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DIRECTIONAL DYNAMICS OF A TRACTOR LOADER BACKHOE

By

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ABSTRACT

DIRECTIONAL DYNAMICS OF A
TRACTOR LOADER BACKHOE

By

Russell Hartley Owen

This thesis presents a study of the directional dynamics of large industrial tractors. These vehicles have special properties which make their dynamics interesting, including soft rear tires, large yaw moments of inertia and low or negative understeer gradients.

A linear yaw plane model was used for the analysis. The lateral compliance of the tires was included via a simplified version of the stretched-string model. Measurements were performed in support of the modeling effort, including inertial parameters, understeer gradient and transient response.

A comparison between calculations and test results showed that lateral compliance is important in the modeling of these vehicles.

To my wife Laurie, whose patient understanding and loving support made this work possible.

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NOMENCLATURE

a	distance from cg to front axle
b	distance from cg to rear axle
cg	center of gravity
C_{α_f}	front tire cornering stiffness
C_{α_r}	rear tire cornering stiffness
F_y	force in the y direction
g	gravitational acceleration (32.2 ft/sec^2)
I_{zz}	Yaw moment of inertia (z axis)
ℓ_o	heighth of cg
m	mass
M_o	applied moment
M_z	moment about z axis
R	radius
r	yaw rate about z axis
$\dot{r}$	time derivative of yaw rate
u	forward velocity (x direction)
v	lateral velocity (y direction)
$\dot{v}$	time derivative of lateral velocity
w	weight
α	slip angle
α_f	front tire slip angle

α_r	rear tire slip angle
$\dot{\alpha}$	time derivative of slip angle
δ	steer angle at tire
σ	relaxation length
θ	pitch angle about y axis
θ_o	equilibrium pitch angle

1.0 INTRODUCTION

This thesis concerns the directional dynamics of a tractor-loader-backhoe (TLB). The TLB is an industrial vehicle with a backhoe on the rear and a bucket loader on the front for excavating and moving soil and other material on the job site. The short wheel base, high moments of intertia and soft rear tires make the dynamics of these vehicles interesting, particularly when commuting in the transport configuration to and from the job site.

The literature contains extensive information on dynamic modeling of farm tractors. Most of this work has been in the area of roll-over and rearward tipping few papers have been published on directional response.

This thesis adapts the linear differential equations commonly used for passenger car studies for use in the study of the TLB. It is shown that a laterally compliant tire model, developed in the automotive and aerospace fields for purposes of modeling high speed effects such as wheel shimmy, is a useful tool in the analysis of TLB directional response at low speeds.

Chapter 2 presents a review of the literature on tractor and industrial vehicle dynamics. Chapter 3 discusses tire modeling and presents a laterally compliant tire model. Chapter 4 deals with the formulation of a simple linear model for tractor directional response. Chapter 5 presents results from tests run on a Ford 555 TLB to determine it's inertial properties, understeer/oversteer gradient, and lateral transient response. Results from a computer simulation of the mathematical model are shown in Chapter 6 and Conclusions and Recommendations are presented in Chapter 7.

2.0 LITERATURE REVIEW

Safety and design considerations have prompted considerable interest in predicting the motions of agricultural and industrial tractors. This section is a review of some of the works published on the rigid body motion of tractors.

Mathematical modeling of tractor dynamics made its debut with McKibben's classic publications in 1927 and 1928 [24, 25]. These still serve as the groundwork for the analysis of rear tipping and weight transfer. Later work in the 1940's and 1950's involved the algebraic evaluation of the stability of farm tractors [23, 36, 57]. These works highlighted the need for further research.

Raney et. al. introduced the use of analog computers to simulate ride properties of tractors in 1961 [32]. This work was followed by similar investigations in 1965 and 1967 [21, 43].

A model for steady state tractor dynamics incorporating the effects of soil properties on traction and rolling resistance was formulated first by Buchele in 1962 [2], and then by Berlage and Buchele in 1966 [1]. Pershing and Yoerger investigated the steady state behavior of road slope mowing vehicles in 1964 [29]. In 1969 they simulated the transient sideslope response using a linear formulation with orthogonal spring-damper systems as tire models [30]. They concluded that ride could be improved with a suspension system and they noted the improvement in calculations of transient response of a nine degree of freedom model as compared to a three degree of freedom model. That same year Unruh, using a similar tire model, formulated equations for an articulated wheel loader with seven degrees of freedom [51]. He simulated side slope stability and presented graphical data of transient response to ground inputs, as well as the vehicle's natural frequencies and mode shapes.

In 1964, Huang et. al. proposed the use of elastic wheel rims to soften the ride of tractors [14]. In 1967, Matthews examined a model using a front suspension system [22], and Smith developed a model incorporating a suspension on both axles [45]. The calculations of both authors indicated improved ride and stability due to the suspensions. Goering and Buchele formulated a nonlinear pitch plane model to examine large amplitude vibrations and rearward overturning [12]. They modeled the radial deflection of the tires using a spring and viscous damper system and an effective roughness approach to provide realistic input.

This growing interest in dynamic modeling produced a need for experimental tire data. Although many works have been published on the tractive performance of tractor tires in soil, very little lateral data was available. This problem was addressed by the publications of Taylor and Birtwistle in 1966 [49], and Schwanghart in 1968 [37]. These form the basis for mathematical tire data used in many subsequent works.

Taylor and Birtwistle describe their experimental work and they present data on the effects of slip angle, camber angle, vertical load, tread pattern and the terrain type on the forces and moments of agricultural tires. Plots of lateral force, overturning moment, aligning moment and rolling resistance are presented for different normal loads.

Schwanghart's work also presents experimental plots of lateral force vs. slip angle for different normal loads. He presents data on rolling resistance, wheel slip and sinkage vs. load for different slip angles and he plots the pressure distributions in the contact patch. One interesting result of this paper is that the lateral force coefficients increase less steeply on soil than on a rigid track, but they attain higher values at larger slip angles. Since sliding is permitted these coefficients are related to frictional properties of the soil, thus they are not cornering stiffness values as used in automotive literature.

The increased availability of fast computers and graphical displays in the seventies led to more sophisticated models and many publications. In 1970 Koch et. al. presented further experimental verification of the nonlinear pitch plane model which Goering and Buchele introduced in 1967 for rearward tipping [16]. In 1971 Gibson et. al. published an investigation of the side-slope stability of logging tractors similar to the work of Pershing and Yoerger in 1964 [11]. Larson and Liljedahl used a mathematical model similar to Unruh's model to study the sideways overturning of row crop tractors [18], and Smith et al. published a paper illustrating the use of vector mechanics to define roll axes for sideways overturning [47].

In 1972, Smith and Liljedahl presented a nonlinear pitch plane model to simulate rearward overturning of farm tractors using a point spring-damper tire model and incorporating power train effects [46]. Thompson et. al. formulated a mathematical model for tire response in the vertical plane to be used with equations for tractor dynamics [50]. They point out that tractors can be statically stable but dynamically unstable. With their model they define an effective base profile to smooth inputs to conventional point follower tire models. Mitchell et. al. used a similar model to develop an automatic control system to prevent rearward overturns [27]. Wolken and Yoerger examined the ride behavior of farm vehicles subject to random inputs using a model similar to that of Pershing and Yoerger in 1969 [56].

In 1973 Krick presented further work with agricultural tires driven in soft ground. He verified that tires developing tractive forces cannot generate as much lateral force as free wheeling tires at the same slip angle [17]. Hudson et al. published a simplified model for simulating dynamic tractor response to trailing loads. He calculated inertia values using the tractor's silhouette and he used linear spring tire models with no slip. This model was used to simulate pitch plane tip behavior while towing implements up slopes [13]. Smith et. al. published a comprehensive work with a three-dimensional nonlinear model [48]. He suggested techniques formerly used to model tractor-semitrailers to simulate towed implements and employed a slip angle tire model using spring rates and cornering coefficients from the works of Krick, Schwang-hart, and Taylor and Birtwistle.

Davis and Rehkugler published a two-part series on agricultural wheel tractor overturns formulating and verifying a mathematical model [7, 8]. Tires were modeled as thin radially deformable disks using radial, lateral and circumferential force coefficients with no lateral deflection. Verification of the math model was done by relating computed results to a scale model study. This mathematical formulation was later distributed as a computer code called SIMTRAC. Gibson and Biller published a work on tip angle calculations of logging tractors and forwarders [10]. Their model assumed that the tractor was rigid and the tires were essentially nondeformable.

In 1976 Rehkugler et al. presented more work on the simulation of tractor overturns using SIMTRAC with published tire data [33]. They modeled tractor overturns on an embankment and verified that surface-tire parameters have a significant effect on overturns. Kelly and Rehkugler presented a paper on the computer graphics display of SIMTRAC runs and proposed its use as a real time simulator for training operators to deal with overturns [15].

In 1977 Masemare and Rehkugler presented results on the influence of tractor geometry and mass on side overturns [20]. They verified that forcing functions for the design of roll-over protection structures (ROPS) were nonlinear functions of tractor mass.

In 1980 Rehkugler formulated a model for simulating the dynamic behavior of articulated-steer, four-wheel drive tractors [34]. He simulated high speed turns on flat soil and concrete using a tire model and data from his work in 1974.

None of these works identify the transient behavior of the laterally compliant tires used on agricultural and industrial tractors. This compliance has a significant effect on the dynamic behavior of these vehicles due to the relatively slow speeds at which they operate. This will be discussed in the next section.

3.0 TIRE MODELING

The majority of the tire models used in the literature on tractor dynamics are based on a quasi-static assumption. They combine tire cornering stiffness and sliding friction to produce one curve for the linear range as well as the non-linear sliding range of slip angles. They do not incorporate transient effects due to lateral compliance in the tires.

Researchers in automotive and aerospace fields have presented several laterally compliant tire models which have been formulated and verified to study the transient lateral response of tires. These models, which were primarily developed to simulate the shimmy phenomenon in aircraft landing gear and automobile tires, have shown that transient lateral force characteristics are determined by the elastic tire properties rather than by frictional effects [9, 26, 28, 39, 40, 44, 52-54]. Transient tire forces are particularly important in low speed dynamics. The effects of lateral compliance are dependent on the forward velocity of the tire because the cornering force build-up is a function of the distance the tire rolls. Thus considerable time lags in force build-up can occur at low velocities.

This effect can be demonstrated with a simple test. A tire is mounted on a test machine at a set slip angle. The tire is then rolled forward at constant speed and its lateral force is measured. The lateral force generated by the tire is plotted versus the distance rolled in Figure 1. Section 3.2 will show that the lag in force build-up can be determined by the elastic tire properties and the distance rolled as characterized by the so-called relaxation length [31]. It has been shown that passenger car tires do not approach effectively instantaneous response until their speed reaches 32 Km/hr and this compliance exhibits considerable significance at speeds around 16 Km/hr [42].

Investigations of wheel shimmy verify this delay between the lateral force and the steering input. It has been shown that this delay increases with steering rate and decreases with forward velocity [41, 42].

3.1 The "Stretched-String" Model

Von Schlippe and Dietrich formulated the "stretched-string" tire model to simulate lateral compliance in 1941 [52]. The equatorial centerline of the tire was modeled as a massless circular string which was elastically connected to the center plane of the wheel and constrained in the circumferential direction. Tension is placed on the string by a uniform radial force distribution to simulate the effects of the inflation pressure. The string has a finite contact length. Figure 2 presents a diagram of this tire model.

Many variations of this model have been proposed [40]. The effects of finite mass of the string have been examined [39, 28]. This enhances the ability of the model to simulate shimmy at moderate frequencies. The string has been replaced with a beam

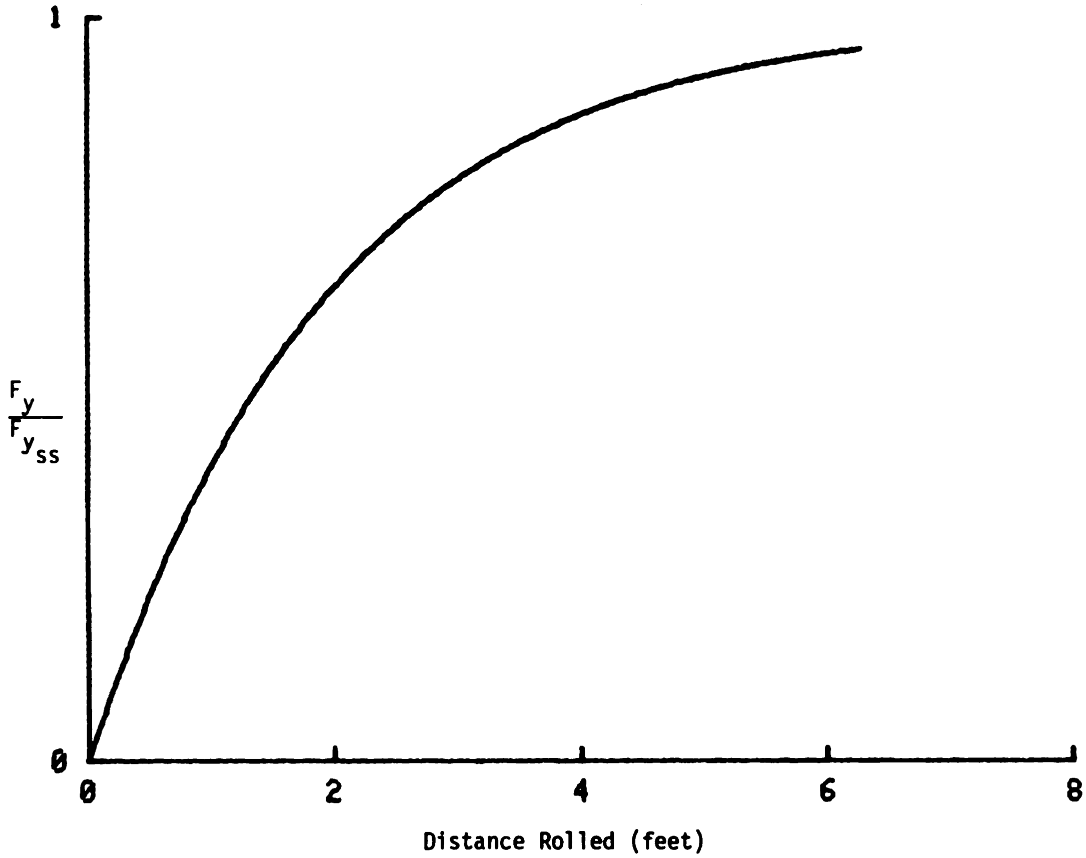

Figure 1. Lateral force buildup for a loaded tire rolling at a fixed slip angle.

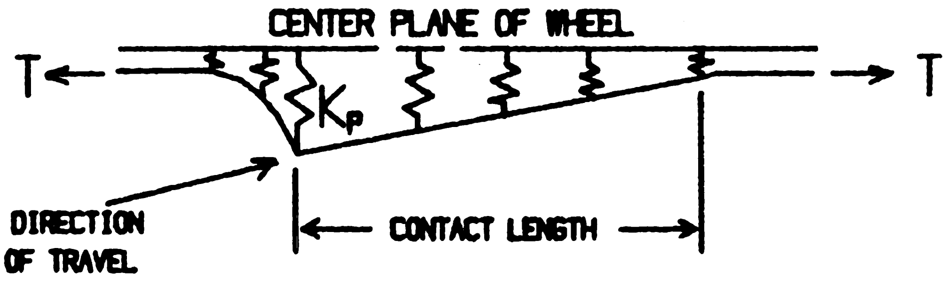

Figure 2. Diagram of the stretched-string model, view of contact patch from above.

to model the stiffness in the tire carcass, allowing for partial sliding at the rear of the contact patch. This gives the tire a continuous slope [9]. Some investigators have given the tire model a finite width by using a thick beam or by using parallel string models to simulate the effects of contact width [28, 53, 54]. But all of these formulations lead to the same general result, that a length parameter related to the undeformed tire and the length of the contact patch are significant factors. The length parameter, which is based on the undeformed tire, will be discussed in the next section.

3.2 Relaxation Length

The relaxation length σ has been defined as (see figure 2):

$$\sigma = \frac{T}{K_p} \tag{1}$$

where T is the longitudinal tension in the tread and K_p is the lateral pneumatic stiffness per unit arc length. It can be shown that σ represents the distance a loaded tire with a slip angle must roll to attain approximately 63% of its steady state lateral force [31].

A great deal of testing has been done to determine the values of relaxation length for aircraft and automotive tires [26, 42, 31, 6]. Some of the methods involve obtaining a plot of lateral force versus distance rolled, as in Figure 1. The lateral force values are divided by the steady state force and this ratio can be plotted versus distance. If this result is plotted on semi-log paper, the relaxation length is the slope of the line and it represents the distance rolled for 63.2% of full force [26]. Other investigators use the 63.2% length and solve the equations of the string model for σ, which results in a lower value [31]. Still other investigators apply a sinusoidal steer input with varied frequency and perform Fourier analysis on the input and output signals to calculate σ. Von Schlippe and Dietrich predicted that σ would be 60% to 90% of the tire radius. Most experimental studies indicate that relaxation lengths for automotive and aircraft tires average approximately one-sixth of the circumference [52, 39].

3.3 The Mathematical Tire Model

Equations of motion for the stretched-string mode have been derived many times in the literature [5, 26, 31, 38, 41, 44, 52]. The model is shown in Figure 2. The derivation is based on two assumptions, namely, (1) a no-slip condition between the contact line and the road surface so the contact line has the same shape as the wheel path and (2) the portion of the string outside of the contact patch rolls into the contact patch in a continuous manner. This provides for a continuous slope at the front of the contact patch. Two equations result, one applies for distances less than the contact length and the other equation applies for distances greater than the contact length.

The equations are:

$$F_y(x) = -2\ K_p\ \alpha_o[(\ell+\sigma)x - x^2/4] \qquad 0 < x < 2\ell \qquad (2a)$$

$$F_y(x) = -2\ K_p\ \alpha_o[(\ell+\sigma)^2 - \sigma^2 e - \frac{(x-2\ell)}{\sigma} \qquad x > 2\ell \qquad (2b)$$

where

F_y = lateral force

K_p = lateral pneumatic stiffness per unit arc length

ℓ = half contact length

x = distance rolled

α_o = kinematic slip angle

σ = relaxation length

As x approaches infinity this model yields the steady state side force.

$$F_y(\infty) = -2\ K_p\ \alpha_o(\ell+\sigma)^2 \qquad (3)$$

The combination $2\ K_p(\ell+\sigma)^2$ is often called the cornering stiffness and is usually indicated by the parameter $C\alpha$.

If it is assumed that contact length can be neglected without a significant loss in accuracy, the following simplified model results:

$$F_y(x) = -2\ K_p\ \alpha_o[\sigma^2 - \sigma^2 e^{-x/\sigma}] \qquad (4)$$

The time derivative of this equation is:

$$\dot{F}_y = -2\ K_p\ \alpha_o\ \sigma\ u\ e^{-x/\sigma} \tag{5}$$

where

$$u = dx/dt \tag{6}$$

and the dot indicates differentiation with respect to time. Using this result, the simplified model can now be written as:

$$F_y(t) = \frac{\sigma}{u}\ \dot{F}_y - C_\alpha \alpha(t) \tag{7}$$

or

$$F_y(t) = \frac{\sigma}{u}\ C_\alpha \dot{\alpha}(t) - C_\alpha \alpha(t) \tag{8}$$

This model is a good approximation at low values of $\dot{\alpha}$ [41].

4.0 A MODEL FOR TRACTOR DIRECTIONAL RESPONSE

A simplified directional response model will be useful to investigate the effects of lateral compliance. Small angular excursions are assumed in the pitch and roll planes and the forward velocity of the tractor is assumed constant. These assumptions allow the mathematical development of a linear yaw plane model.

A simple way to visualize this model is to picture a bicycle which cannot tip over and whose wheels cannot leave the ground. The characteristics of the front and rear tires are combined into single tires as on a bicycle. For linear range maneuvers, this model provides good insight into the transient and steady state directional response of the tractor.

Figure 3 presents a free-body diagram. The state variables and parameters are defined in the nomenclature listing. The equations of motion are:

$$m(\dot{v} + ur) = \Sigma F_y \tag{9}$$

$$I_{zz}\dot{r} = \Sigma M_z \tag{10}$$

The force and moment summations are functions of the slip angle α. For traditional linear models, the relaxation length is assumed

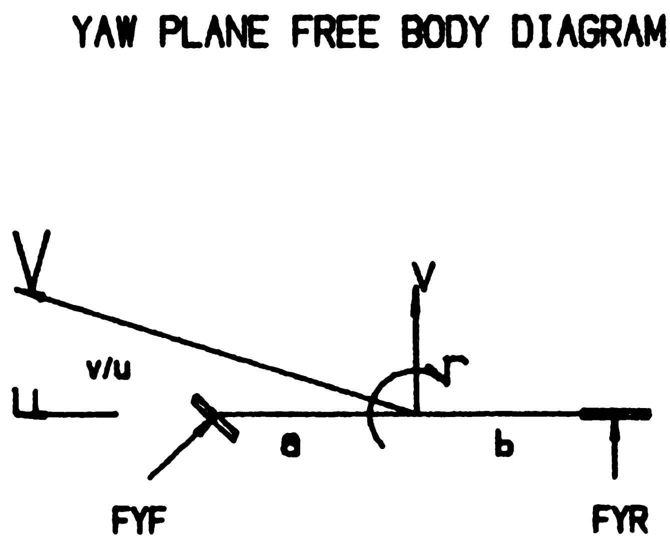

Figure 3. Yaw plane free body diagram of the bicycle model for directional response.

to be negligibly small, and the lateral forces and the moments are given by:

$$\Sigma F_y = -C_{\alpha_f}\alpha_f - C_{\alpha_r}\alpha_r \tag{11}$$

$$\Sigma M_z = -C_{\alpha_f}\alpha_f a + C_{\alpha_r}\alpha_r b \tag{12}$$

where

$$\alpha_f = \frac{v}{u} + \frac{ar}{u} - \delta \tag{13}$$

$$\alpha_r = \frac{v}{u} - \frac{br}{u} \tag{14}$$

Equations 9 and 10 can now be rewritten

$$\dot{v} = -ur - \frac{C_{\alpha_f}\alpha_f}{m} - \frac{C_{\alpha_r}\alpha_r}{m} \tag{15}$$

$$\dot{r} = -\frac{aC_{\alpha_f}\alpha_f}{I_{zz}} + \frac{bC_{\alpha_r}\alpha_r}{I_{zz}} \tag{16}$$

Equations 13 and 14 indicate that the slip angles are a function of the kinematics of vehicle motion. If the effects of lateral compliance are incorporated into this model, the slip angles are derived from equations of the following form, as discussed in Chapter 3.

$$\dot{\alpha} = \frac{u}{\sigma}(\alpha_o - \alpha) \tag{17}$$

where α_o is the kinematic slip angle, as indicated by equations 13 or 14, and α is the new compliant slip angle. Incorporating compliance into the model yields

$$\dot{v} = ur - \frac{C_{\alpha_f}\alpha_f}{m} - \frac{C_{\alpha_r}\alpha_r}{m} \tag{18}$$

$$\dot{r} = - \frac{C_{\alpha_f}\alpha_f a}{I_{zz}} + \frac{C_{\alpha_r}\alpha_r b}{I_{zz}} \tag{19}$$

$$\dot{\alpha}_f = \frac{1}{\sigma}\,(v + ar - u\alpha_f - u\delta) \tag{20}$$

$$\dot{\alpha}_r = \frac{1}{\sigma}\,(v - br - u\alpha_r) \tag{21}$$

The next two chapters will show that the use of a compliant tire model is a crucial component of successfully simulating tractor directional response.

5.0 TRACTOR TESTING

Several tests were conducted to provide vehicle parameters for the directional response model. The vehicle tested was a Ford model 555 tractor-loader-backhoe. Tests were run to determine inertial properties, the understeer/oversteer gradient and lateral acceleration of the vehicle in several linear range step steer maneuvers. These tests are described in the following sections.

5.1 Inertial Properties

The tractor was tested for its inertial properties at the Highway Safety Research Institute of the University of Michigan. Details of the test methodology are presented in Reference 55.

The longitudinal position of the center of gravity was determined by supporting the vehicle by its frame rails on knife edges and finding its balance point. The lateral position was assumed to be on the midplane of the vehicle.

The heighth of the center of gravity was determined by supporting the TLB by knife edges at the longitudinal c.g. position and applying a known pitch moment. The angle induced by the pitch moment yields the desired height. The height ℓ_0 of the center of gravity is:

$$\ell_o = \frac{M_o}{w} \operatorname{ctn} (\theta_i - \theta_o) \tag{22}$$

where

M_o = applied moment

w = weight of vehicle

θ_i = tip angle

θ_o = equilibrium angle

The pitch moment of inertia was determined using a large pendulum-type swing. The TLB was driven on to the swing and the system was allowed to come to rest. A small oscillation was introduced and the period of the swing was measured. The period of the oscillation yields the desired moment of inertia. The results of these tests are presented in Table 1.

TABLE 1

INERTIAL PROPERTIES OF THE FORD 555 TLB

Wheelbase	6 ft. 8 in.
c.g. position	
aft of front axle	65.32 $\pm$ .06 in.
height above ground	40.85 $\pm$ .098 in.
Pitch moment of inertia	262130 $\pm$ 1038 in-lb-sec^2
Weight	14675 lbs.

5.2 Understeer/Oversteer Gradient

An excellent discussion of the understeer/oversteer gradient is presented in Reference 3. Some of that information will be summarized here as an introduction to a discussion of the test procedure.

The understeer/oversteer gradient is an important measure related to vehicle directional response. An intuitive sense of the understeer/oversteer gradient can be gained by considering a vehicle in a steady turn at a steady speed. The vehicle is said to be understeer if an increase in velocity requires an increase in steer angle to remain on the same radius turn. Oversteer vehicles, on the other hand, require reduced steer to remain on the same radius when the velocity is increased. The transition from understeer to oversteer, so-called neutral steer, denotes vehicles which remain at the same radius when speed is increased. The understeer/oversteer gradient is the term used to quantify this property.

Tests were run on the TLB to determine its understeer/oversteer gradient. The tractor was driven in a steady turn with the steering system clamped at a fixed steer angle. This clamp was necessary because the hydraulic steering system would not hold a constant steer angle due to fluid bleeding within the steering cylinders. The radius R of the turn was measured at low velocity. The velocity was then increased and radius measurements were taken for each velocity.

Figure 4 presents a plot of the test data in the form $1/R$ vs. lateral acceleration U^2/R. The understeer/oversteer gradient

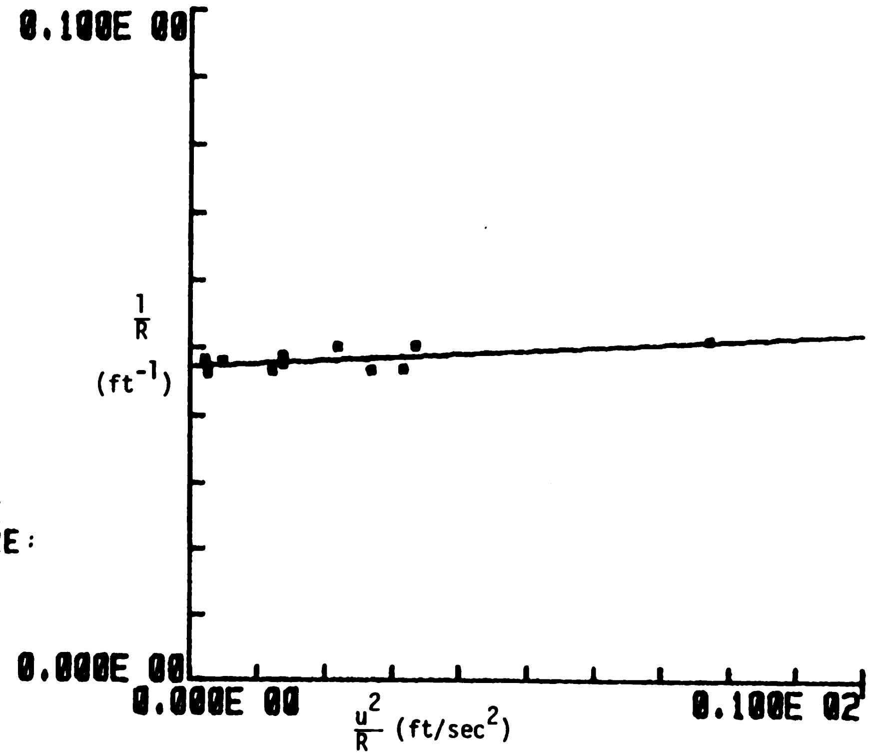

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Figure 4. Measured data from constant steer angle test.

may be calculated based on the slope of this line [19]. The understeer/ oversteer of this tractor was determined to be -0.19 degrees per g, a nominal amount of oversteer.

5.3 Transient Testing

Transient maneuvers were performed with the vehicle in order to further study it's directional response. The maneuver used in the test is commonly called a "J turn". The vehicle travels straight at constant speed and then turns sharply, holding a constant steer angle. This test condition approximates, as closely as possible, a step-steer input.

The lateral acceleration was recorded by a Schaevitz servo-accelerometer, model LSBC-1, mounted close to the c.g. of the tractor. The output of the accelerometer was recorded on a Bruel-Kjaer, model 7003, FM tape recorder mounted in the tractor. The data was retrieved with a Hewlett Packard stripchart oscillograph. Several step-turns were recorded and the data is presented in Figures 5 and 6.

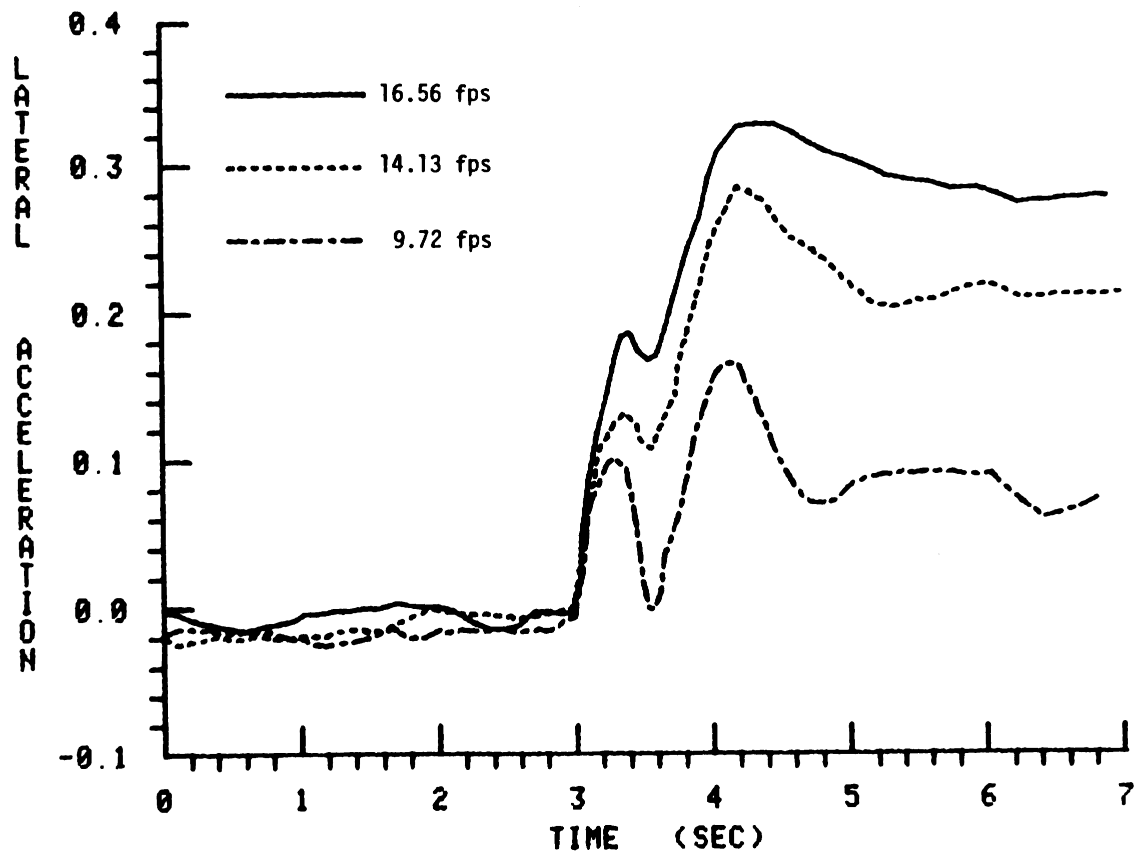

Figure 5. Step turn lateral acceleration data, steer angle held to approximately 14 degrees.

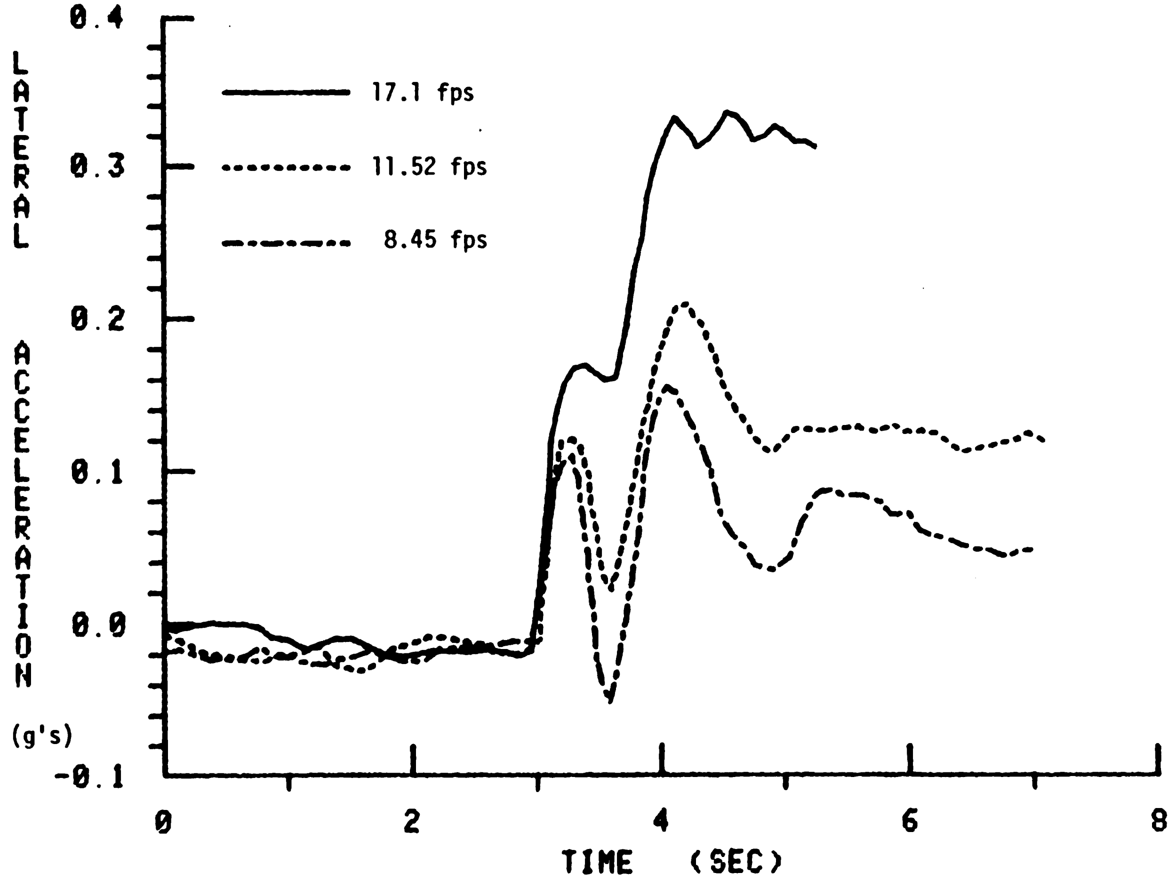

Figure 6. Step turn lateral acceleration data, steer angle held to approximately 14 degrees.

6.0 SIMULATION RESULTS

Several computer simulations were conducted to elucidate the directional response of the TLB. The linear model presented in Chapter 4 was integrated numerically using the HPCG software of the Case Center for Computer-Aided Design [4].

Most input parameters were acquired or deduced from the test data presented in Chapter 5. Measured wheelbase and c.g. position were entered directly. The yaw moment of inertia was assumed to be equal to the measured pitch moment of inertia. Due to the lack of published tire data for agricultural or industrial tractors, the tire parameters were estimated. The understeer/oversteer gradient determined in Chapter 5 was used to formulate a linear relationship between the front and rear cornering stiffness, as shown in Figure 7. The TLB used in the tests was equipped with 11.0L x 16, 10 ply, Goodyear tires, inflated to 44 psi, on the front axle. The rear tires were Armstrong 16.9 x 28, 8 ply, heavy duty tractor lugs inflated to 24 psi.

Based on a visual inspection of the front tires, which had the characteristics of small truck tires rather than passenger car tires, and the plot of Figure 7, a value of 250 lb/deg was selected for each front tire. Figure 7 then yielded 1013 lb/deg for each rear tire. The implications of changes in this choice

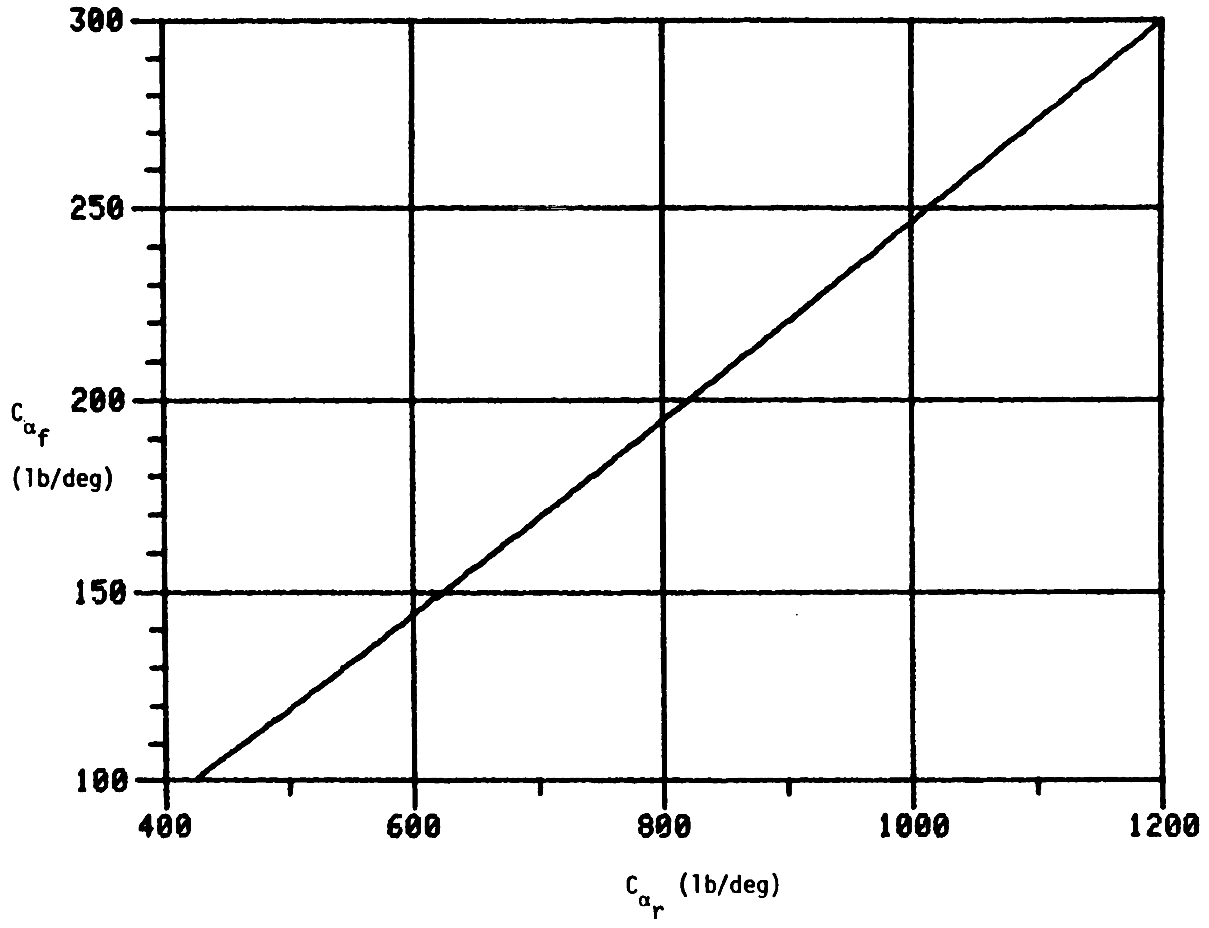

Figure 7. Front cornering stiffness vs. rear cornering stiffness for Ford 555 TLB (per tire).

will be discussed in subsequent sections. The relaxation length of the front tires was assumed to be 60% of their radius or 0.78 ft. The rear tires were simulated with various relaxation lengths.

A summary of the input data is presented in Table 2. Computed results are presented both for step-steer and sinusoidal steer simulations, and design changes are recommended.

6.1 Step-Steer Results

Step-steer maneuvers, simulated at speeds of 8, 20 and 30 fps, are shown in Figures 8 through 10. In each case, results are presented for three rear relaxation lengths, 0.0, 5.0, and 10.0 feet. A qualitative comparison of these results and the test results in Figures 5 and 6 indicates that the rigid tire model, denoted by zero relaxation length, is not useful for modeling the transient phase of the maneuver. The laterally compliant model carries most of the important information regarding the nature of the response, with the best fit resulting from a relaxation length of 5 feet. In particular, the simulations show the same oscillatory character as the low speed tests, with good amplitude and frequency correlations. Figure 10 indicates that, as the speed increases, the relaxation length diminishes in importance.

6.2 Sinusoidal Steer

Calculations were also made for sinusoidal-steering, or lane change maneuvers. The steering input, which has a frequency equal to the apparent characteristic frequency in Figure 8, is shown in Figure 11. Figures 12 through 14 present lateral

TABLE 2

INPUT PARAMETERS FOR SIMULATION RESULTS

Figure no.	C_{α_f} lb/deg per tire	C_{α_r} lb/deg per tire	14° input	σ_f(ft)	σ_r(ft)	Velocity (fps)
8	250	1013	step	varied		8
9	250	1013	step	varied		20
10	250	1013	step	varied		30
12	250	1013	Sinusoid	varied		8
13	250	1013	Sinusoid	varied		20
14	250	1013	Sinusoid	varied		30
15	varied	1013	step	.78	5	8
16	varied	1013	Sinusoid	.78	5	8
17	varied	varied	step	.78	5	8
18	varied	varied	Sinusoid	.78	5	8

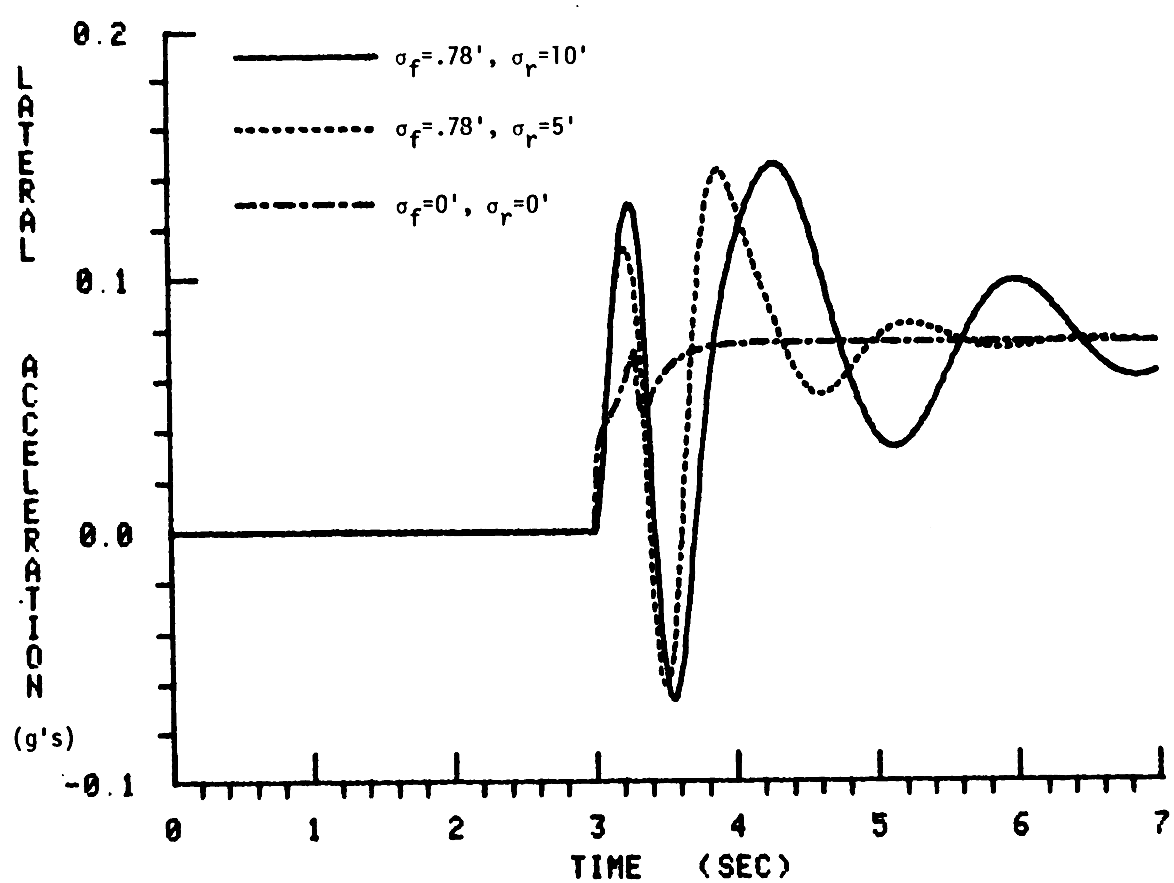

Figure 8. Step steer response, 8 fps, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 14 degree steer angle.

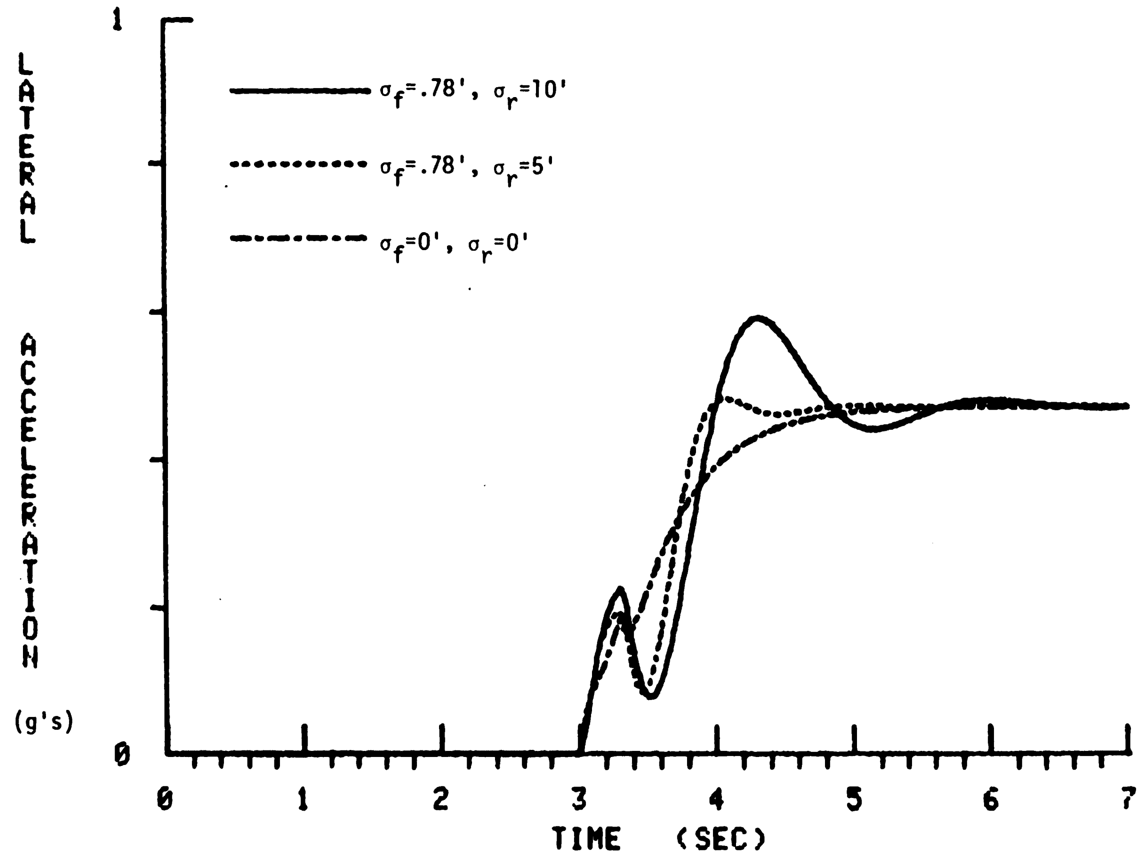

Figure 9. Step steer response, 20 fps, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 14 degree steer angle.

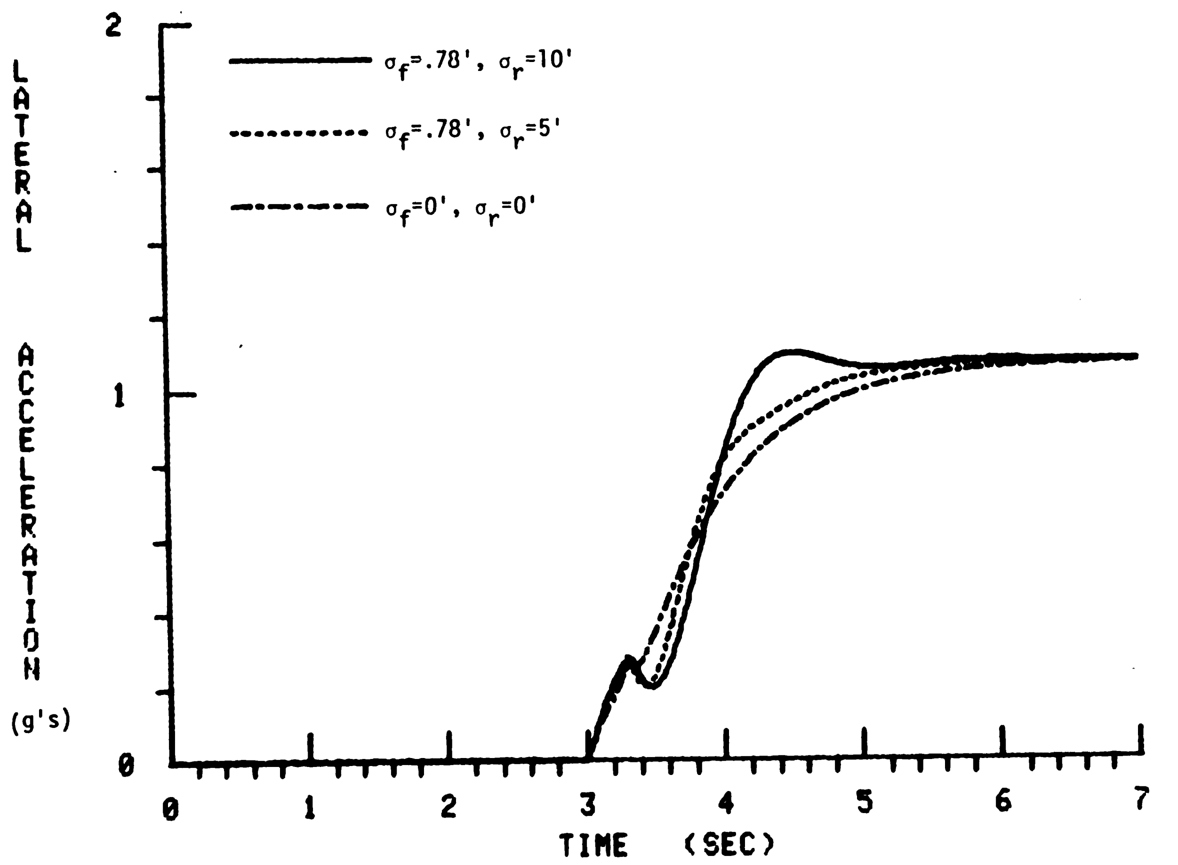

Figure 10. Step steer response, 30 fps, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 14 degree steer angle.

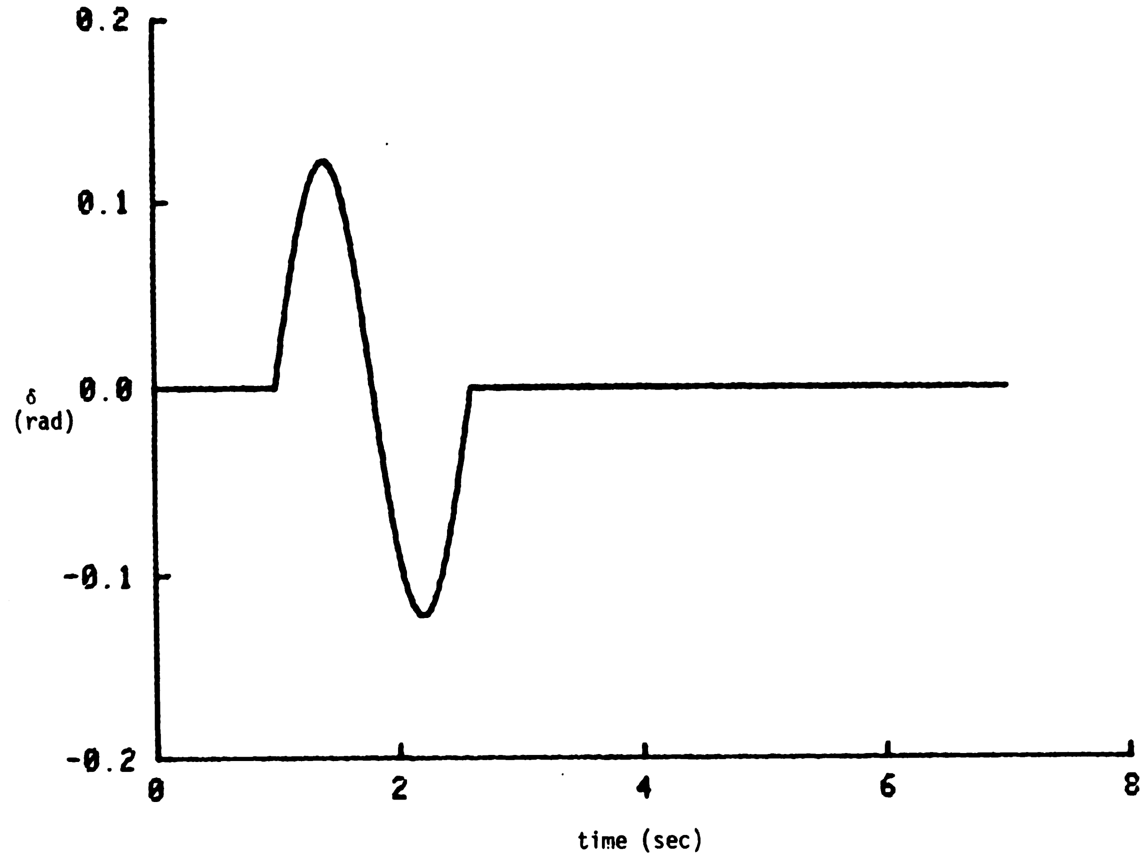

Figure 11. Sinusoidal steer input.

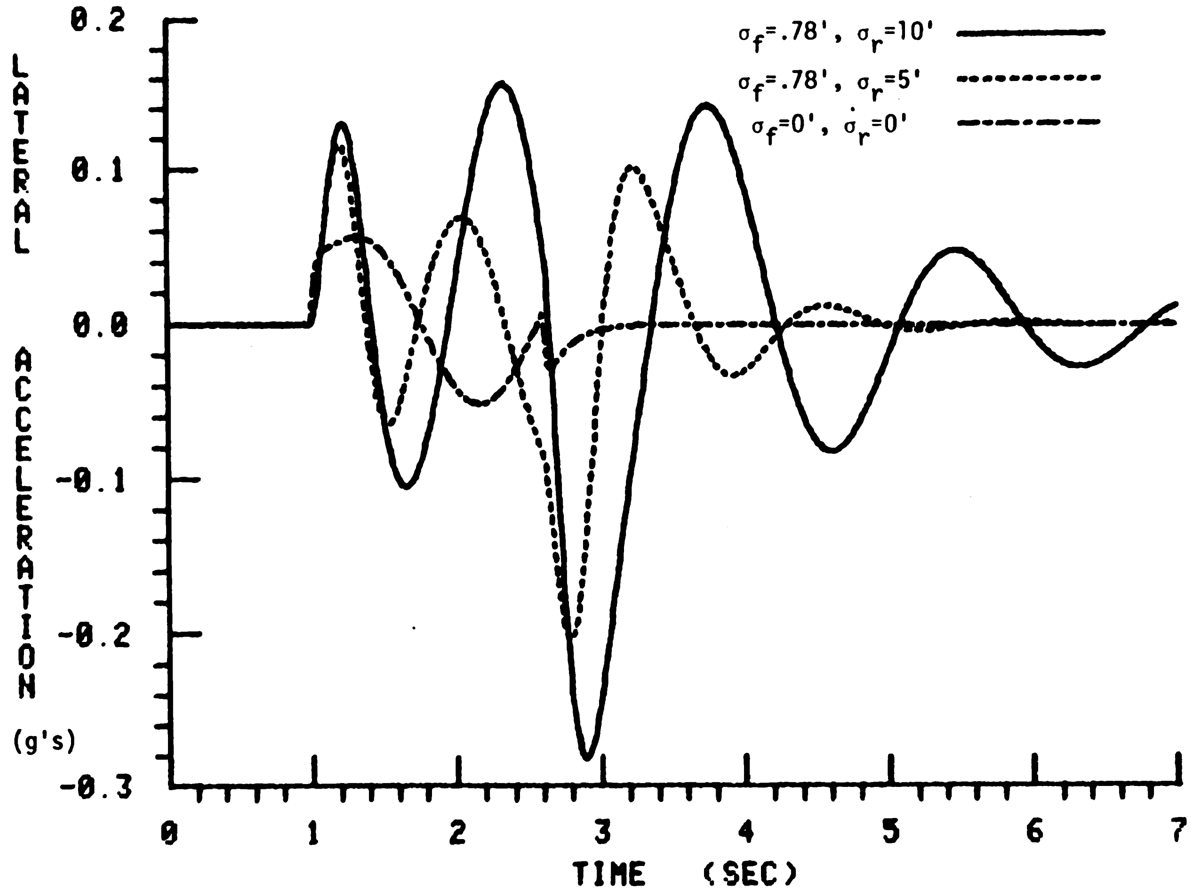

Figure 12. Sinusoidal steer response, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 8 fps.

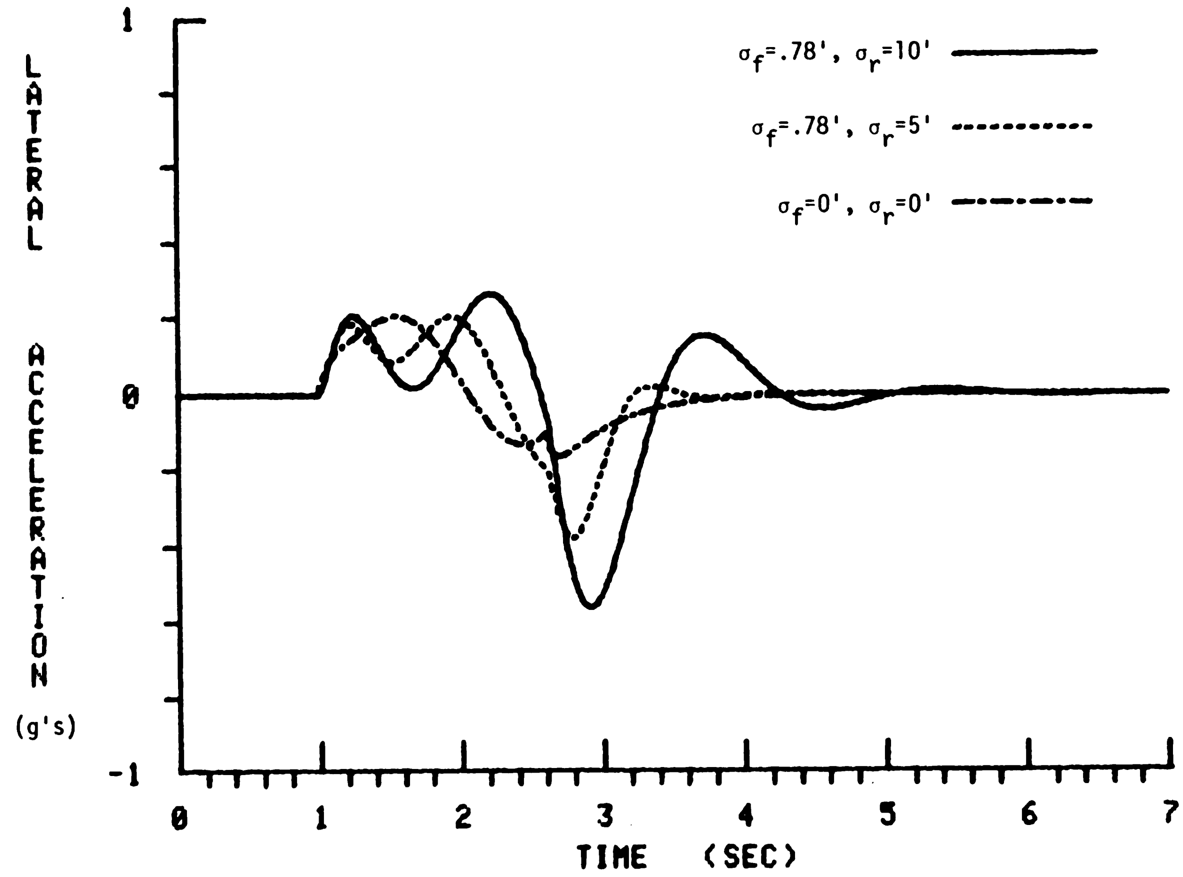

Figure 13. Sinusoidal steer response, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 20 fps.

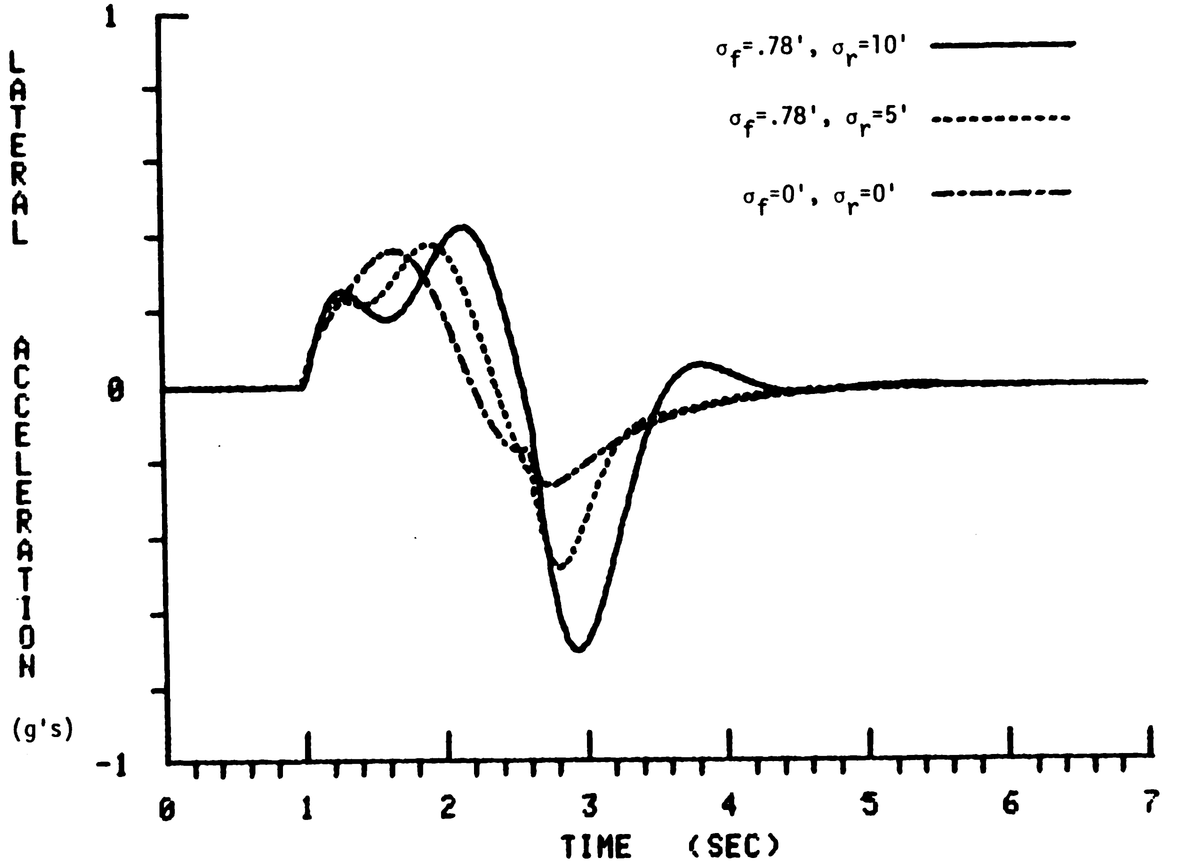

Figure 14. Sinusoidal steer response, C_{α_f}=250 lb/deg per tire, C_{α_r}=1013 lb/deg per tire, 30 fps.

acceleration results for three different rear relaxation lengths at 8, 20 and 30 fps. Again the results of the rigid model significantly differ from the laterally compliant model as low speeds, and the differences become less at higher speeds.

The simulation can be used to elucidate the effect of design changes which might be considered to improve the directional response of the tractor. An important step in this direction would be to increase understeer. This could be accomplished through structural changes which would move the center of gravity forward. Design changes of this magnitude are expensive and not very practical. An easier method to increase the understeer would be to reduce the cornering stiffness of the front tires.

Figure 15 shows a comparison of step steer response for 2 configurations, both with rear relaxation lengths of 5 ft., but one with the front C_α reduced to 100 lb/deg. This figure shows marked improvement in the directional response. Figure 16 shows similar improvement for the sinusoidal steer response.

6.3 Some Comments on Tire Parameters

The previous calculations have been based on tire parameters which appear to be quite speculative. However, given the measured -0.19 deg/g understeer/oversteer gradient, fairly large changes in the tire cornering stiffness magnitudes do not change the character of the results. Figure 17 presents step-steer results with different cornering stiffness pairs obtained from Figure 7. These plots illustrate similar behavior through a wide range of

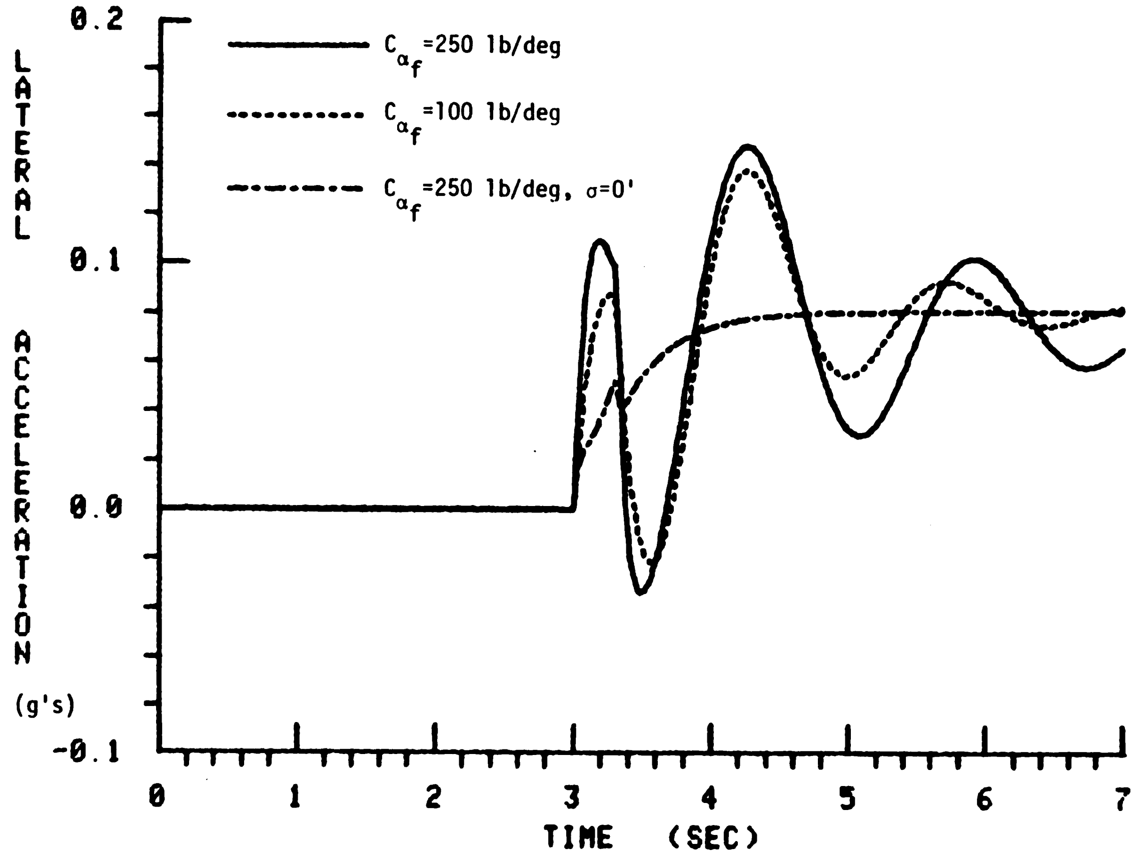

Figure 15. Step turn response comparison with soft front tires, C_{α_r}=1013 lb/deg per tire, 8 fps, σ_f=.78', σ_r=5'.

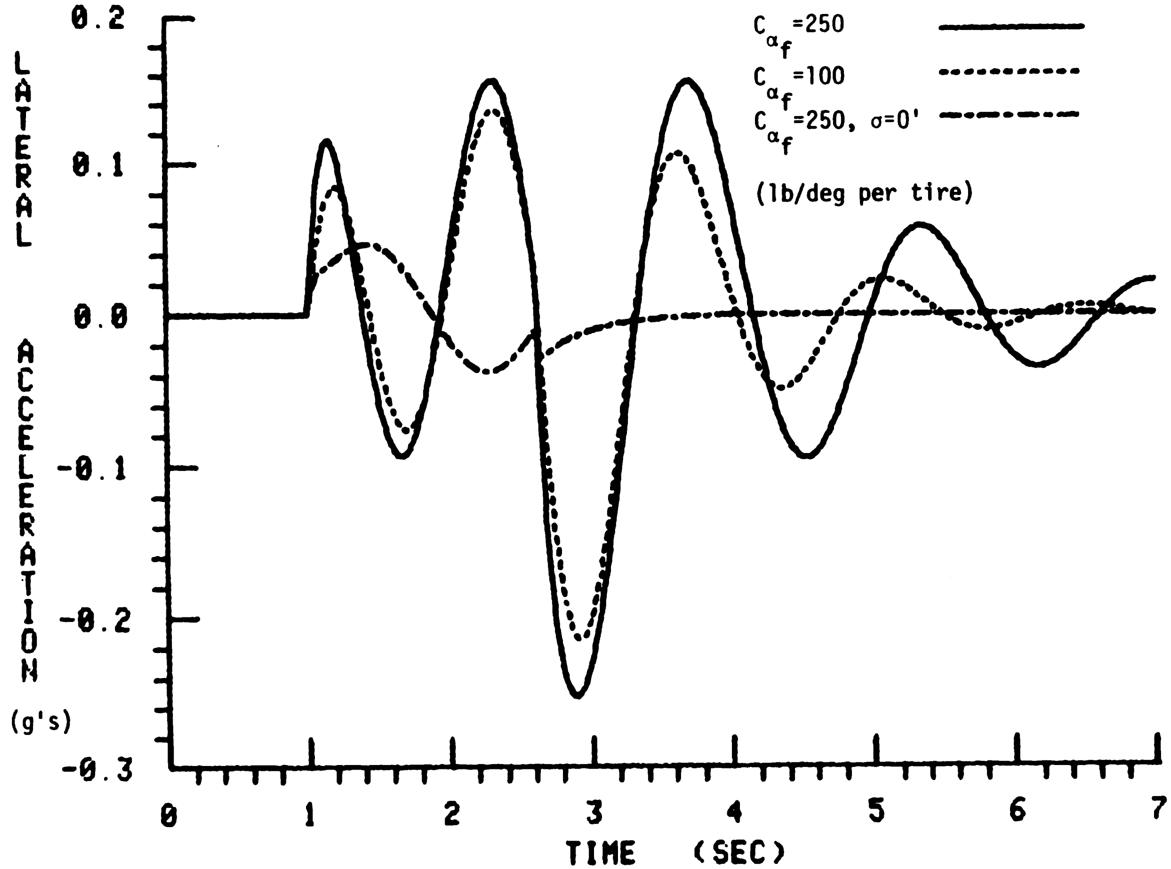

Figure 16. Sinusoidal steer response, comparison with soft front tires, C_{α_r}=1013 lb/deg per tire, 8 fps, σ_f=.78', σ_r=5'.

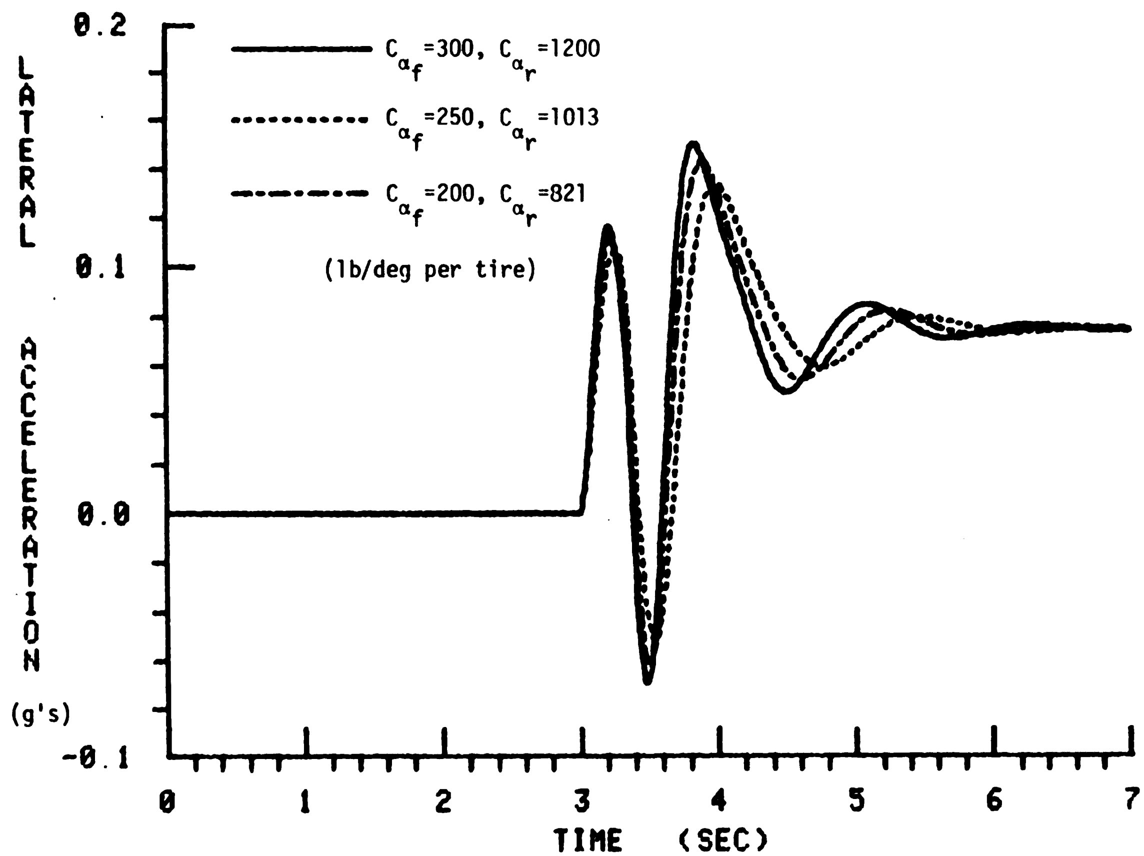

Figure 17. Step steer response for various tire stiffness pairs, 8 fps, 14° steer, σ_f=.78', σ_r=5'.

C_α values, with the same steady state result. Figure 18 shows the same trend for a sinusoidal steer input.

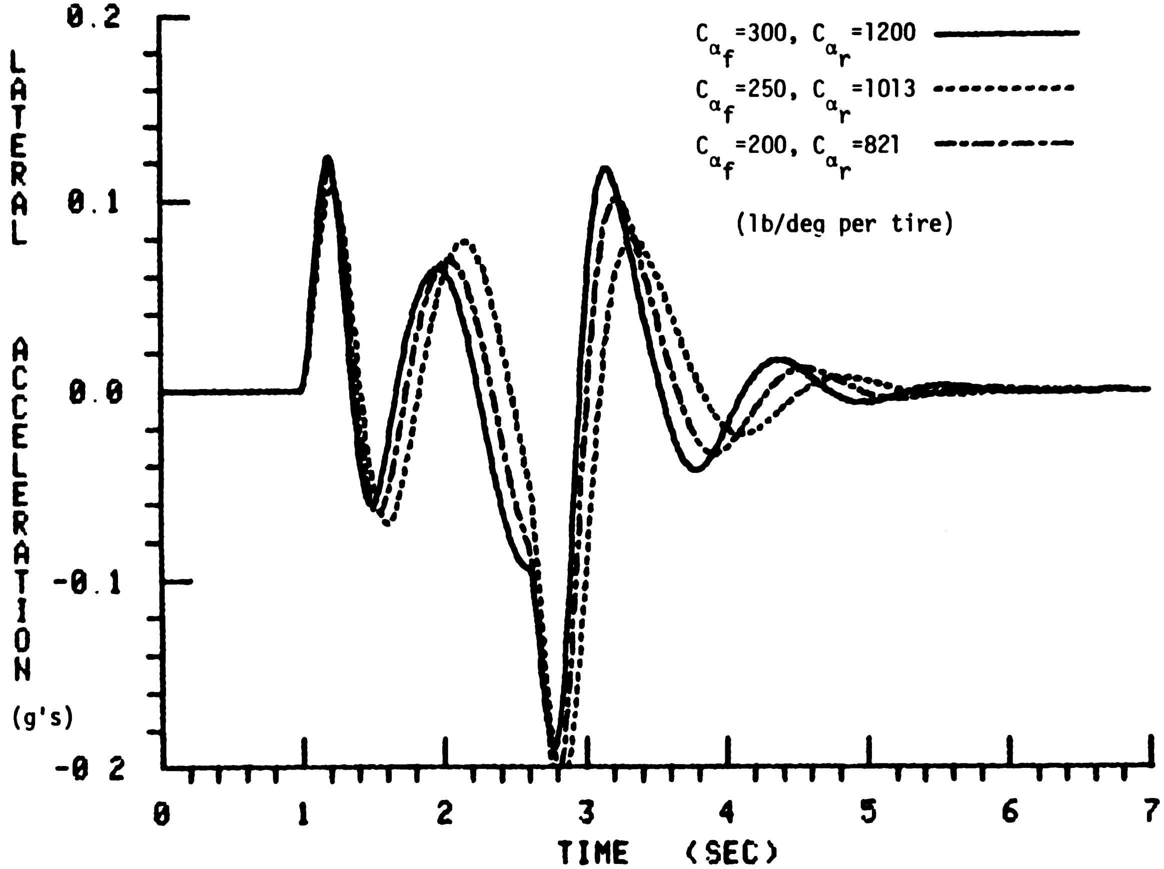

Figure 18. Sinusoidal steer response for various stiffness pairs, 8 fps, σ_f=.78', σ_r=5'.

7.0 CONCLUSIONS AND RECOMMENDATIONS

Two conclusions are justified based on the results presented in this thesis:

1) A compliant tire model is necessary for successful simulation of the TLB.

2) The directional response of the TLB is improved with softer front tires.

Some recommendations are also in order.

Further work is needed to improve the compliant tire model for this application. The model should also allow partial sliding along the entire contact surface for simulating tires on soil, and should deal with the evidence that the relaxation length varies with vertical deflection [26].

Tire testing is needed to determine relaxation lengths and cornering stiffness parameters for agricultural and industrial tractor tires. The need for testing of this type is highlighted by the importance of transient tire effects in the dynamics of these vehicles.

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