations whose solution would yield the trajectory of a vehicle subject to time-varying steering or braking. However, since even these linearized equations of vehicle motion are quite difficult to handle in the general case, it is not surprising that simulation has been a tool frequently used by vehicle dynamicists.

Perhaps the best known early computer simulation was developed in 1961 by Ellis (2) who developed a three-degree-of-freedom analog computer model for studying the lateral motion of an articulated vehicle. Since that time, the advent of more and more sophisticated computing equipment has led to the possibility of simulations of increasing complexity.

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Presently, many research facilities make use of highly nonlinear passenger car simulations with at least fourteen degrees of freedom, including six degrees of freedome for the vehicle body (the so-called sprung mass), a vertical or "wheelhop" degree of freedom for each wheel (or unsprung mass), and a spin degree of freedom for each wheel. (See, for example (3).) In the case of articulated vehicles, of course, proportionately more degrees of freedom are necessary. Since these simulations must necessarily deal with a fair amount of algebraic and trigonometric details which are quite inconvenient in a purely analog operation (for instance, tangents of various angles, and perhaps a variety of multiplications and divisions to relate the state variables to the shear forces at the tire-road interface), significant digital capability is required. Thus, these large-scale simulations tend to be either hybrid or purely digital.

There are, in fact, conditions when it is most advantageous to use purely digital facilities. For example, the size of the program may severely tax available hybrid facilities, or it may be desirable to be able to "deliver" the simulation to a variety of independent locations. Whatever the rationale, it is clear that many vehicle dynamicists are turning to digital simulation.

In the present paper, two problems implicit in the simulation of articulated vehicles are discussed. In Section 2, a method of handling very large coulomb friction in a digital context is considered, and in Section 3 a fresh view of the simulation of the mechanics of the hitch point is given. Finally, some results from the simulation for which this work was done are presented.

II. COULOMB FRICTION

The friction force produced by two sliding surfaces is classically defined as

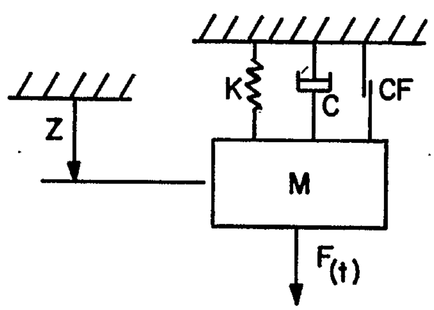
$$f \leq \mu N \tag{1}$$

where
- f is the force of friction,
- μ is an experimentally derived parameter,
- N is the contact "normal force" between two sliding surfaces.

Equation (1) is empirical in nature, and describes, in an approximate manner, an observed phenomenon. To illustrate this point, consider the simple system shown in Figure 1.

ILLUSTRATION 1

Mass Spring System with Viscous Damping and Coulomb Friction



The equation of motion of the mass M shown is

$$\overline{CF} = Mg + F(t) - Kz \tag{2a}$$

for $\dot{z} = 0$, $|\overline{CF}| < CF$

otherwise,

$$M\ddot{z} + Kz + C\dot{z} + CF \frac{|\dot{z}|}{\dot{z}} = Mg + F(t) \tag{2b}$$

where CF is the maximum allowable magnitude of the coulomb friction force $\overline{CF}$, F(t) is the driving force on the system, K is the spring rate, C is the viscous damping coefficient, z is the displacement of the mass M (z=0 at the free length of the spring), and g is the gravitational constant.

From Equation (2a), it can be seen that no motion is possible for the system initially at rest until the magnitude of the quantity Mg + F(t) - Kz becomes greater than $|\overline{CF}|$. At this point, motion ensues which is described by Equation (2b). The motion of the mass will continue to be described by Equation (2b) until the system again meets the conditions of Equation (2a).

In developing a digital simulation of a system with coulomb friction, Equations (2a) and (2b) present special problems. Since the velocity $\dot{z}$ is known only at discrete times, the time when $\dot{z}$ equals zero cannot easily be found. Thus, the actual time to switch from solution of Equation (2b) to solution of Equation (2a) is not known. There are a variety of ways to circumvent this problem, some of which are considered below:

(a) Continuously solve Equation (2b). This method is unsatisfactory (especially for large amounts of coulomb friction) since the system will "chatter" around the static equilibrium position. A slightly negative $\dot{z}$ produces large coulomb friction, which causes large positive $\ddot{z}$. When the large positive $\ddot{z}$ is integrated over a short time, positive $\dot{z}$ results. The cycle then repeats with opposite signs. The period of this "chatter" is twice the integration time step.

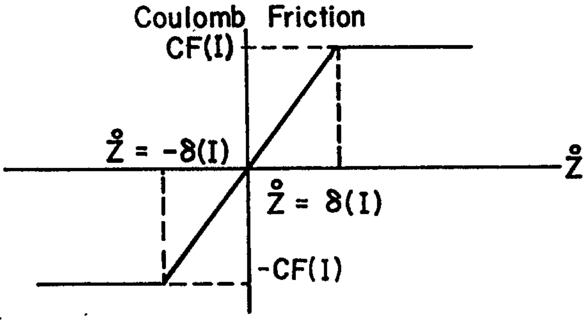
(b) Use an "equivalent viscous damping." By this method an increased value of C is chosen to compensate for the elimination of coulomb friction. This method can be useful when the coulomb friction forces are small compared to the velocity sensitive forces, as in certain automobile applications, but, in general, it cannot yield satisfactory results in truck dynamics since the forces of coulomb friction are normally much larger than those of viscous friction.

(c) Use a limiting (saturation) function. A schematic diagram showing the function used in the simulation is given in Figure 2. This function effectively eliminates the problems encountered with the methods described in (a) and (b).

To eliminate the possibility of chatter, the magnitude of δ should be large enough that, if the coulomb friction were the only dynamic force in the suspension, the velocity across the suspension cannot change from δ to a negative value in one integration time step. This stipulation may be applied with the aid

ILLUSTRATION 2

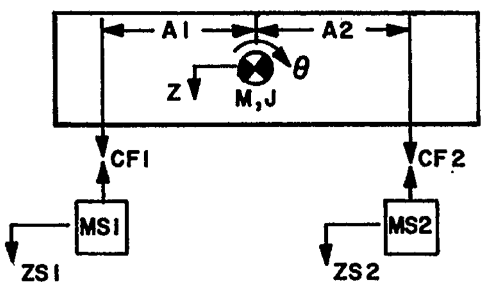
Coulomb Friction Represented By Limiting (Saturation) Functions



of the simple free-body diagram of a system of sprung and unsprung masses given in Figure 3, which can represent a highway vehicle restricted to straight line motion. The velocities across the suspensions, which are the relative velocities between the sprung mass and the front and rear unsprung masses are, respectively:

ILLUSTRATION 3

System Used in Sample Calculations of δ



$$SD1 = \dot{ZS1} + A1\dot{\theta} - \dot{Z} \qquad (3)$$

$$SD2 = \dot{ZS2} - A2\dot{\theta} - \dot{Z} \qquad (4)$$

The accelerations may be assumed to be limited in the following way:

$$|\ddot{Z}| \leq \frac{1}{M}(CF1 + CF2) \qquad (5)$$

$$|\ddot{\theta}| \leq \frac{1}{J}(CF1(A1) + CF2(A2)) \qquad (6)$$

$$|\ddot{ZS1}| \leq \frac{CF1}{MS1} \qquad (7)$$

$$|\ddot{ZS2}| \leq \frac{CF2}{MS2} \qquad (8)$$

During a time interval Δt, the changes in the velocities across the front and rear suspensions are therefore limited by

$$|\Delta S1D| \leq \Delta t\left[CF1\left(\frac{1}{MS1} + \frac{1}{M} + \frac{A1^2}{J}\right) + CF2\left(\frac{1}{M} + \frac{A2 \cdot A1}{J}\right)\right] \qquad (9)$$

$$|\Delta S2D| \leq \Delta t\left[CF2\left(\frac{1}{MS2} + \frac{1}{M} + \frac{A2^2}{J}\right) + CF1\left(\frac{1}{M} + \frac{A2 \cdot A1}{J}\right)\right] \qquad (10)$$

Thus, to preclude chatter, the coulomb friction break points, $\pm\delta(I)$ in Figure 2 for suspensions 1 and 2 may be set to

$$DEL1 = |\Delta S1D| \qquad (11)$$

$$DEL2 = |\Delta S2D| \qquad (12)$$

In a simulation involving a three-dimensional model, one would, of course, have to include the suspension velocity change due to axle roll and body roll. Break points, however, may easily be found using expressions similar in character to Equations (9) and (10). One will find that these break points are very small in the case of a passenger car. However, in commercial vehicles one might have

CF = 3000 pounds

MS = 1500/g

and, using Δt = .005 as a typical integration time step, the break point arrived at would be approximately

DEL $\cong$ 5 inches/sec.

Since velocities measured in a highway test of such a suspension may be expected, in many cases, to be far in excess of 5 inches/second, reasonable* results may be anticipated from this model.

*One unreasonable result stemming from this model should be noted: The presence of coulomb friction in a suspension leads to the conclusion that the static equilibrium configuration is not unique. (Note the static equilibrium position in Figure 1 must only be in the range $\frac{Mg-CF}{K} < z < \frac{Mg+CF}{K}$.) Since the friction in the present model approaches zero with $\dot{z}$, the simulated equilibrium position is $z = \frac{Mg}{K}$.

The solution method for the break points has an analogy in other situations, e.g., the break point applicable to the coulomb friction generated in yaw at the fifth wheel of a tractor-semitrailer. In the following section we will deal with the more difficult problem of the determination of the appropriate forces and roll moment transmitted through the fifth wheel.

III. THE FIFTH WHEEL

Articulated vehicles, be they a tractor-semitrailer rig capable of transporting 100,000 lb. payloads or a passenger car-recreation vehicle, must make use of some mechanical interconnection. This interconnection, which shall here be termed the fifth wheel, typically has the role of restricting one point near the rear of the tractor and one point near the front of the semitrailer to a common location while allowing the heading angle of the tractor and semitrailer to be quite different. Thus the vehicle "articulates" at the hitch and therefore is able to track around corners which would be impossible for a non-articulated vehicle of comparable length.

The analysis of the mechanics of the fifth wheel, as presented here, departs radically from previous analyses, most notably that of Leucht (4) and Mikulcik (5). It will be beneficial at this juncture to briefly review their work.

The vehicle model of Leucht entails four degrees of freedom, namely, the yaw plane coordinates X and Y and yaw angle ψ of the tractor, and the articulation angle of the trailer relative to the tractor. It was assumed that the fifth wheel could transmit a yaw moment (due to friction) but no pitch or roll moment. The lateral transfer of wheel loads experienced by the tractor is calculated on the basis of quasi-static considerations with the aid of an input parameter described as the roll rate distribution.* Since roll moments cannot be transferred by the fifth wheel, the roll moments on the trailer are balanced entirely by the lateral transfer of load on the tires of the trailer.

*Note that, since roll is not included in the model, the system is statically indeterminate and thus requires this additional parameter.

The vehicle model of Mikulcik entails eight degrees of freedom, namely, three coordinate and three rotational degrees of freedom for the sprung mass of the tractor, and two rotational degrees of freedom for the sprung mass of the trailer. The fifth wheel constraint is quite carefully conceived mathematically. When the tractor and semitrailer are in line, the respective roll angles are constrained to be equal, and the appropriate adjustments are made in the presence of an articulation angle. The roll moment transmitted by the fifth wheel is precisely that moment required by the geometric constraint.

Both of the above models constitute a reasonable simulation of certain braking and/or handling maneuvers. However, in applications in which it is crucial to transmit a roll moment through the hitch it is obvious that the "ball hitch" representation of Leucht leads to serious problems. It is less obvious but nevertheless empirically demonstrable that, since the tractor of a commercial tractor-semitrailer combination may be expected to be very pliable in torsion, the roll moment found by assuming a rotational constraint between a rigid tractor and a rigid semitrailer may also be in error. Thus in the analysis to be presented herein, the tractor and semitrailer each have six degrees of freedom—there is no geometric constraint at the fifth wheel. There is rather a force and moment constraint in which tractor and trailer are subject to equal and opposite forces and moments dependent on the difference in the fifth wheel position and orientation as measured on the tractor and the semitrailer.

There are benefits to this new formulation:

1. While fifth wheel constraint results very similar to the models of Leucht or Mikulcik may be simulated by proper choice of fifth wheel constraint parameters, one may also choose the roll moment distribution parameter based on empirical work.

2. Since the dynamic coupling caused by the rigid fifth wheel constraint has been removed, no matrix inversion is required to solve for the acceleration components of the tractor and the semitrailer. Further, since the forces and moments at the fifth wheel are no longer direct functions of vehicle accelerations, they may be computed in a straightforward manner.

THE FORCE TRANSMITTED AT THE FIFTH WHEEL

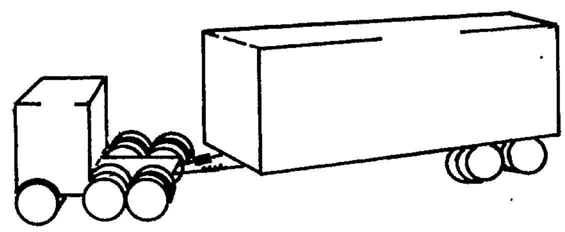
Initially, the fifth wheel position of the tractor and semitrailer are assumed to be identical. As the simulation run proceeds, however, forces developed at the tire-road interface will cause disparate paths for the fifth wheel position of the tractor and the semitrailer; a distance δ will develop between them. A linear spring and dashpot are the assumed connection at the fifth wheel as is shown in Figure 4. The force transmitted is then

$$\bar{F} = KFW \cdot \delta + CFW\,\dot{\delta} \qquad (13)$$

where KFW and CFW are constants describing the spring rate and dissipation.

ILLUSTRATION 4

Fifth Wheel Coupling Model



The direction of $\bar{F}$ is assumed to be along a line through the fifth wheel location of the tractor and semitrailer. Note there is no requirement that the parameters KFW and CFW relate to the actual mechanics of the fifth wheel; they must only prevent large displacement between tractor and semitrailer at the fifth wheel. The following are the requirements for the model:

(a) δ must remain small

(b) KFW and/or CFW cannot be large enough to cause unduly high oscillation frequencies in the dynamic system (and thus necessitate shortening the integration time step Δt).

In the present work, the spring rate KFW has been chosen such that, in a hypothetical straight line braking maneuver in which the vehicle is decelerated at 32.2 ft/sec^2 via action of the tractor braking system only, the spring may be expected to deflect less than 1 inch. This criterion is met by setting

$$KFW = (W1 + WS) \text{ lbs/in} \qquad (14)$$

where W1 is the sprung weight of the trailer

WS is the unsprung weight of the trailer.

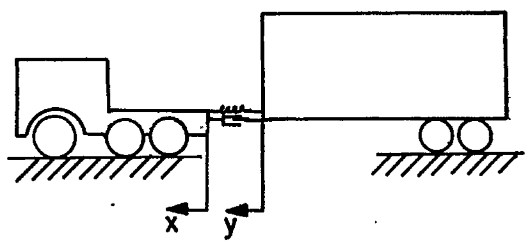
This formulation leads to K values which may be expected to be well within an acceptable range as far as natural frequencies are concerned. (Note that the total spring rate of the tires on the tractor rear axles may be much higher.)

The damping CFW is chosen in the following fashion. Consider the simplified articulated vehicle of Figure 5, again in a straight line maneuver. For the situation with no trailer braking, the equation of longitudinal motion of the trailer may be written

$$\frac{(W1 + WS)}{g}\ddot{y} + (KFW)y + (CFW)\dot{y} = KX + C\dot{X} \qquad (15)$$

ILLUSTRATION 5

Simplified Articulated Vehicle



where W1 + WS is the total weight of the trailer sprung and unsprung masses. Considering the tractor motion as an independent function of time, Equation (15) may be rewritten

$$\ddot{y} + 2\zeta\omega_n\dot{y} + \omega_n^2 y = f(t) \qquad (16)$$

where

$$\zeta = \frac{CFW}{2\left[\frac{KFW}{g}(W1 + WS)\right]^{1/2}} \qquad (17)$$

CFW is chosen such that the dimensionless damping ratio ζ in Equation (16) is set to 0.5. In this fashion, unrealistic transients due to the non-rigid fifth wheel coupling are virtually eliminated.

These methods for the choice of KFW and CFW are non-rigorous and, it would seem, may be susceptible to give erroneous results for some range of vehicle parameters. However, this model has proven very satisfactory in the large variety of vehicles already simulated.

THE MOMENT TRANSMITTED THROUGH THE FIFTH WHEEL

The roll moment, which is assumed to be the product of constant KRM and the difference in roll angles ϕ and ϕ_t of the tractor and semitrailer fifth wheel, is applied along the longitudinal axis of the tractor. This is an approximation since ϕ and ϕ_t are not measured about parallel axes; however, quite reasonable roll moments should be expected for reasonable articulation angles. (Note, a large articulation angle would imply that pitch angles would also be a measure of the roll moment, and thus the present analysis would require modification. It is not, however, the goal of this simulation to deal with large articulation angles; to carefully model the jackknife phenomena to its conclusion requires more sophisticated tire model and fifth wheel model than have been considered in any previous work or will be considered here.)

The restoring moment constant KRM is entirely different in purpose from the "spring rate" KFW. The measure of the "proper" operation of KFW is that $|\delta|$ be small; it seems clear that only the proper fifth wheel force can effect that end. The predicted difference in roll angles between tractor and semitrailer will be quite small, however, independent of the choice of KRM. The value of KRM is chosen not to keep the difference between the roll angles small; rather it is chosen to transmit the proper roll moment across the fifth wheel.

An experimental procedure suitable for the determination of KRM is given in (6). To summarize briefly, a roll moment is applied to the trailer of a tractor-semitrailer combination, and the resultant lateral load transfer for both the tractor and the semitrailer is measured with scales under the wheels. Algebraic calculations then lead to an appropriate KRM value. It should be noted that, to approximate the model of Leucht one would choose KRM = 0, while to approximate the model of Mikulcik, one would choose KRM as large as possible.*

The techniques presented in sections two and three above were developed for use in the articulated vehicle simulation which has been developed at the Highway Safety Research Institute (HSRI) of the University of Michigan. Some details of this simulation and some sample results will be given in the following section.

IV. AN ARTICULATED VEHICLE BRAKING AND HANDLING SIMULATION

The Motor Vehicle Manufacturers Association (MVMA) Truck and Tractor-Trailer Braking and Handling Project was begun at HSRI in mid-1971 with the expressed purpose of establishing a digital computer based mathematical method for predicting the longitudinal and directional response of trucks and tractor-trailers. This simulation is now complete and is in use at various MVMA member companies. Further details of the project are given in (7); it is our purpose here to give only a brief description and some sample results.

The articulated vehicle simulation entails up to 32 degrees of freedom including six degrees of freedom for each sprung mass, two degrees of freedom for each axle for up to five axles, and a spin degree of freedom for up to ten wheels. Key parameters, including tire parameters, were measured by HSRI for use in the simulation (8, 9), and extensive vehicle testing was done to validate the simulation.

The testing used to validate the simulation was performed with the articulated vehicle shown in Figure 6. Some typical results from this validation effort are shown in Figures 7 and 8.

In Figure 7, empirical and simulated results are given for the articulated vehicle in a steady turn. It should be noted that measured steer angles were used in the simulation. These were, as one might expect, significantly different from side to side. For the purposes of Figure 7, average steer angles were plotted. Another slight difficulty is that the test data was taken at speeds slightly different

*Consistent with the integration time step of the simulation.

ILLUSTRATION 6

Articulated Vehicle Used for Testing

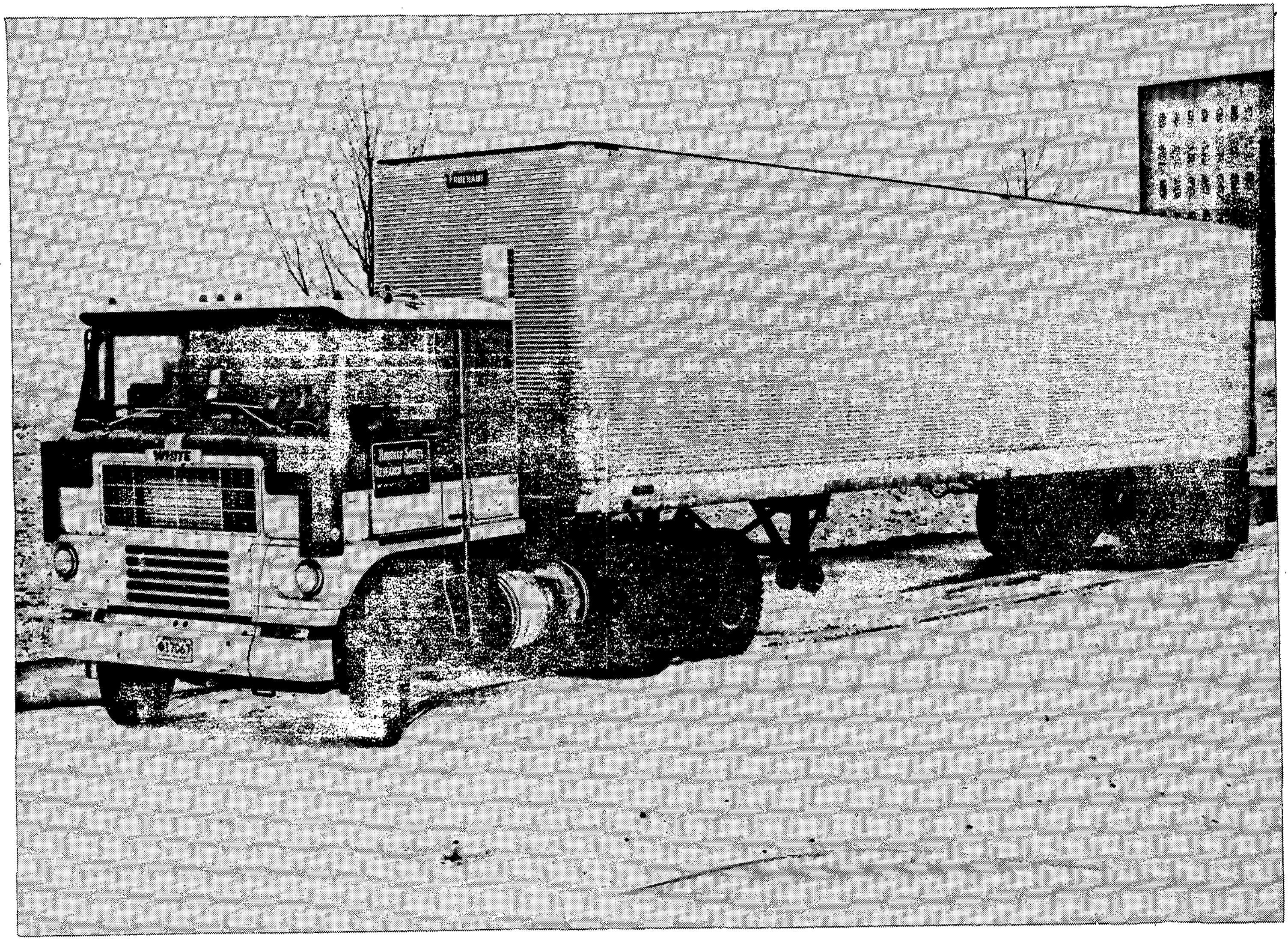


ILLUSTRATION 7

Steady Turn, Empty Trailer, Dry Surface. Simulated Speed: 40 ft/sec

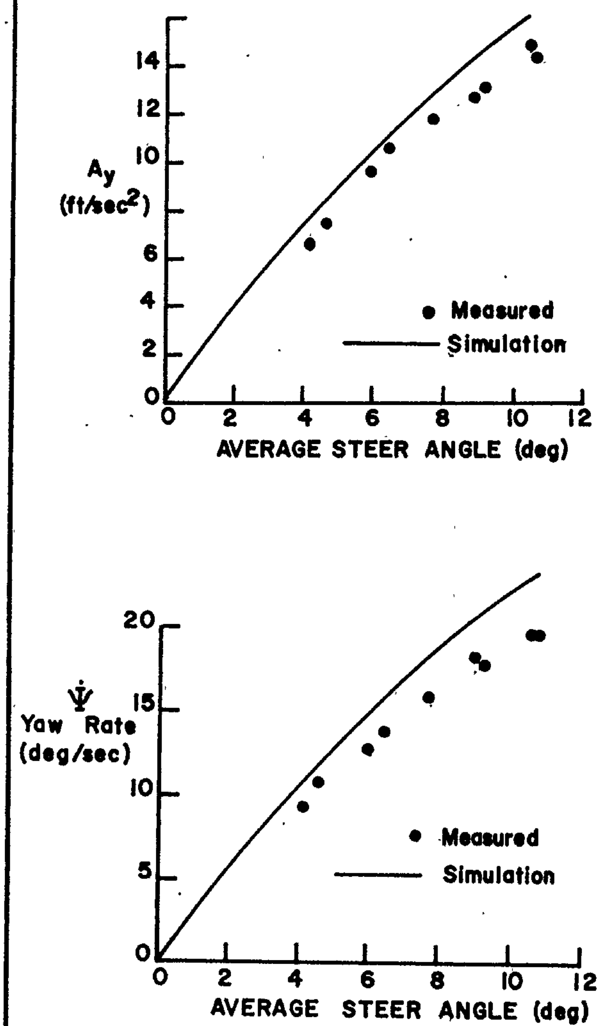
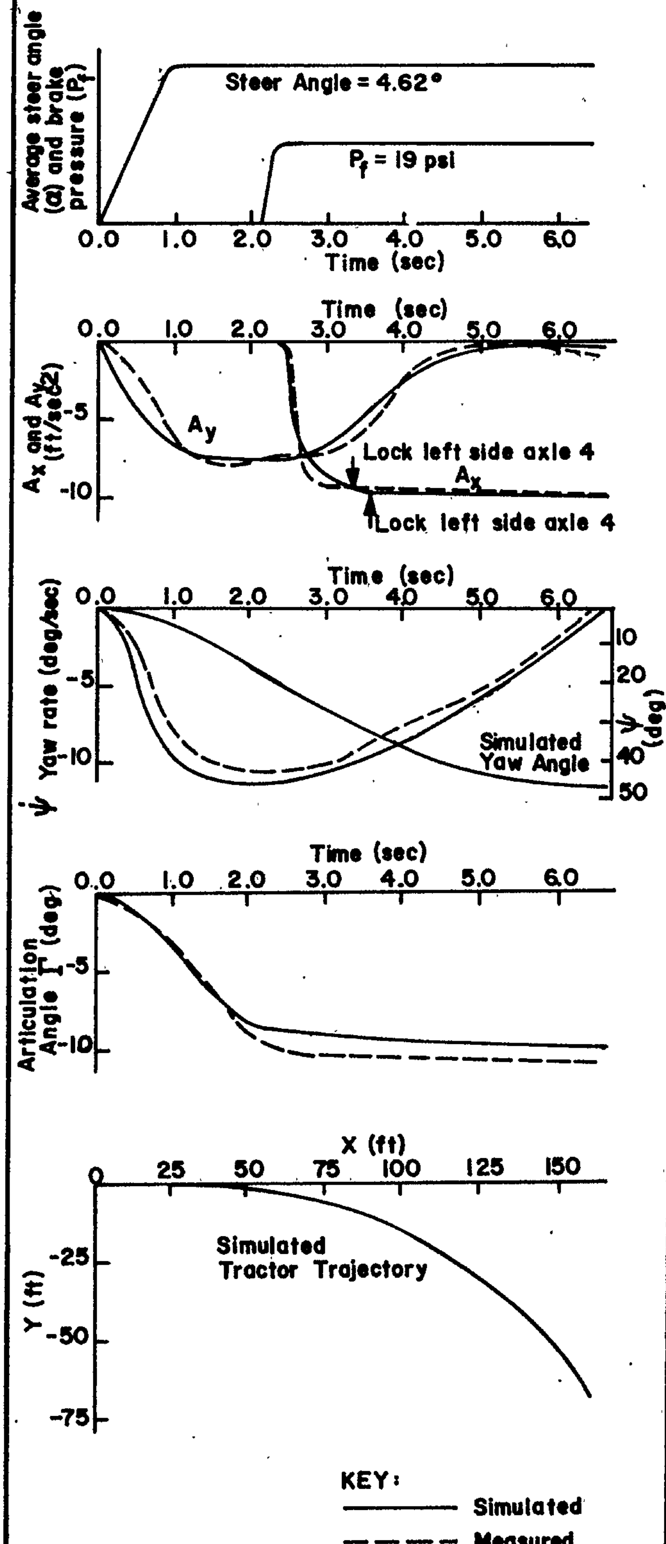


ILLUSTRATION 8

Time History of a Braking-In-A-Turn Maneuver, Empty Trailer, Dry Surface Initial Speed: 30 ft/sec



from the "nominal speed" desired for each test. To facilitate the meaningful superposition of experimental and empirical results in the figure, the average of the speed at which the data was taken was used in the simulation.

In Figure 8, empirical and simulated results are given for a braking-in-a-turn maneuver. As may be seen from the figure, the input steer angle results in a lateral acceleration of about 8 ft/sec^2 (the negative sign on lateral acceleration Ay indicates a left turn), and at two seconds into the turn brakes were applied resulting in a deceleration of about 10 ft/sec^2. Note that wheel lockup on the left side of axle 4, the leading trailer-tandem axle, is accurately predicted.

For the most part, the validation effort was as successful as is indicated in Figures 7 and 8. This work is discussed in detail in Reference 6.

V. CONCLUSIONS

The simulation of articulated vehicles is a complex task. Two potentially troublesome aspects of such a simulation have been discussed in this paper, namely, the simulation of suspensions involving large amounts of coulomb friction, and the use of appropriate constraint equations for the fifth wheel. A method has been suggested for each of these problems, and results indicating the success of the methodology have been presented.

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