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A study of the cornering stability of a heavy-duty truck rear suspension with and without roll steer characteristics

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A STUDY OF THE CORNERING STABILITY OF A HEAVY-DUTY TRUCK REAR SUSPENSION WITH AND WITHOUT ROLL STEER CHARACTERISTICS

by

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CERTIFICATE OF APPROVAL

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(date)

Professor in Charge

Chairman of the Department

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My friends at Mack Trucks, Inc.

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ABSTRACT

An investigation of the cornering stability of a typical heavy-duty tractor and semi-trailer combination was performed. Two tractor rear suspension types were evaluated. One type exhibits strong roll steer characteristics, using a combination of three leading and three trailing links to locate the tractor's two drive axles. The second type uses (six) trailing links to locate the axles, and does not have strong roll steer tendencies. Cornering stability behavior due to changes in the location of the suspension links relative to the tractor frame were also investigated. A computer model created for the Automatic Dynamic Analysis of Mechanical Systems (ADAMS) software package from Mechanical Dynamics, Incorporated (MDI) was used for the analysis.

The cornering stability limit for a tractor+trailer combination can be defined as the threshold at which a tractor goes from an understeer cornering mode to one of oversteer. Once the tractor is oversteering it may either jackknife or have the trailer swing out from the intended path. By employing the Handling Diagram technique, developed by Pacejka, this threshold can be determined from a plot of vehicle lateral acceleration vs. the difference between the front tire and rear tire slip angles. It is the point where the plot goes through an inflection point.

The results of the analysis showed that the rear suspension with strong roll steer characteristics yielded higher levels of lateral acceleration before the cornering stability limit was exceeded. This was true for all linkage angles, though this threshold level decreased as the link angle increased.

The model employed for the analysis was very simple in such areas as tractor front suspension, steering and tire forces. The results indicate that a more refined model could also be used to study stability questions during the early phases of a suspension development cycle and reduce the extent of prototype testing required.

INTRODUCTION

The turning characteristics of a typical tractor + semi-trailer combination were investigated to determine the effects of tractor suspension on the vehicle dynamic stability.

A computer model of a three axle tractor pulling a loaded single axle trailer at 62 miles per hour was used for the study. This set of parameters was chosen both to be able to utilize an existing computer simulation model with a minimum of modifications, and as a vehicle combination and speed that is common on interstate highways.

The particular turning characteristic that was studied is the ability of a tractor to maintain an understeer handling mode as lateral acceleration is increased. Understeer can be described as a vehicle's tendancy to point towards the outside of the path of a turn, also known as "plowing". The opposite of understeer is oversteer. This is the tendancy of a vehicle to point towards the inside of the path of a turn. The transition of a tractor from understeer to oversteer has been shown to be a requirement for the onset of the directional instabilities known as tractor/trailer jacknifing and trailer-swing.[1] (Note: Please refer to the Glossary in Appendix A for further definitions of these and other terms.) As long as the tractor remains in the understeer regime the combined vehicle retains directional stability. Overall stability may be lost due to the vehicle rolling over, but it has been shown that roll over generally occurs at levels of lateral acceleration above the onset of oversteer (oversteer threshold) and was not pursued.[2] Other investigations have studied rollover instability in detail[3], with a variation of the computer model used as the basis for this study coming from such an investigation.

The directional stability of a heavy-duty truck during turning manuevers is greatly influenced by the design of the tractor's rear suspension. There are several styles of heavy-duty truck suspensions currently marketed for highway operation. These suspensions are typically composed of combinations of springs and kinematic linkages, which are used to locate the tractor drive (powered) axle(s). The combination of the suspension and the drive axle(s) is referred to as the bogie. When viewed from the side the kinematic linkages typically are simple four-bar mechanisms. One of the links of the mechanism may even be the leaf spring used to support the axle's load.

This study will look at some of the effects of this linkage's design on the lateral acceleration level at the oversteer threshold. Only bogies with two drive axles will be modeled. Two types of bogie suspensions will be compared: The first type has an axle's links located in front of that axle, as shown in Figure 1. The second type has the links for both axles pointed towards a center point, as shown in Figure 2. Both types of suspensions are currently used on vehicles designed and built in the United States.

The bogie suspension shown in Figure 1 maintains the drive axles roughly parallel to each other as the chassis rolls about its longitudinal axis as in Figure 3. By maintaining the relative axle alignment any roll about the longitudinal axis has a relatively small effect on the orientation of the tire, and thus on vehicle tracking. This type of suspension will be termed a "non-roll steer suspension".

The bogie suspension shown in Figure 2 is such that the angle between the drive axles varies as the vehicle rolls about its longitudinal axis as in Figure 4. By allowing this, the drive axles contribute some turning effort to the overall vehicle, rather than providing only a resistance to lateral slipping.This type of suspension will be termed a suspension with roll steer characteristics, or a "roll steer suspension".

The oversteer threshold can be modified by varying the suspension linkage geometry within the two groups defined above. The height of the frame attachment points above the connecting points on the axles will be used as a model parameter to demonstrate this concept. (Note: This dimension is a variable in heavy duty truck design, influenced by such factors as vehicle ground clearance and allowable bending stresses in the chassis frame.)

There are also other factors that affect the tractor during roll. An example of these factors is the weight transfer to the outside tires, causing the frame to flex and the tires to deflect far beyond their linear range. Additional factors beyond the vehicle's mechanical design include variables such as driver steering input and road conditions. All of these were not considered as part of this investigation.

The Automatic Dynamic Analysis of Mechanical Systems (ADAMS) kinematic/dynamic analysis software by Mechanical Dynamics, Inc. (MDI) was used for this study. It was chosen for its ease of use and flexibility. ADAMS allowed the vehicle to be modeled as a interconnected grouping of discrete parts, avoiding the tedious process of defining the equations of motion for the system as a continuum. Any individual part could then be modified as required. ADAMS also allows FORTRAN subroutines to be linked to the model analysis, allowing the definition of relationships that are beyond the capability of a less robust program.

NON-ROLL STEER SUSPENSION

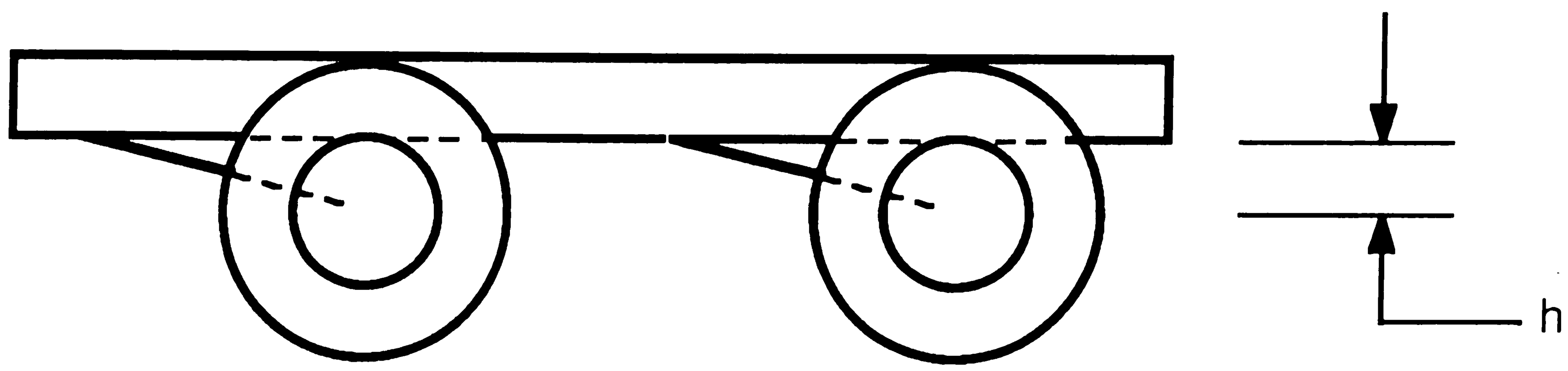


Figure 1

ROLL STEER SUSPENSION

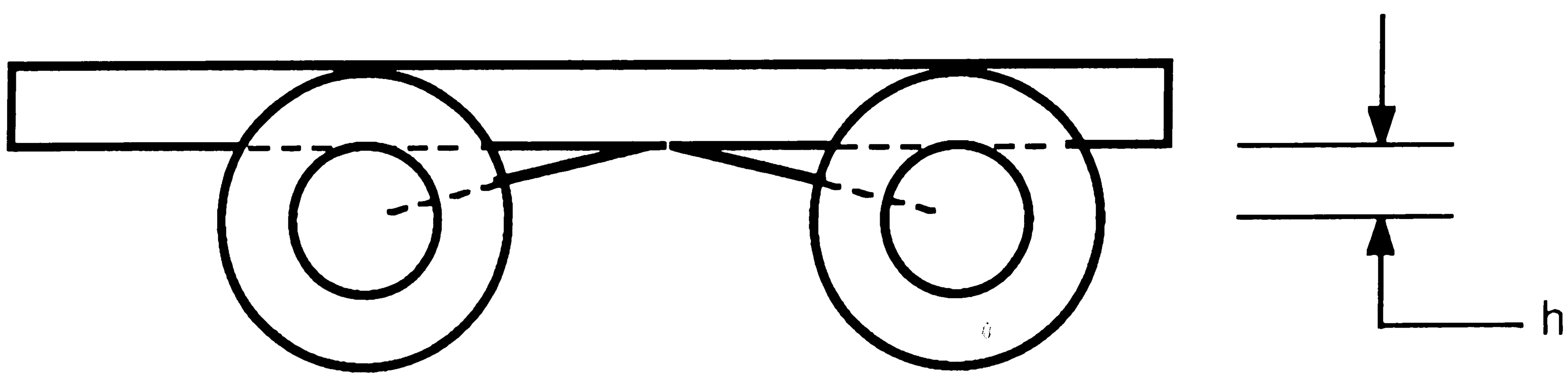


Figure 2

Vehicle Roll with a
"Non-Roll Steer Suspension"

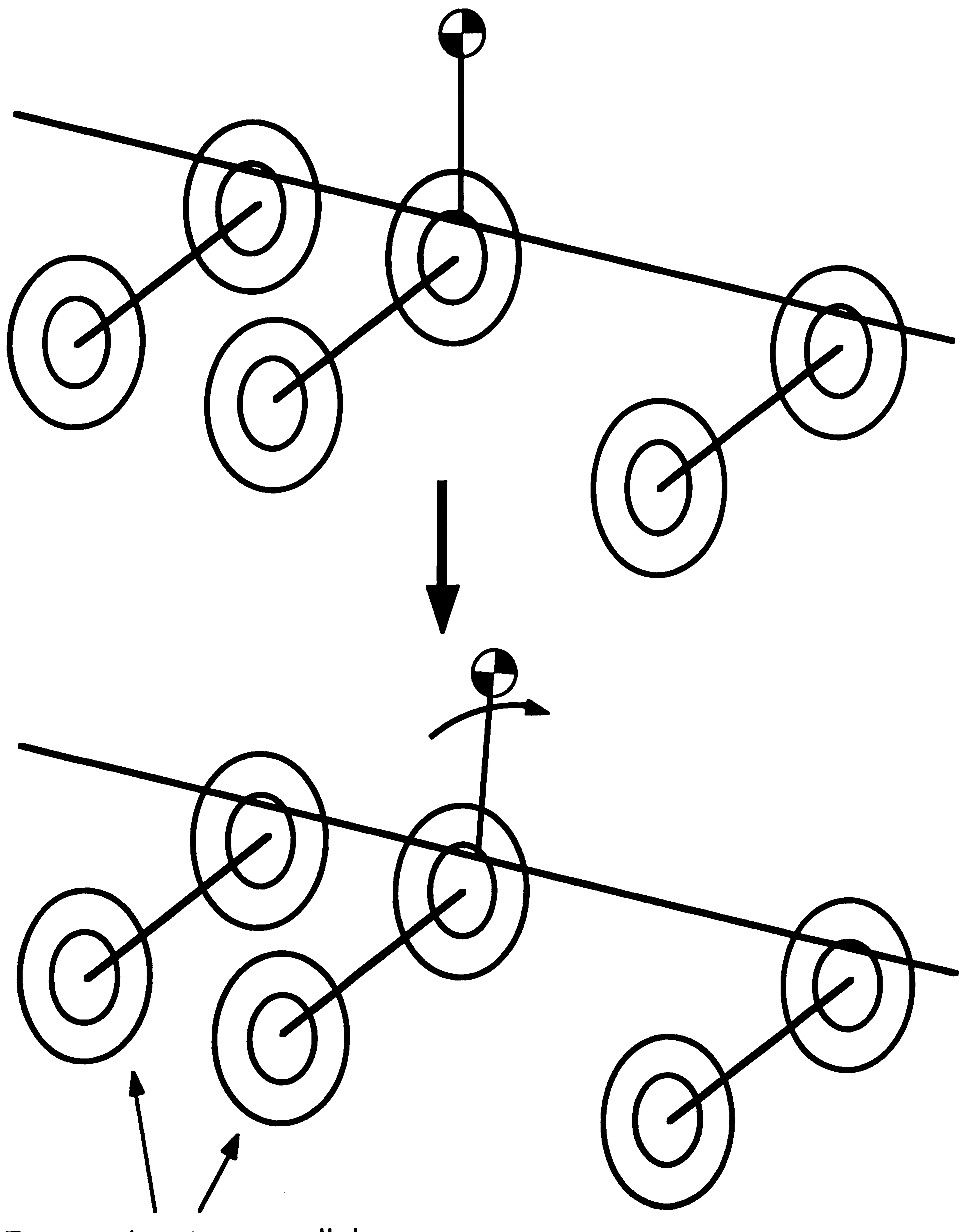


Figure 3

Vehicle Roll with a
"Roll Steer Suspension"

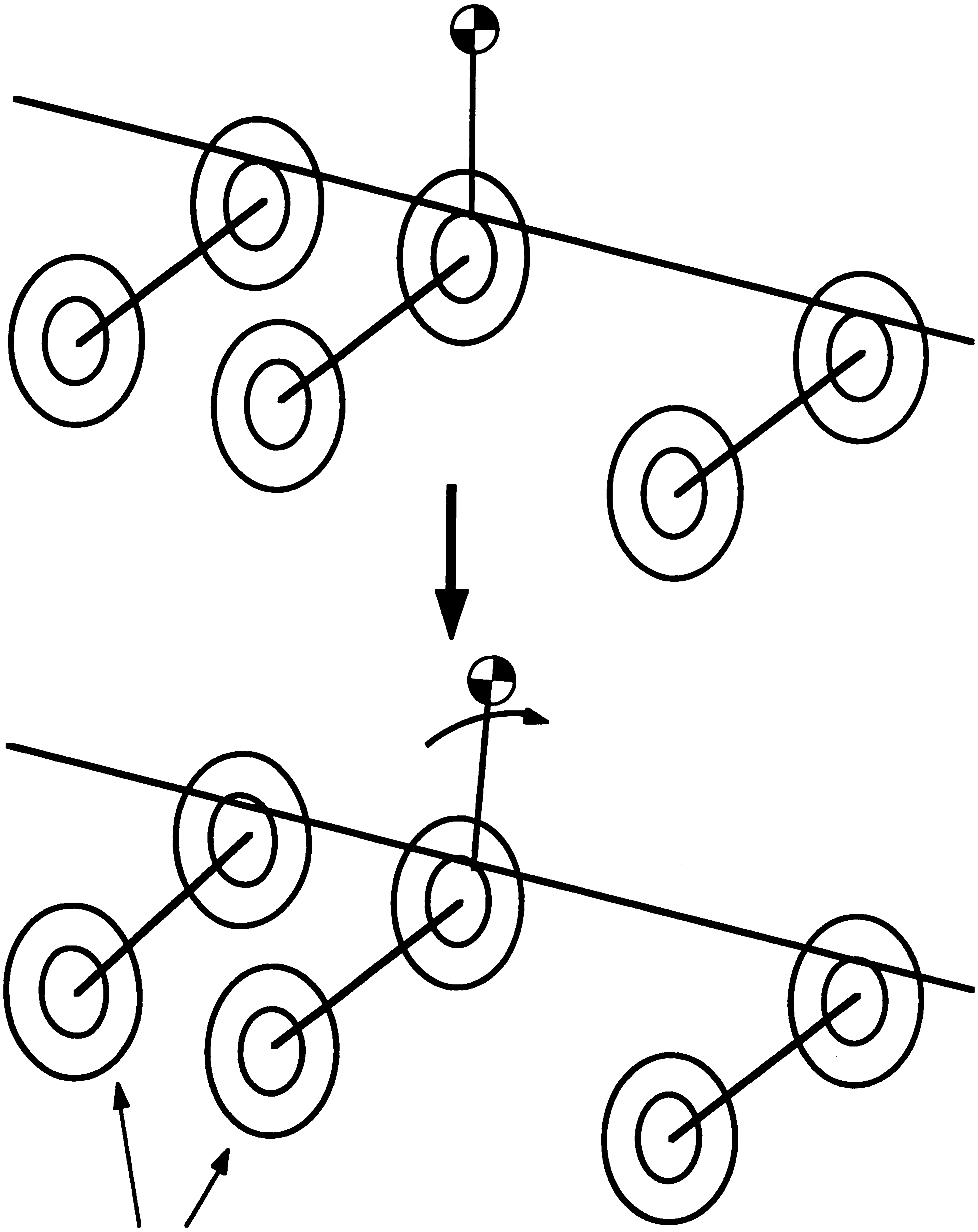


Figure 4

BACKGROUND

The modern heavy-duty truck and trailer combination vehicle is a common sight on the highways and urban areas in North America. These vehicles are seen in various combinations: a tractor (the section containing the driver, controls and powertrain, often referred to as the "truck") and a 40 to 48 foot trailer, a tractor and two 27 foot trailers, and a tractor and a single 27 foot trailer. They are used to haul nearly any type of freight, from light packages to cement. The weights of these vehicles can approach 80,000 lbs. when loaded . Because of their large loaded masses and the increasingly smaller size of the typical automobile the dynamic characteristics of these vehicles is a subject of more than casual scrutiny. The particular dynamic characteristic under investigation in this study is the cornering ability of a tractor/trailer.

A technique for investigating the handling (cornering) behavior for any wheeled vehicle known as the "handling diagram" was developed by H.B. Pacejka[4] . The handling diagram is a plot of the difference of slip angles vs. lateral acceleration for a given vehicle, where "difference of slip angles" is the absolute value of the difference between the slip angle of the tractor front tires and the tractor rear tires. The slip angle is the angle between the center plane of the tire and its true velocity. The difference between the slip angles for the tires on the same axle are usually small, and will be assumed to be negligible unless otherwise stated. An example of a typical handling diagram is shown in Figure 5. The rate of change of the difference between the slip angle is also an indication of the level of understeer for a vehicle.[5] The slope of the curve is the rate of change of the amount of understeer present with respect to lateral acceleration, with negative slope indicating increasing understeer. Thus the level of lateral acceleration where the slope of the handling diagram goes from negative to positive is the point at which the vehicle from an understeer to an oversteer condition.[6] It should be noted that the curve's slope will not always go through such a transition. A vehicle may show increasing levels of understeer as the lateral acceleration increases, as shown in Figure 6.

It has been shown that "for any form of dynamic instability to occur the tractor must be in oversteer."[7] The dynamic instabilities being referred to are jack-knifing and trailer swing. Thus the level of lateral acceleration at which the transition from understeer to oversteer occurs is of a prime importance. If the transition occurs below the vehicles rollover threshold, at a level of lateral acceleration that could be expected to occur in typical operation, then the handling of that vehicle would be questionable. (NOTE: It has also been shown that for a 6x4 tractor + trailer combination there is not a single handling curve, but rather a family of curves that define the handling characteristics for the vehicle.[8] An example of this is shown in Figure 7. Each curve would be valid for a given forward velocity. This added variable was not explored in the work presented here. All comparisons were made at identical values of forward velocity.)

The testing of actual tractor + trailer vehicles to determine their handling diagrams would be a very expensive and time consuming process. Analytical models can be used to reduce the amount of physical testing that is required. The scope of this investigation was limited to such an analysis.

As previously noted, the ADAMS kinematic and dynamic analysis package was used for the computer simulation of a tractor/trailer vehicle. ADAMS proved to be an especially useful software package for this investigation. It treats a mechanism as a series of rigid parts, each with its own mass properties (mass, moments of inertia and products of inertia). These parts are then interrelated by a combination of kinematic constraints and forces. The constraints can range from from one that will require two points (in different parts) to be coincident, to one that simulates a Hooke (universal) joint. Similarly, forces may be simple linear relationships or complete FORTRAN subroutines that represent the forces and moments created at the tire/road interface. The creation of these forces and constraints is simplified by the ability to define a new coordinate system at any point in space, with any desired orientation. These points may fixed with respect to the global coordinate system, or be free to move with one of the mechanism's parts.

The kinematic and dynamic constraints combine with prescribed motions and initial conditions to produce motion in the mechanism. This motion may be referenced to a global coordinate system or a coordinate system fixed to an individual part (Local Part Reference Frame or LPRF).

The output from an ADAMS analysis can be the position of a part, its velocity or its acceleration, as a function of time. The forces maintaining the kinematic constraints can also be requested. Finally, this information can be used as input to a FORTRAN subroutine to determine more complex relationships.

TYPICAL HANDLING DIAGRAM

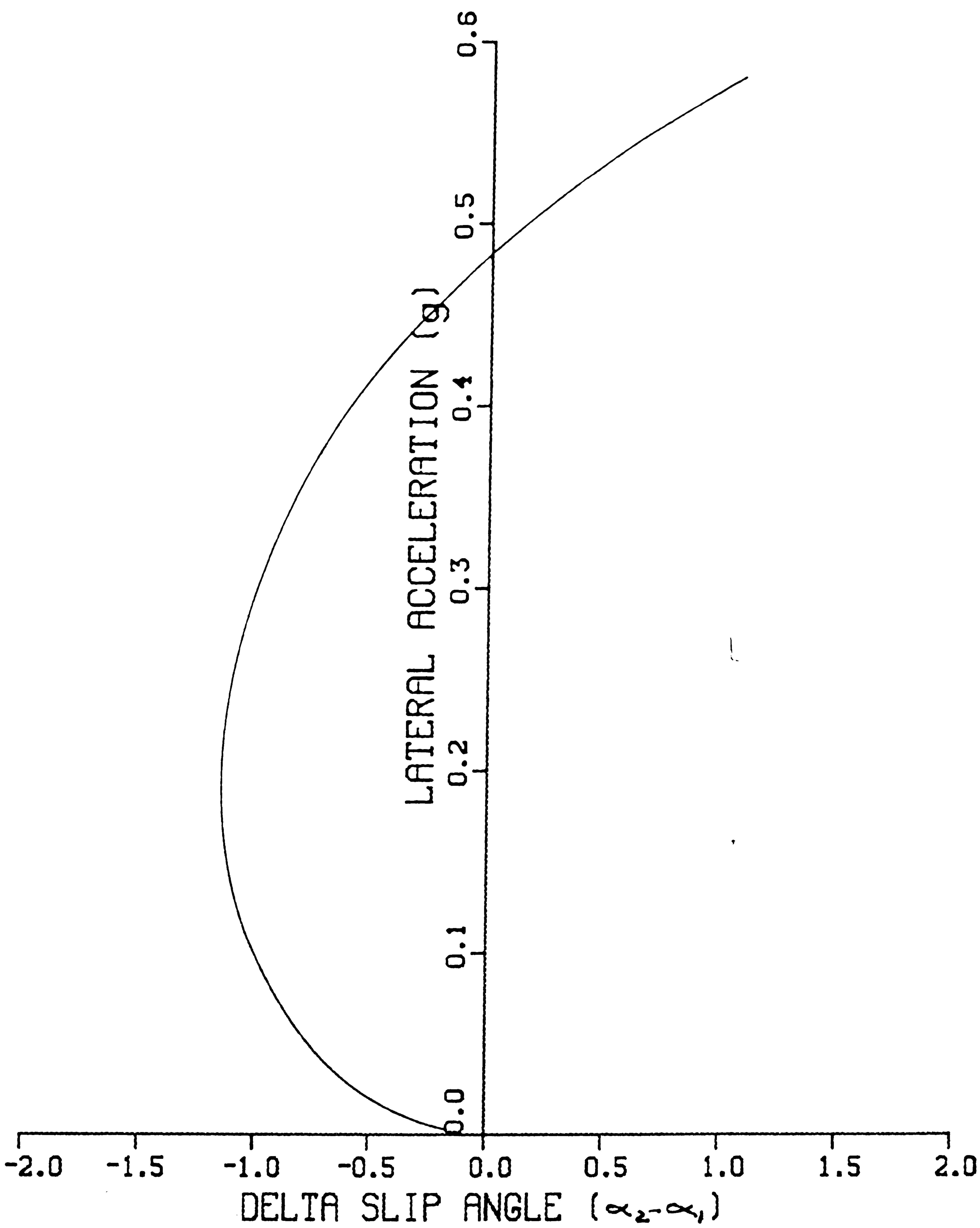


Figure 5

A HANDLING DIAGRAM
WITH CONTINUAL UNDERSTEER

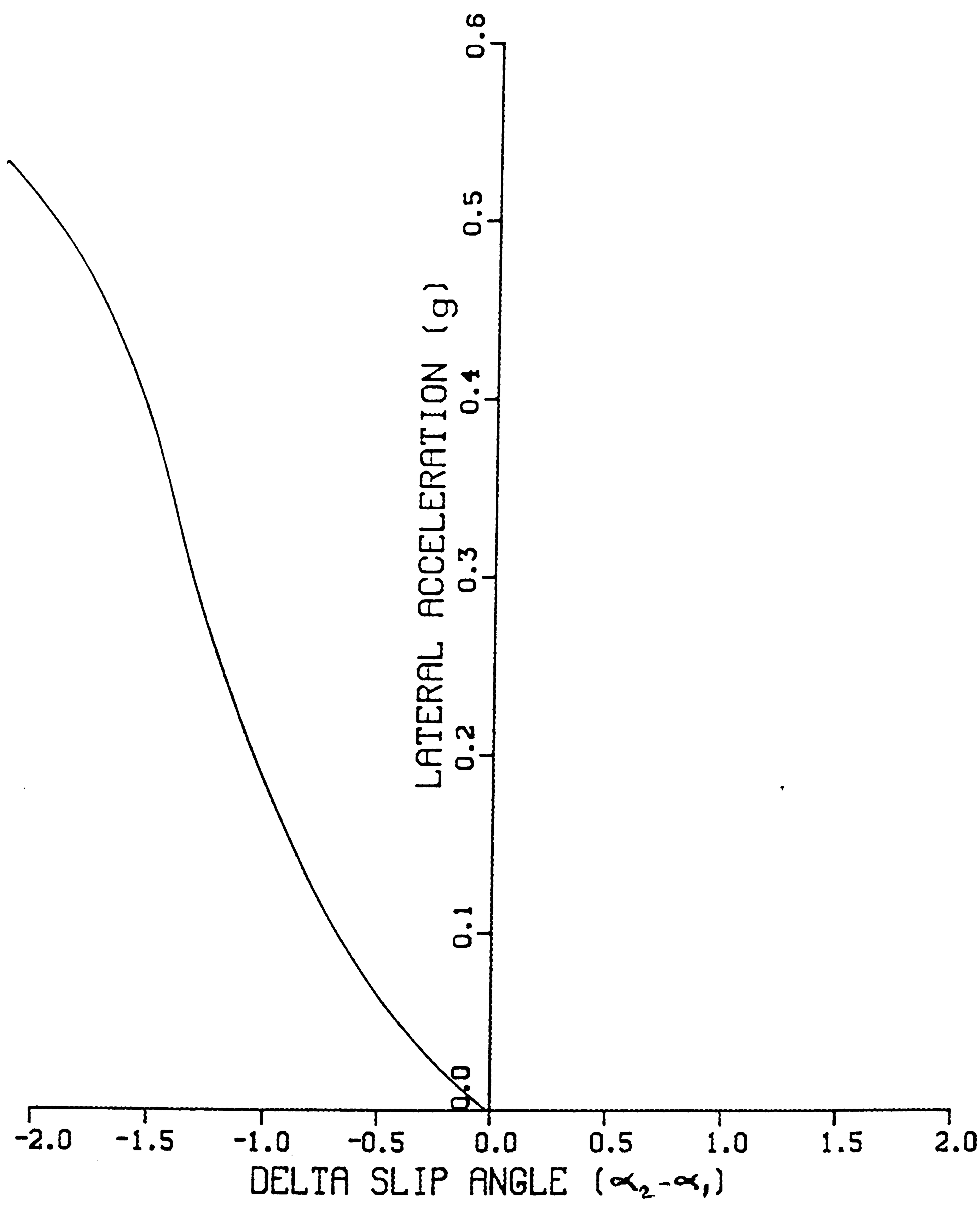


Figure 6

FORWARD VELOCITY EFFECTS
ON THE HANDLING DIAGRAM

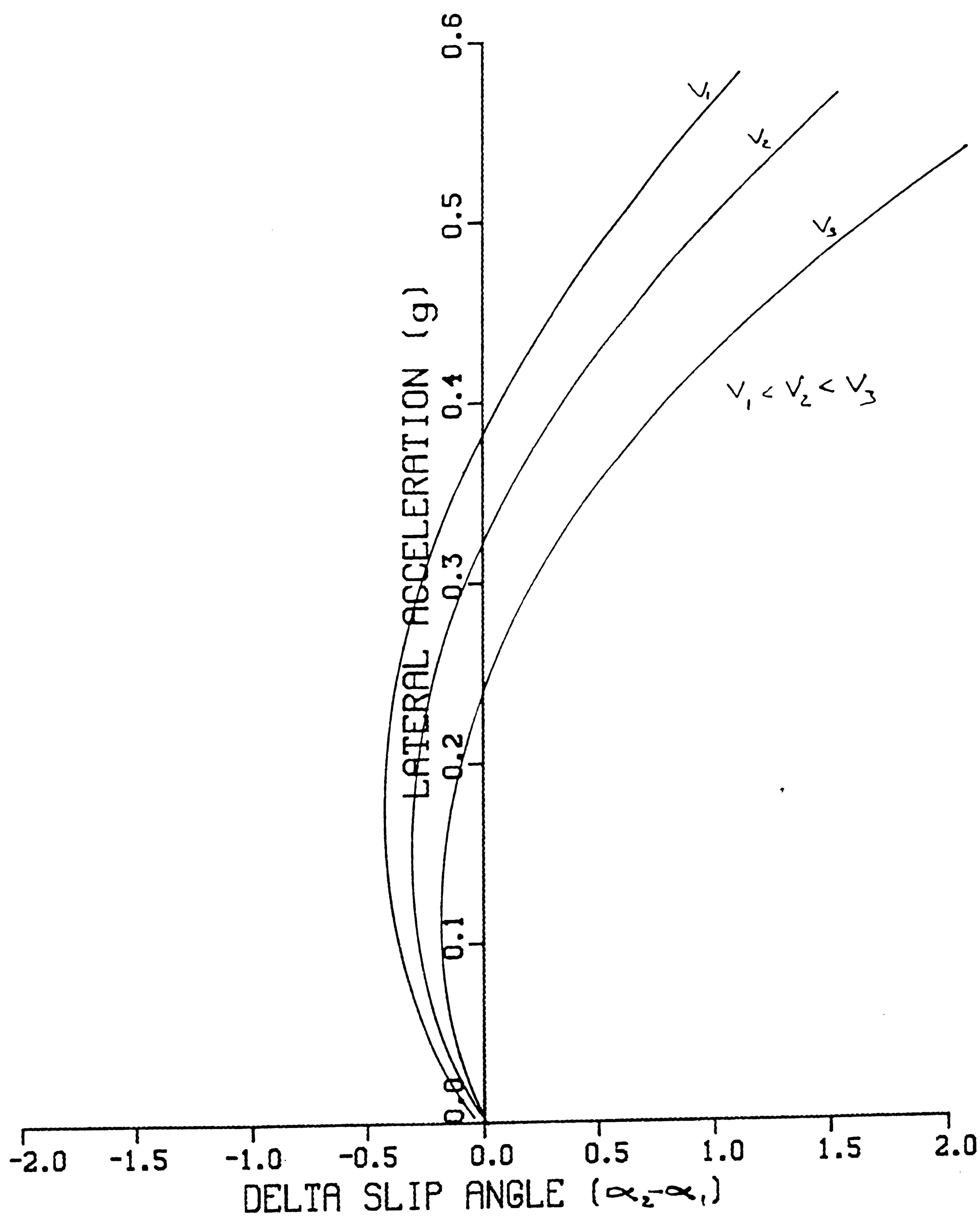


Figure 7

MODEL DEFINITION

The vehicle was modeled as a three axle tractor with a single 27 foot,single axle trailer. The code for an existing two axle tractor + multiple trailer model that had been developed at the University of Michigan for trailer roll-over stability studies was provided by MDI. This model was modified into a single trailer model by removing the second trailer and the single axle "dolly" that provided support for the the trailer's front end and connected it to the first trailer. The single trailer remained "connected" to the tractor by kinematic constraints that simulate the behavior of a fifthwheel. A second drive axle was added to the tractor. The position of both drive axles were changed such that the overall chassis wheelbase remained the same. A complete listing of the ADAMS model is included in Appendix B. (Note: For a 4x2 tractor the wheelbase is the distance between the front axle and the rear axle. For a 6x4 tractor the wheelbase is the average of the distances from the front axle to each of the two rear axles.)

Both the tractor and the trailer had an initial forward velocity of 1093.6 inches per second (62 MPH). The tractor's steering angle was the following function of time:

STEERING ANGLE = 30*(((TIME-0.5)/30)-SIN(2*PI*(TIME-0.5)/30)/2/PI))) ; TIME > 0.5

This function had the tractor steer straight ahead for the first 0.5 second, then the steering angle was increased following the above cycloidal cam rise type curve. The steering angle was implemented by pivoting the entire front axle about the tractor's yaw (z) axis, in the same manner as a child's toy wagon as shown in Figure 8. A more accurate model of a prototype steering system was not necessary, but could be added as part of a more detailed investigation.

The tires were modeled as a combination of three pairs (two translational and one rotational) of non-linear springs and non-linear dashpots. Again the simplified model was felt to be sufficient for this initial investigation. MDI offers a FORTRAN package called ADAMS/TIRE that could be used for further research. It models the tire forces as three non-linear translational forces and three non-linear moments. These forces are functions of many parameters, including deflection and its time derivatives.

In addition to values already mentioned the tractor and trailer had the following general characteristics:

Tractor

Weight = 10,000 lbf

Wheelbase = 120 inches

CG Height (above ground) = 44 inches

Front Axle Weight = 1200 lbf

Rear Axle(s) Weight = 2300 lbf

Front Tire Radius (loaded) = 21.2 inches

Rear Tire Radius (loaded) = 21.0 inches

Front Axle Spring Rate (vertical) = 2400 lbf/inch

Rear Axle(s) Spring Rate (vertical) = 1600 lbf/inch/axle

Front Tire Spring Rate (vertical) = 4500 lbf/inch

Rear Tire Spring Rate (vertical) = 9000 lbf/inch

Trailer

Weight = 30500 lbf

Length (kingpin to axle) = 270 inches

CG Height (above ground) = 81 inches

Axle Mass = 1500 lbf

Tire Radius (loaded) = 21.0 inches

Axle Spring Rate (vertical) = 20000 lbf/inch

Tire Spring Rate (vertical) = 9000 lbf/inch

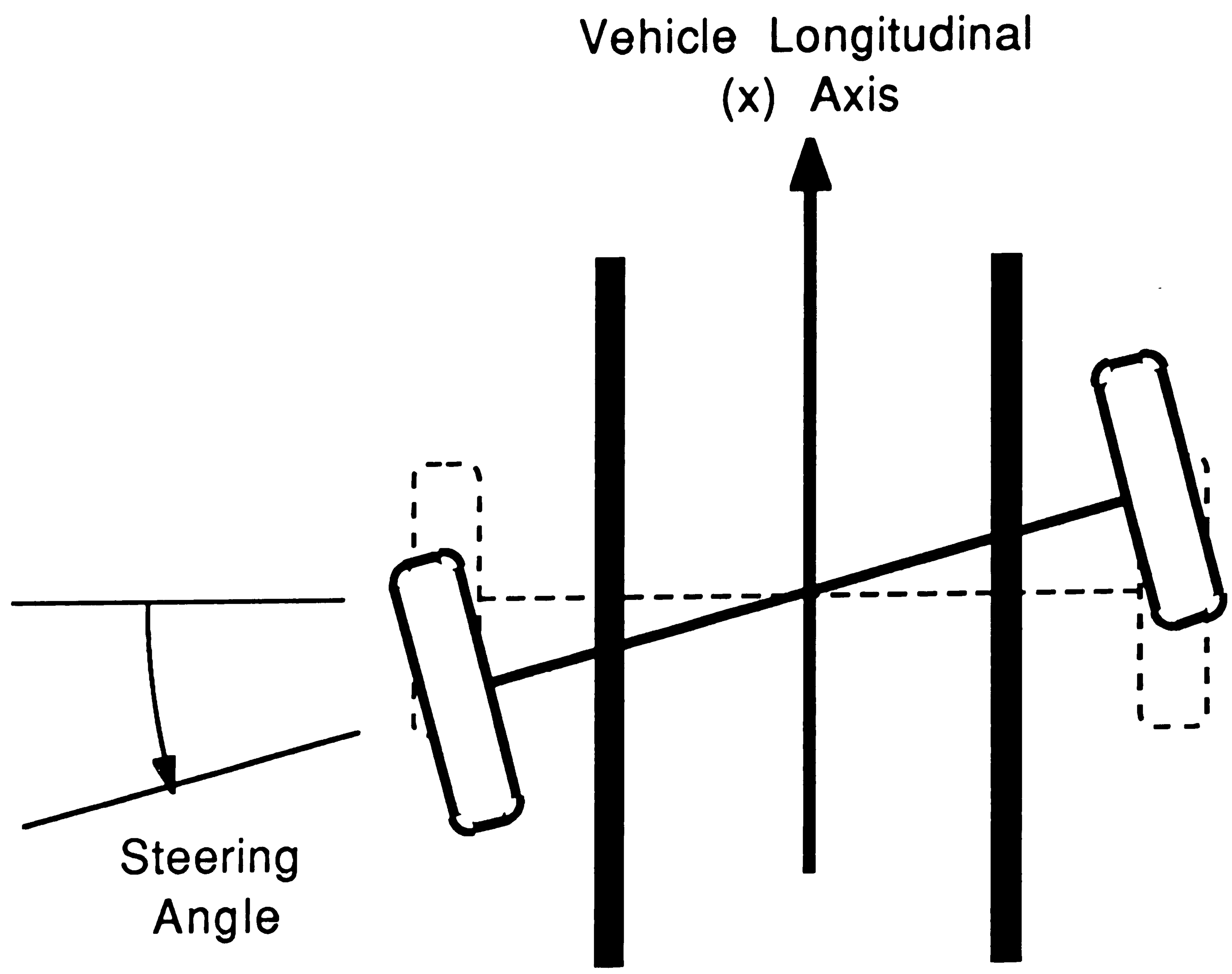
(Note: The model tractor rear tires and the trailer tires are double the spring rates of the front tires because prototype trucks typically use two tires per side at these locations.)

Two basic models were created, differing in the type of tractor rear suspension used. The first used a rear suspension whose links connected to the frame forward of the axle attachment point to provide the non-roll steer suspension, as shown in Figure 1. For the second model the links for the tractor's front rear axle attached to the frame rearward of that axle, creating a roll steer type of rear suspension, as shown in Figure 2.

Three links were used to locate and orient each of the tractor's rear axles as shown in Figure 9. The center link was constrained to remain in the tractor's xz plane as it pivoted. This kept the center of the axle in the tractor's xz plane. The two outer links were left free to shift in the y direction to prevent the suspension from binding as the axle twisted about the tractor's longitudinal axis. Translational springs were located between the axle end of each outer link and the frame to provide the tractor rear suspension. All three links were parallel when the tractor was at rest. The center link was displaced in the negative z direction from the other two, forming a three dimensional parallelogram mechanism, as shown in Figure 10.

For both types of tractor rear suspension models the difference in height between either end of each link, as measured at the static equilibrium position, was used as a model parameter. This dimension is denoted by the "h" in Figures 1 and 2. This variable ranged from zero and over ten inches, as shown in Figures 11 and 12. At zero all the links are horizontal when the vehicle is at rest. At h=10.57 inches the links are at an angle of 25 degrees.

MODEL STEERING



Top View

Figure 8

ADAMS MODEL SUSPENSION
(ONE DRIVE AXLE SHOWN FOR CLARITY)

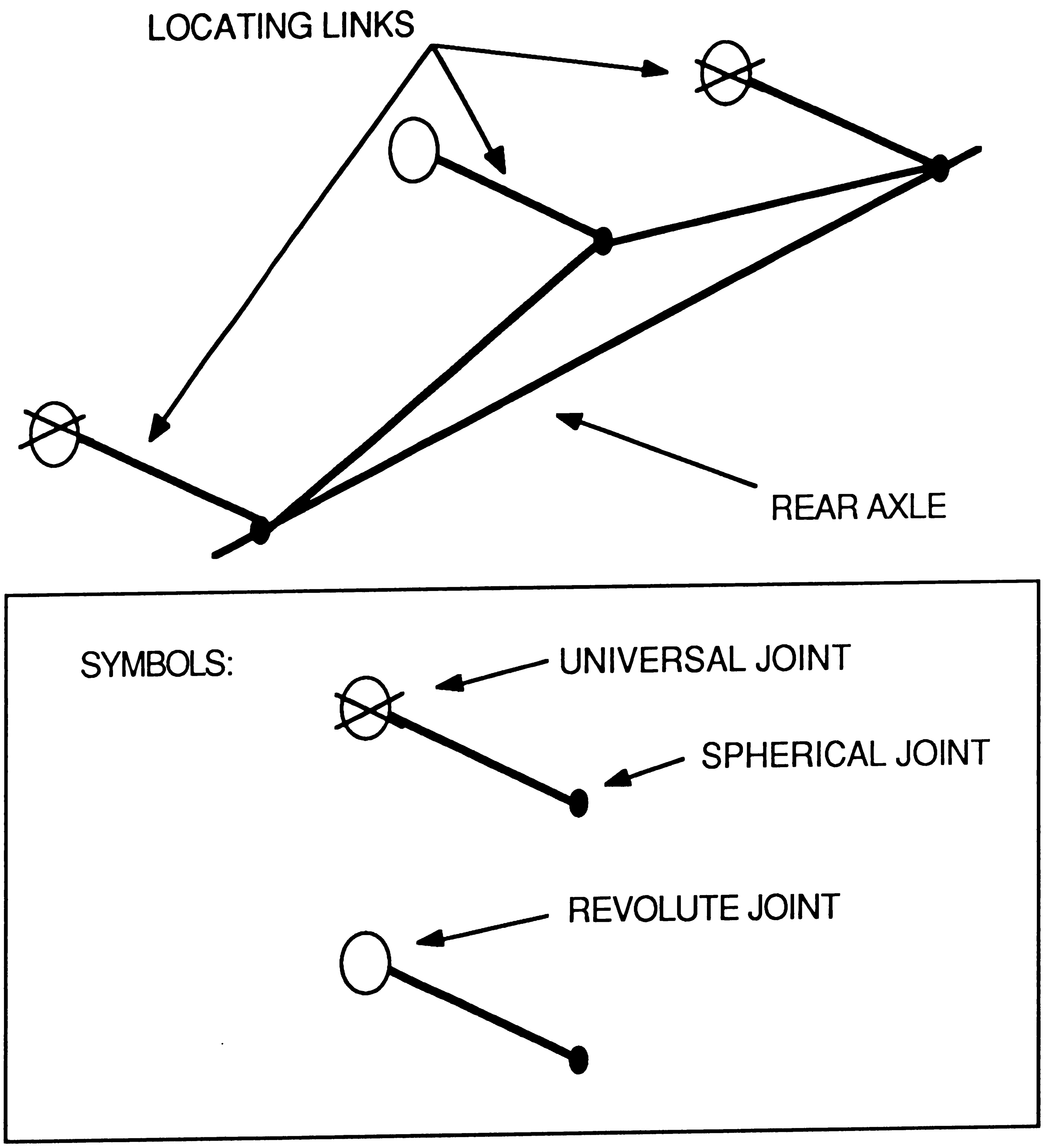
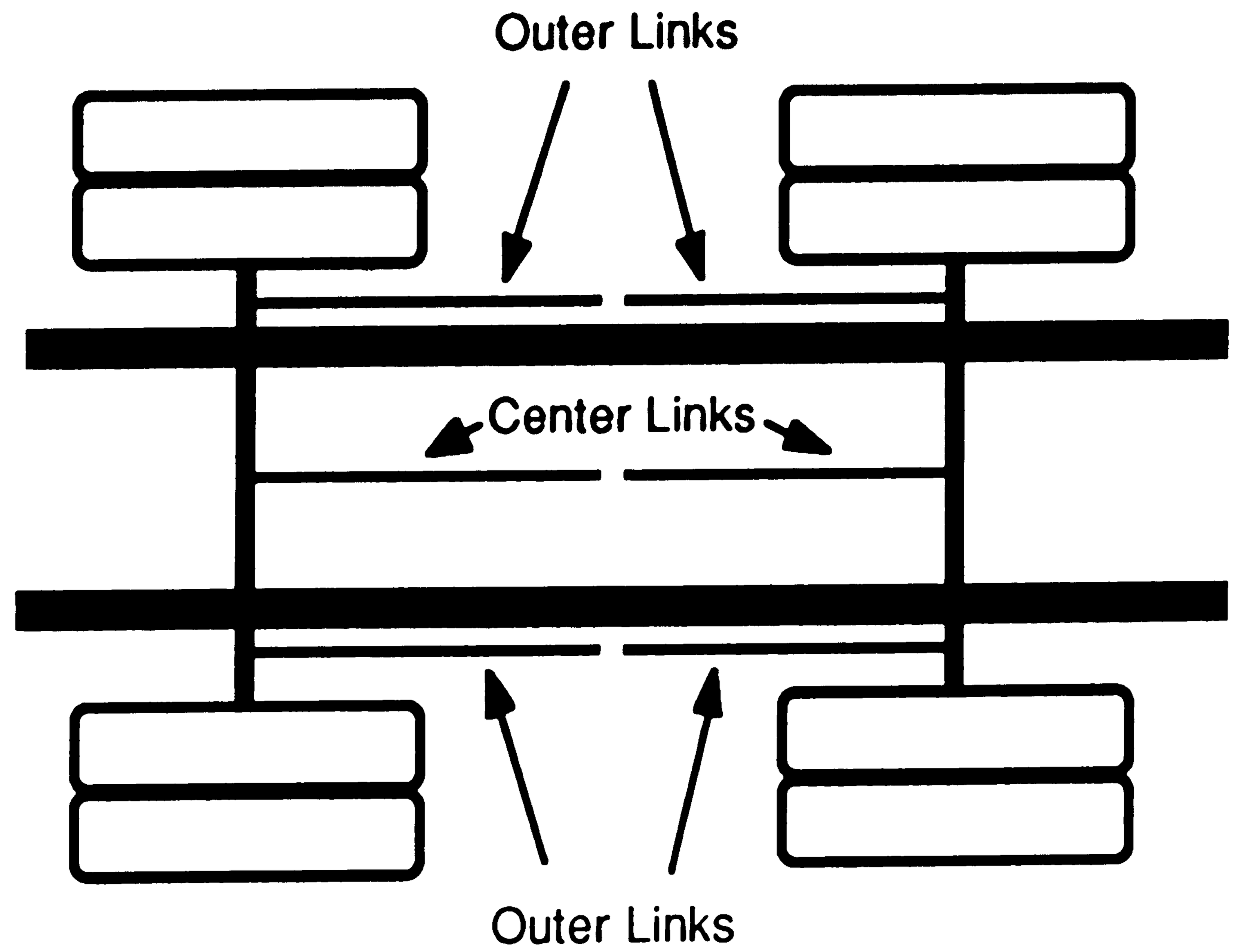
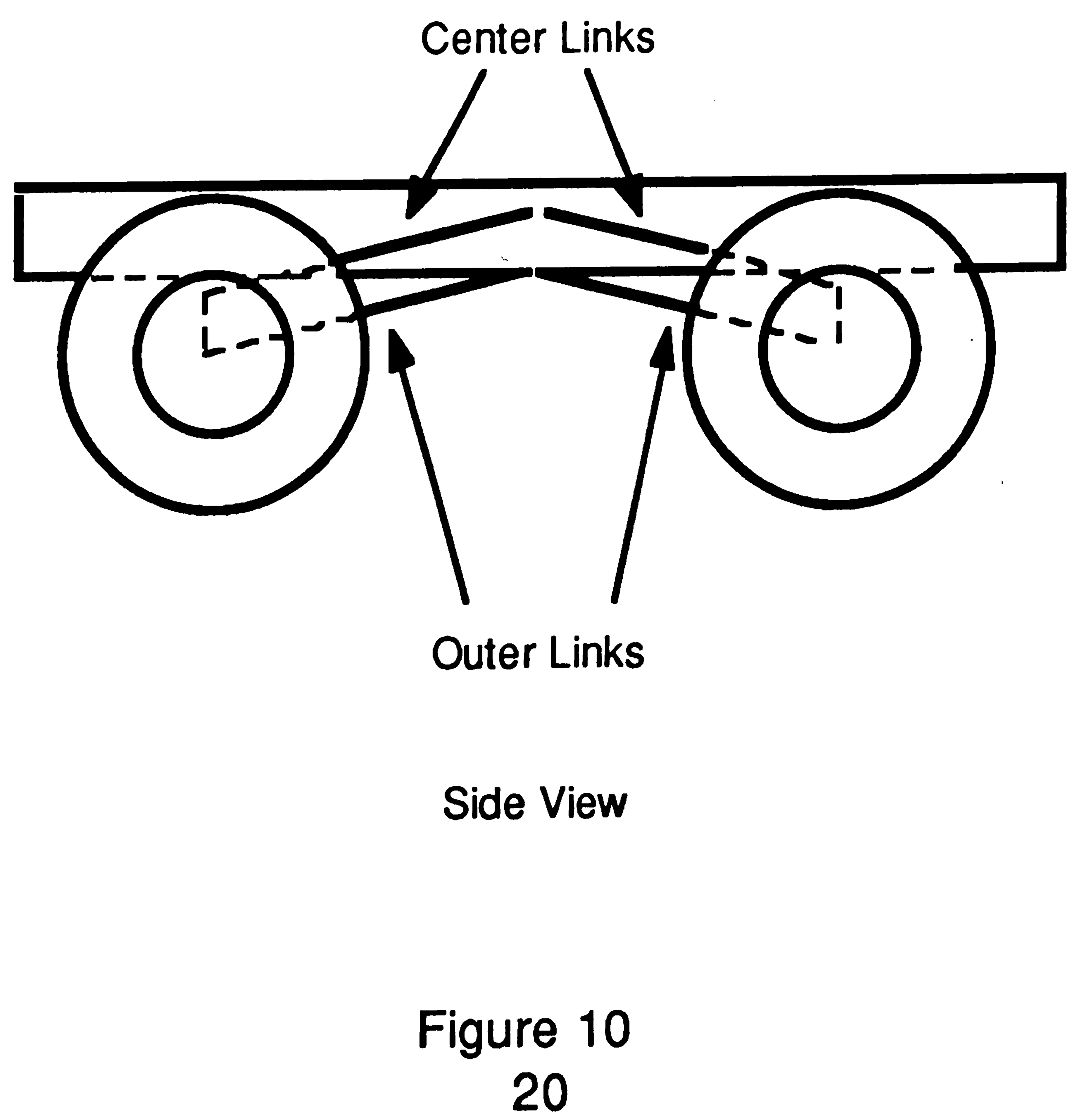


Figure 9

ROLL STEER SUSPENSION



Top View



Side View

Figure 10

EXAMPLES OF "h" VARIATION FOR
A NON-ROLL STEER SUSPENSION

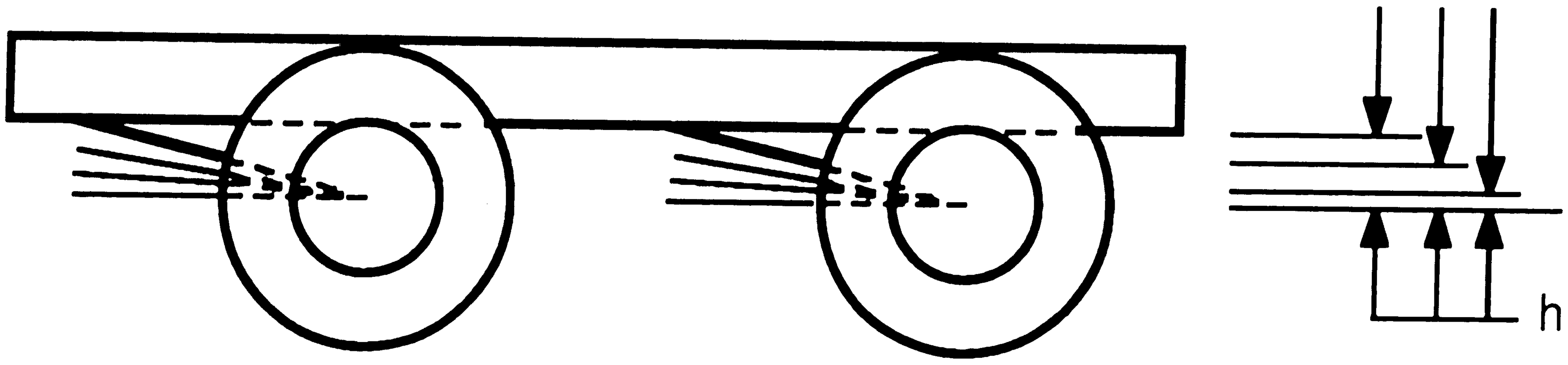


Figure 11

EXAMPLES OF "h" VARIATION FOR
A ROLL STEER SUSPENSION

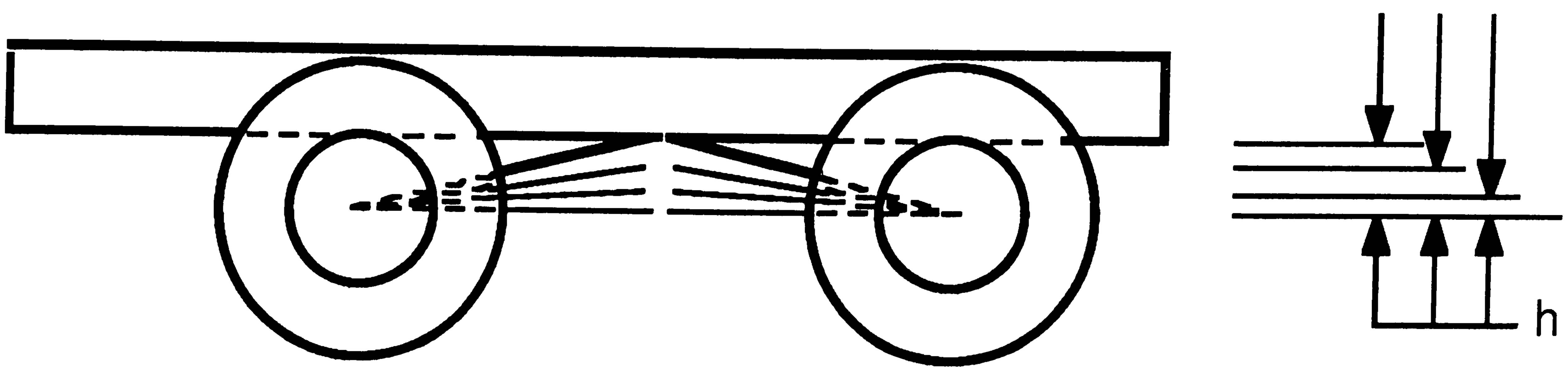


FIGURE 12

PROCEDURE

The initial step in the chassis analysis was the input of the vehicle model, as it was received from MDI. Once the model was loaded onto the Lehigh University Computer Center (LUCC) VAX 8530, work was started on the FORTRAN subroutine to request the lateral acceleration and slip angle data from ADAMS. ADAMS allows multiple user-written FORTRAN subroutines to be linked to the basic executable code. Such subroutines can be used to define complex force relationships, such as those inside an automobile shock absorber, and complex information relationships, like those used for this work.

In order to obtain the vehicle's lateral acceleration for the handling diagram a phantom part was added to the model. The phantom part was fixed relative to the tractor and constrained to act as the projection of the tractor's center of gravity onto the ground plane. The details of the definition of this part are described in Appendix B. The lateral acceleration was the phantom part's LPRF y axis component of its total acceleration as shown in Figure 13.

Similar phantom parts were set up to act as the projections of the front axle center and bogie center onto the ground plane. These phantom parts were used to calculate the tire slip angles for the axles. The ratio of the LPRF y axis component to the LPRF x axis component of a particular phantom part's instantaneous velocity was the tangent of the slip angle.

The slip angle calculation took the steering angle into account, as shown in Figure 14. Since the model was steered by rotating the entire front axle, like a toy wagon, and the rotation of the front axle LPRF is the definition of that part's rotation, the front axle steering angle was already included in the calculation of the slip angle.

The bogie center was used for the 6x4 tractor slip angle calculation instead of the rear axle center, as in a four wheeled vehicle (or 4x2 tractor). For a rear suspension with roll steer characteristics it would not be appropriate to use the center of either of the rear axles as the reference point to calculate the rear slip angle, as the slip angle for the other rear axle would differ. Instead an average value must be used. The bogie center lay on the plane of symmetry (perpendicular to the vehicle's longitudinal axis) for the bogie motion of the roll steer rear suspension, and thus provided the average slip angle. For a bogie without roll steer characteristics either the bogie center, or either of the rear axle centers could have been used, as all would have approximately equal values (neglecting the effects of the different distances to the front axle).

These velocities and the acceleration were requested from the analysis by a FORTRAN subroutine named REQSUB (Appendix C). The velocities were converted into slip angles in this subroutine. The bogie slip angle was then subtracted from the the front axle slip angle. The lateral acceleration, difference between the slip angles, and the individual slip angles were output to a file as part of an information array.

To test REQSUB a simulation of a model consisting of only the 4x2 tractor was performed. This model was run at a constant speed, at a constant steering angle. This resulted in reasonably steady state values for lateral acceleration and slip angles. There were some fluctuations in the model results. These were attributed to the low damping in the tractor and the high jerk imparted by the steering angle motion. The steering angle had been set to go from 0 degrees to 2 degrees, following a sinusoidal cam profile, over a 0.3 second time interval. This results in theoretically infinite jerk at the beginning and end of the transitions,' producing the undesired oscillations. The details of the model verification are included as Appendix D.

Since the goal was to produce simulations with varying levels of lateral acceleration and observe the resulting difference between the slip angles, a constant model velocity along a decreasing radius path was used. Thus the V**2/R value for lateral acceleration would be constantly increasing, until the analysis failed due to the modeled vehicle rolling over. Similar results could have been achieved by either having the model traverse the same path with increasing velocity, or by having the model maintain constant velocity and traverse different paths to produce a family of curves.

These alternatives were not chosen due to the construction of the ADAMS model. The increasing velocity alternative was rejected because in the model there are no forces acting to either accelerate or retard the vehicle along the longitudinal axis. The vehicle was given an initial velocity along its positive X axis, and maintained that velocity throughout the analysis. In order to have the velocity increase, a force would have had to be added to the model, adding to its complexity and changing it further from the established model provided by MDI.

The fixed path alternative was also rejected due to the model's construction and the workings of ADAMS. The path of the vehicle during the simulation is determined only by the forces developed by the model's tires. In order to have the model follow a predetermined path it would have been necessary to construct a feedback control system between the vehicle's position, velocity and acceleration, and its steering motion. While this would have been possible through a FORTRAN subroutine like the one used to produce the information needed for the handling diagrams, it would have added unnecessary complications to the model. Also many additional simulations would have been required to produce the information required to create complete handling curves.

To produce the required decreasing radius path a steering input that increased linearly with time was initially chosen. While this steering input was very simple, it created problems in the model. The extreme levels of jerk this imposed on the model created oscillations in the lateral acceleration and slip angles. The steering input was not originally recognized as the cause, and so variations in the model initial velocity and damping values were tried to reduce the oscillations. These had minimal affect on the model behavior, and were left in the model during subsequent analyses.

Once the extreme (theoretically infinite) jerk was recognized the steering input was modeled as a cam profile displacement. Two solutions were explored: a cycloidal cam profile, and a "4-5-6-7 polynomial" cam profile.[10] A comparison of results from analyses using each of these profiles showed the cycloidal steering input to produce fewer and smaller oscillations at lower values of lateral acceleration. This can be seen in Figure 15.

Each analysis consisted of two simulations of the same model. The first simulation solved for the model's static equilibrium position. This allowed the vehicle to "settle" on its suspension. After the vehicle reached static equilibrium a dynamic simulation was started. This took the model from its rest state and applied the initial velocity conditions. By performing these simulations in this manner any large vertical motions or changes in vertical forces were avoided. The steering displacement was delayed by 0.5 second (simulation time) to allow any smaller fluctuations to dampen.

After the modeling and analysis problems had been resolved the final model simulations were run. ADAMS command procedures were created to initiate the analyses in batch mode on the LUCC VAX 8530. An example of these command procedures is shown and detailed in Appendix E. The analyses were run in batch mode because each individual analysis required at least thirty CPU minutes to execute.

Five sets of analyses were performed, each set consisting of a run of the model with rear suspension roll steer characteristics and one of the model with the non-roll steer rear suspension. The value for "h" was changed between each set such that the angle that the rear axle locating links were inclined from the ground plane ranged from 0 to 25 degrees, in increments of 5 degrees, as measured with the vehicle at its static equilibrium position.

VEHICLE LATERAL ACCELERATION

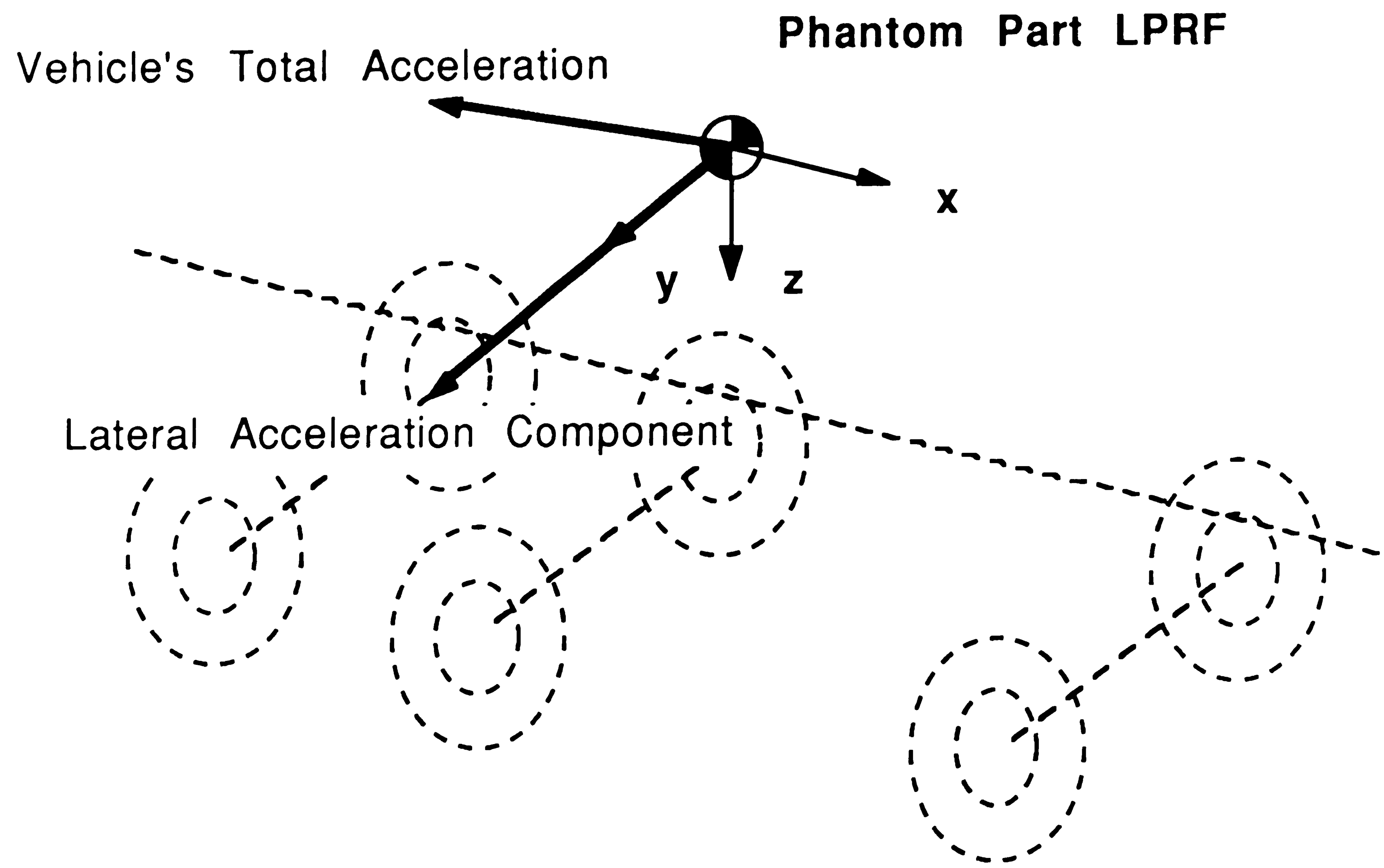
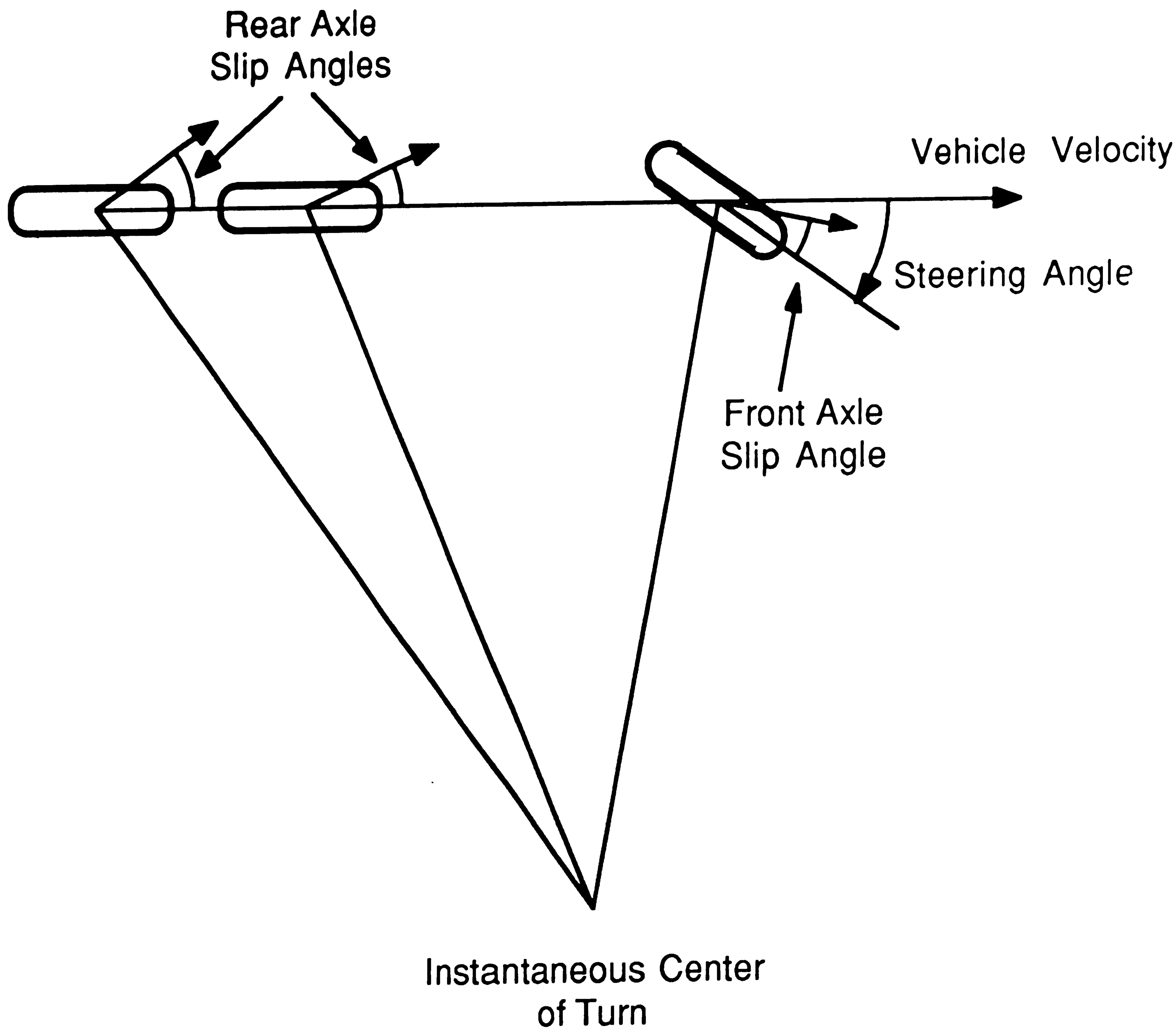


Figure 13

SLIP ANGLE DEFINITIONS



Each full axle is represented by a single tire.
Slip angles for tires on the same axle are
approximately equal.

Figure 14

CYCLOIDAL VS.
4-5-6-7 POLYNOMIAL
STEERING INPUT

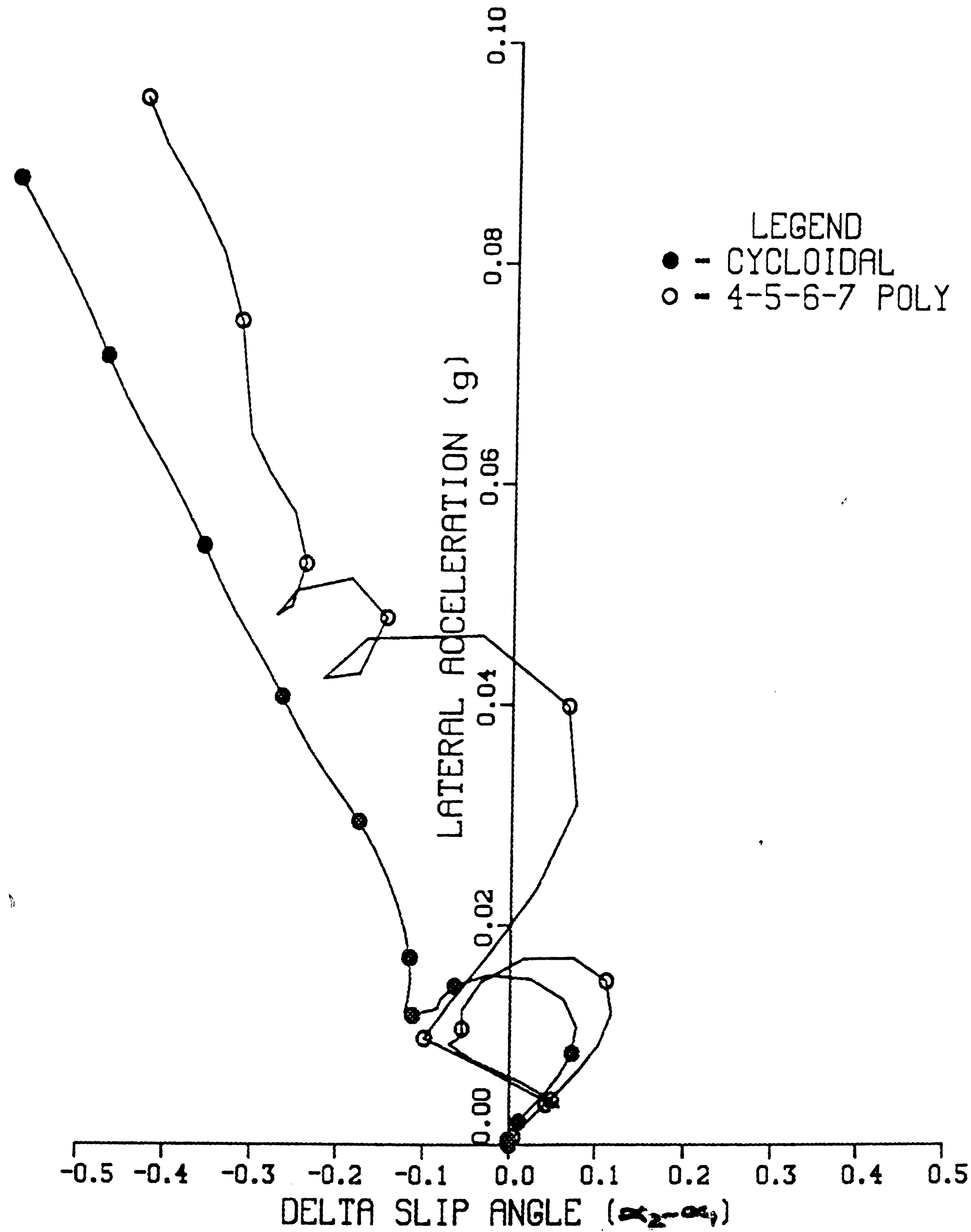


Figure 15

RESULTS

The results of the analyses were plotted as handling diagrams on Figures 16 and 17. Figure 16 shows the roll steer suspension and Figure 17 is the non-roll steer suspension. Each diagram contains the traces for all five pivot height values ("h"). Figure 18 shows the extreme values of "h" plotted for both cases plotted on the same graph. All the curves were plotted directly from the ADAMS results, with no curve fitting attempted. The only data reduction that was performed was the blanking of results at the upper and lower end. The results shown in these figures agree both in magnitude and overall shape with the graphs in Reference 1.

The lower end results were edited to avoid confusion between the various curves near the graph's origin. All the traces passed through the origin and undergo fluctuations in both lateral acceleration and (delta) slip angles in its neighborhood. The fluctuations at low levels of lateral acceleration caused difficulty in identifying the individual curves. The reason for them was not determined.

The upper end results also had large, erratic fluctuations. These were due to the vehicle lifting its inside tires at high levels of lateral acceleration. In several analyses the model actually rolled over, causing the traces to have extreme perturbations. The ADAMS post-processor allowed both the slip angles and lateral acceleration to be plotted with respect to simulation time. This was used to edit the results to before a time when the tires lifted from the pavement.

The item of interest in all the figures was the point at which the slope of a trace goes from negative to positive, as lateral acceleration increases. This threshold value for lateral acceleration indicates where the tractor goes from an understeer condition to one of oversteer.

For the suspension with roll steer characteristics the levels of lateral acceleration for the threshold point ranged from greater than 0.5 g to greater than 0.3 g, for h=0 and h=10.57 respectively. The exact value of the lateral acceleration was difficult to fix due to the extremely gradual change in curve slopes.

For the suspension without roll steer characteristics, the levels of lateral acceleration for the threshold point ranged from greater than 0.4 g to less than 0.25 g.

In Figure 18 it was observed that the curves for h=0 are very similar, though the non-roll steer suspension model has a strange saw toothed trace for the entire range of lateral acceleration. The reason for this shape was not determined, but it was repeated during all runs of that particular model. It is quite clear from the traces for h=10.57 that the threshold point was higher for the suspension with roll steer characteristics.

NON-ROLL STEER
SUSPENSION

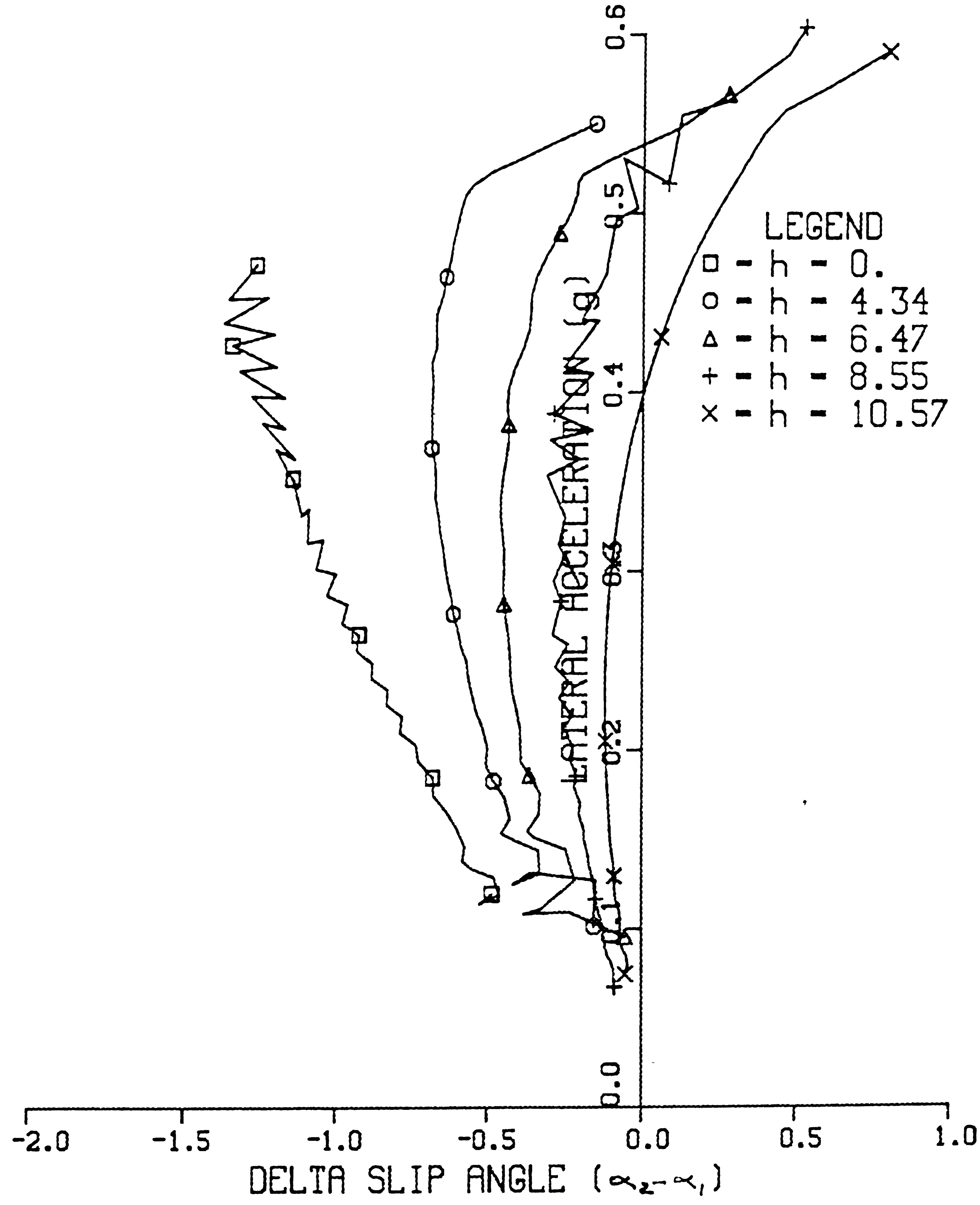


Figure 16

ROLL STEER
SUSPENSION

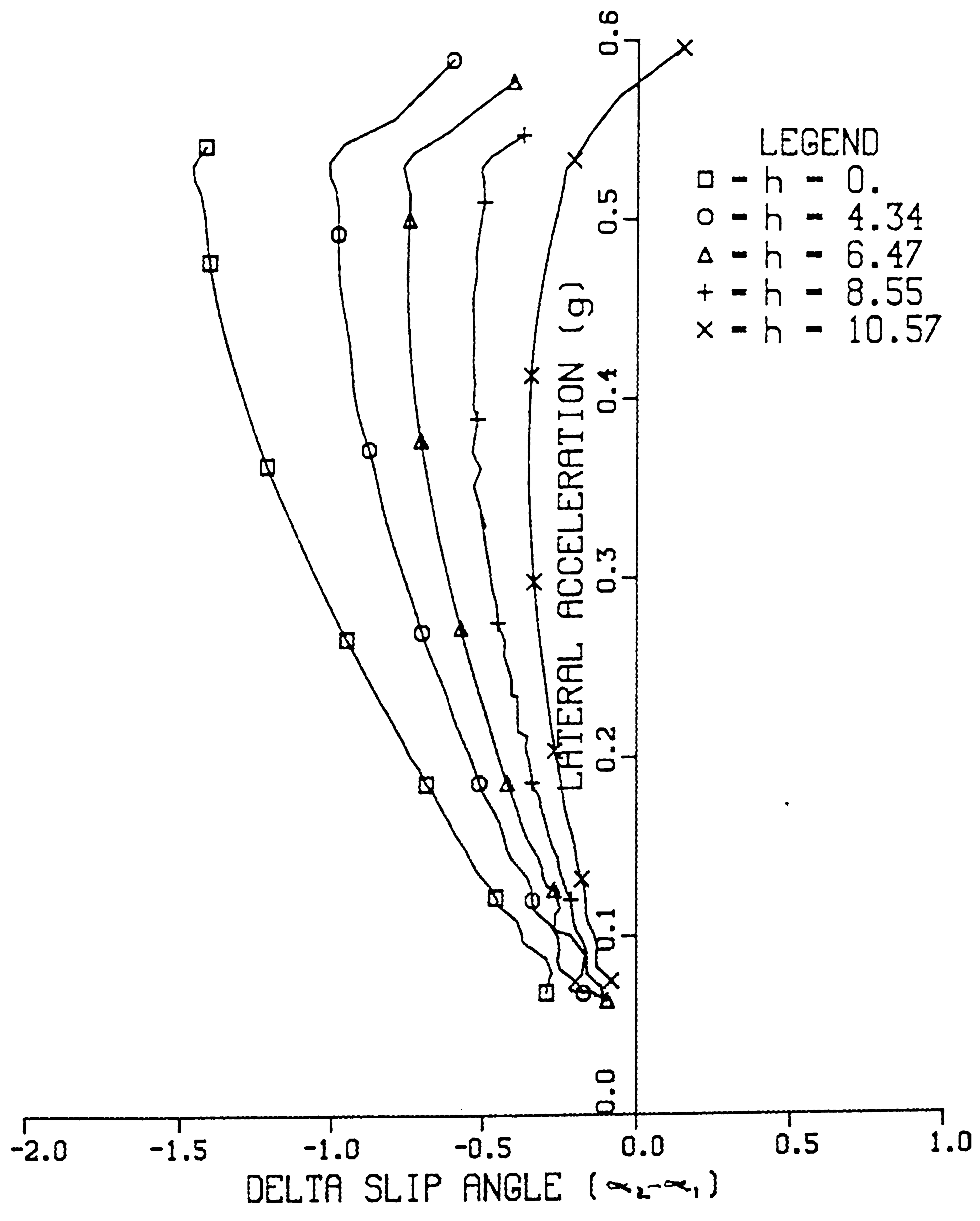


Figure 17

ROLL STEER VS.
NON-ROLL STEER
SUSPENSION

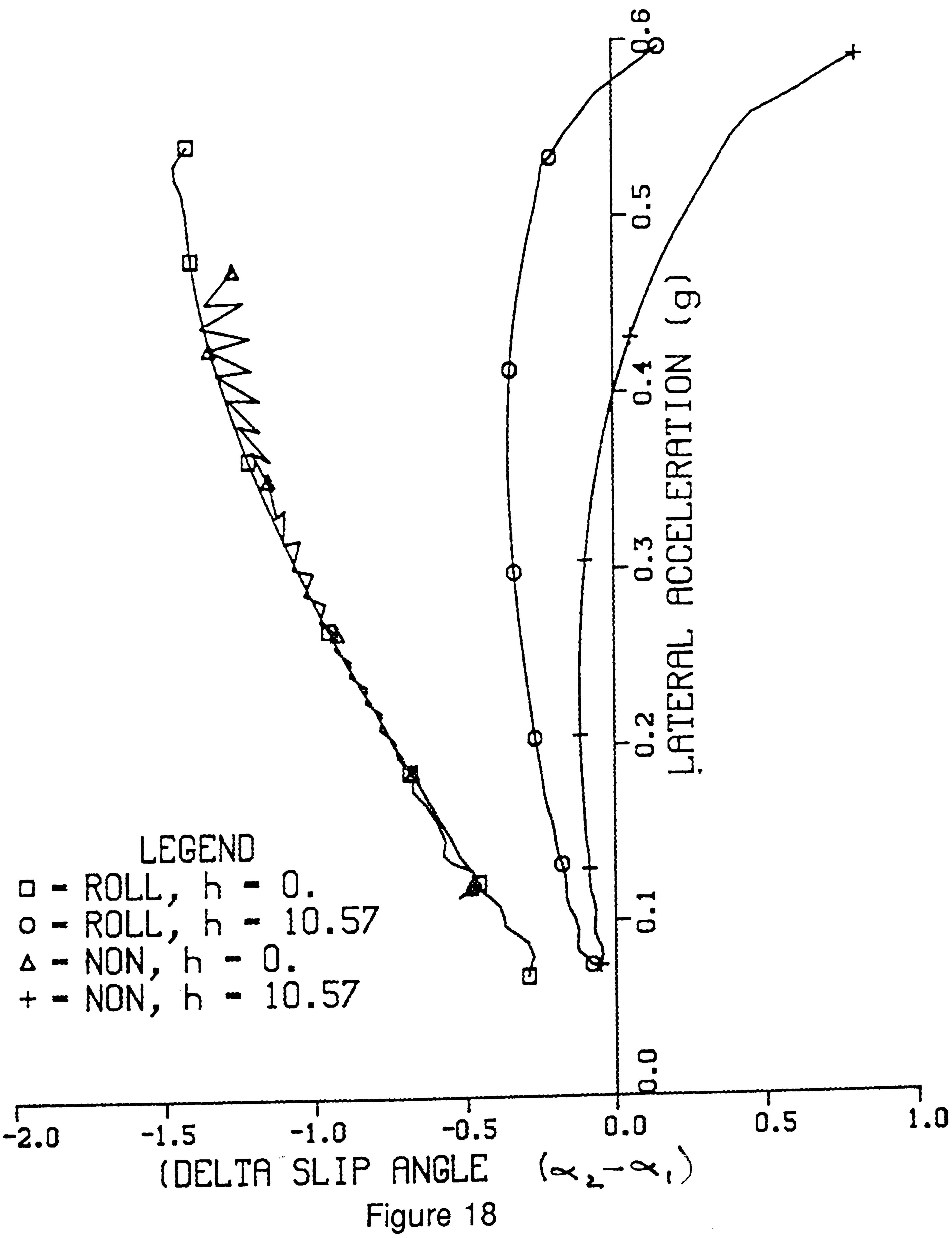


Figure 18

CONCLUSIONS

The results exhibited in Figures 16, 17, and 18 lead to the conclusion that the oversteer threshold points occur at higher levels of lateral acceleration for the model with the rear suspension with the roll steer characteristics. This implies that the model has better cornering ability with the roll steer suspension.

The level of lateral acceleration at which the oversteer threshold point was reached decreased as "h"was increased. This can be attributed to the increasing amount of roll steer that occurs as the link angle is increased. The roll steer linkage functions as the difference between the projection of the inside and outside links onto the ground plane. When the links are nominally horizontal the projections are not greatly affected by the chassis roll. As the angle is increased any variations of that angle, such as those created by chassis roll, have a more pronounced effect on the projected lengths. This is the same process by which small variations of an angle which is nominally zero produce negligible variations in the cosine of that angle. The variations in cosine increase as that angle is increased.

It would not be appropriate to try to extrapolate the cornering ability for an actual tractor + trailer combination vehicle from this model. This model was very simplistic in its steering and tire force mechanisms. It is possible to conclude that for this model, stability decreased as the height of the tractor end of the suspension linkages was increased. The trends were quite pronounced, and suggest that the analysis technique employed for this study could be used to reduce the amount of full scale testing that would be required during a truck suspension development project. Variations in the "h" value, as dictated by vehicle hardware layout, could be evaluated for their relative effects on stability prior to building a prototype vehicle.

Further studies should pursue these observations. Additional computer simulations should use a more complete front suspension and steering linkage as well as the more refined ADAMS TIRE model for tire forces. Further variations could include the effects of linkage end bushing compliance, load shifting (especially for bulk liquid tank trailers), and vehicle roll-over stability.

An explanation for the reduced cornering stability with a non-roll steer rear suspension when compared to one with roll steer characteristics can be seen by examining the Ackerman steering geometry of a prototype 6x4 tractor, both with and without roll steer characteristics. The Ackerman steering geometry for a four wheeled vehicle is shown in Figure 19. The principle is that for a zero speed turn the intersection of lines perpendicular to the projections of the wheel planes onto the ground plane should intersect at a common point. This point is the virtual center for all the tire paths, and thus the all tires are in pure rolling motion. This means that all the tire slip angles are zero.

In practice this only occurs at one designed value of the steering angle, typically about twenty degrees. This is due to the inability of a simple four-bar steering linkage of the type used for heavy duty truck steering systems to produce perfect Ackerman geometry at steering angles other than the design angle for a specified vehicle wheelbase. This dependence of Ackerman steering geometry on the vehicle wheelbase is in contradiction with the vehicle design requirements that need large variations of the wheelbase lengths and the economic pressures to keep part variations to a minimum.[11]

For a 6x4 vehicle with a non-roll steer rear suspension the intersection of the virtual centers is not possible. Ackerman steering geometry can only be set up by considering the centerline of the bogie, as shown in Figure 20. This induces the rear tires into some lateral compliance since the tires attempt to roll in a straight path and are being forced into following the vehicles curved path. This causes some distortion of the tire's footprint at the ground plane, reducing the percentage of the the total lateral tire forces available to produce lateral acceleration and lowering the vehicle's oversteer threshold.

For a 6x4 vehicle with a rear suspension that has roll steer characteristics of the type shown in Figure 21, it is still possible for all the virtual centers to intersect, and thus have a pure rolling condition. Then the entire lateral force capability of the rear tires is available to produce lateral acceleration. This results in a higher theoretical cornering stability. Even when rear axle alignments do not change drastically enough for all the virtual centers to intersect, the bogie virtual center no longer is at an infinite distance from the vehicle. The bogie tires are then trying to follow a curved path which is closer to the actual vehicle path than the straight path of the non-roll steer suspension. The tire footprints are therefore not as distorted and the tire is then capable of producing higher level of resistance to slipping.

Ackermann Geometry

For a 4 x 2 Vehicle

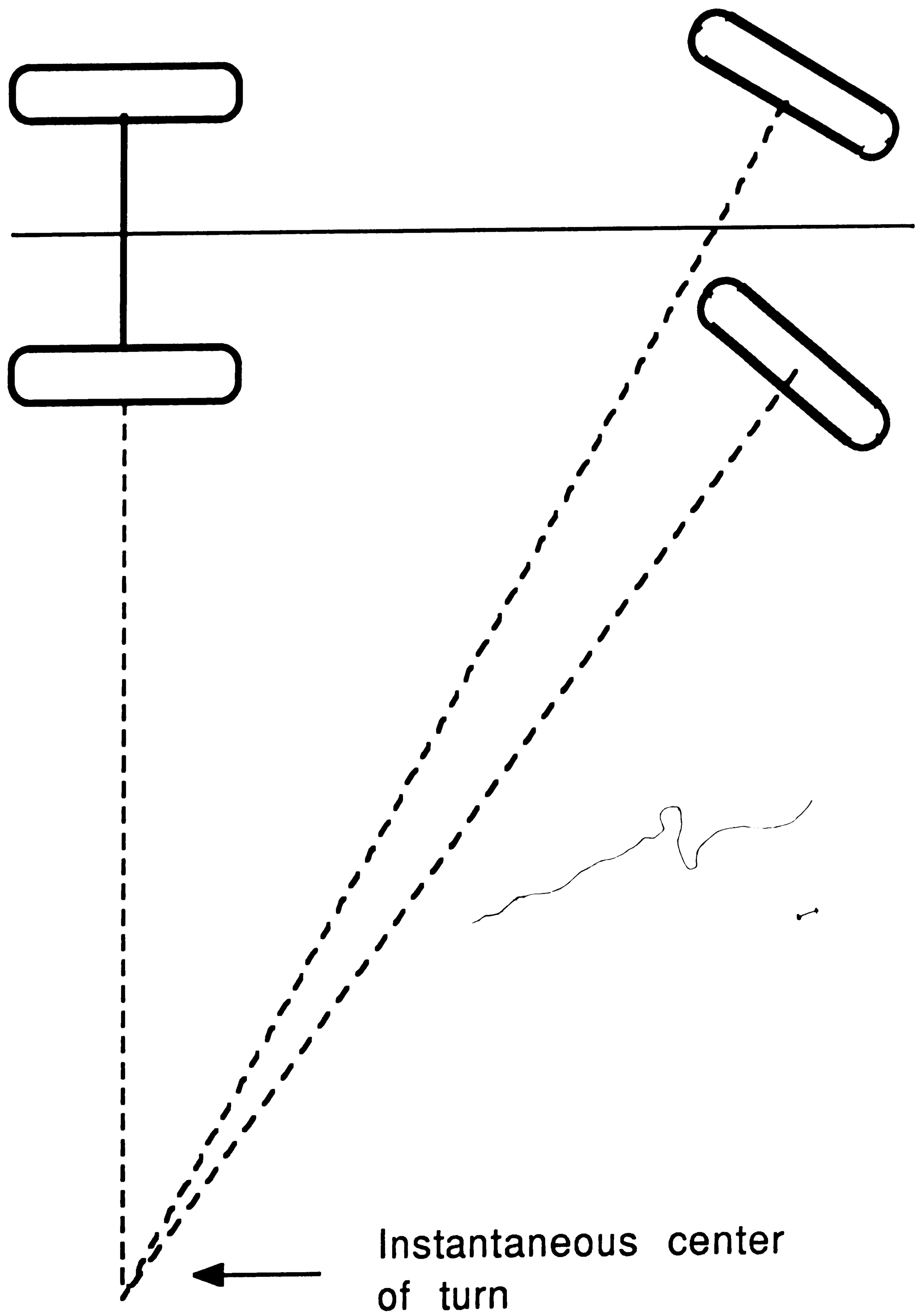


Figure 19

Ackermann Geometry
For A 6 X 4 Vehicle
Without Roll Steer

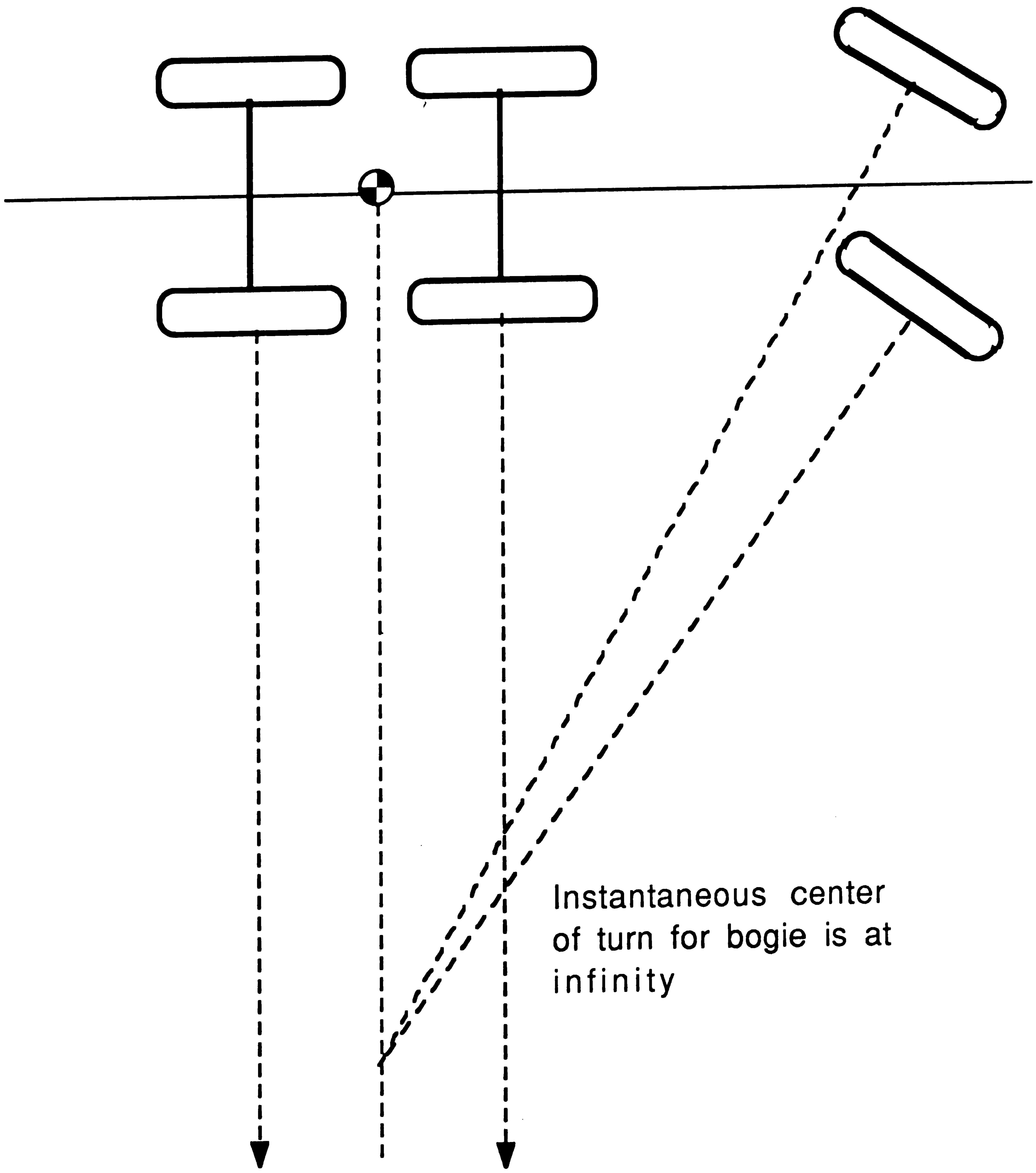


Figure 20

Ackermann Geometry
For A 6 X 4 Vehicle
With Roll Steer

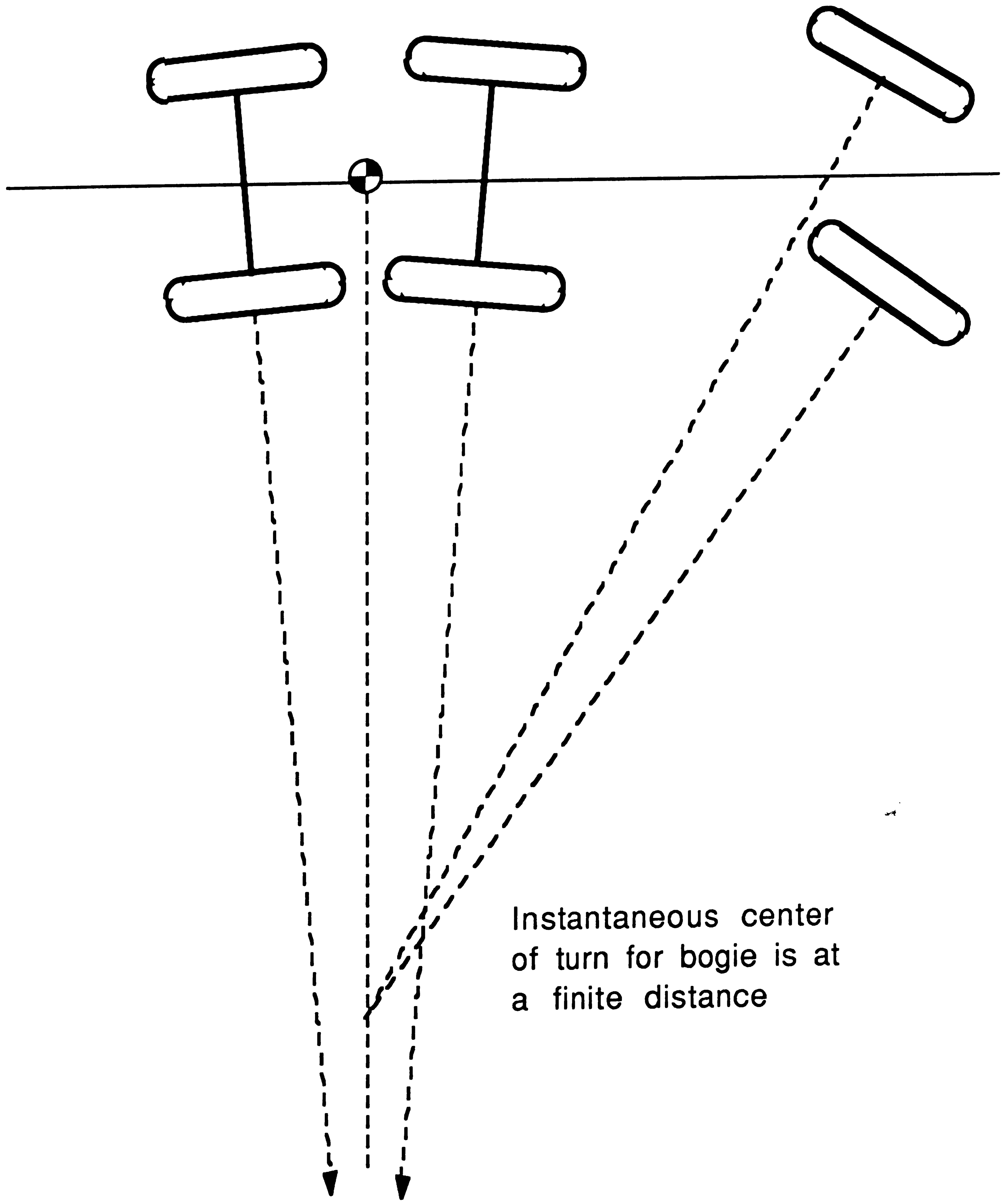


Figure 21.

APPENDIX A - GLOSSARY

The following terminology was used in the descriptions of the various physical aspects of heavy-duty trucks.

Tractor- A vehicle designed to pull a trailer or semi-trailer.

Truck - A vehicle designed to carry a payload without a trailer, such as a delivery truck. Also used as a general term meaning a truck, tractor or tractor+trailer combination.

4x2 - A tractor with one powered axle and one non-powered axle.

6x4 - A tractor with two powered axles and one non-powered axle.

Semi-trailer - A trailer with an axle at one end only. For the purposes of this work the terms trailer and semi-trailer are used interchangeably, both implying a true semi-trailer.

Bogie - The two drive axles of a 6x4 tractor . It could also mean the two rear axles of a trailer.

Bogie Wheelbase - The longitudinal distance between the two bogie axles (as measured between the axle centerlines).

Fifth Wheel - The connection between a tractor and semi-trailer. Kinematically, it is a constraint on the rotation of the semi-trailer with respect to the tractor, allowing only pitch and yaw.

Jack-knife - When the tractor bogie skids causing the tractor to swing away from the intended direction of travel, while the trailer tries to follow the intended path. The name comes from the shape of a partially opened pocketknife.

Trailer Swing - When the trailer axle(s) skid, causing the trailer to swing away from the intended direction of travel.

The industry standard Society of Automotive Engineers vehicle dynamics terminology was used. All definitions are taken from the Society's "Vehicle Dynamics Terminology" publication.[12] Important definitions and sign conventions are given here:

Earth-Fixed Axis System (X,Y,Z) - A right hand orthogonal axis system fixed to the ground. The X-Y plane is the road (ground plane). The Z axis is directed downward. This is also referred to as the Global Coordinate System in ADAMS nomenclature. See Figure 22.

Vehicle Axis System (x,y,z) - A right hand orthogonal axis system that is fixed to the vehicle. Its origin is the vehicle's center of gravity. The x axis (longitudinal axis) is directed forward. The y axis (lateral axis) is directed towards the passenger's side of the vehicle (for a left hand drive vehicle). The z axis is directed downward. All rotations also follow normal right hand rule convention. The roll axis is the vehicle's x axis. The pitch axis is the vehicle's y axis. The yaw axis is the vehicle's z axis. See Figure 22.

Lateral Acceleration - The component of the acceleration vector of the vehicle's center of gravity perpendicular to the vehicle x axis and parallel to the road plane.

Wheel Plane - The central plane of the tire, normal to the spin axis.

Slip Angle - The angle between the intersection of the wheel plane and the road plane, and the direction of travel of the center of tire contact. Due to the simplified modeling of the tires in this model, which doesn't account for tire deformation, it is not necessary to limit the definition to the center of tire contact.

Steer Angle - The angle between the projection of the longitudinal axis of the vehicle and the line of intersection of the wheel plane and the road surface.

Suspension Roll - The rotation of the vehicle sprung mass about the x axis with respect to a transverse axis joining a pair of wheel centers or line parallel to the vehicle y axis through the bogie center.

Roll Steer - The change in steer angle of front or rear wheels due to suspension roll.

Compliance Steer - The change in steer angle of front or rear wheels resulting from compliance in suspension and steering linkages and produced by forces and/or moments applied at the tire-road contact. This includes roll steer as well as steering changes due to compliance about other vehicle degrees of freedom.

Neutral Steer - The condition where the steering angle does not have to change in order to maintain a constant radius turn at increasing levels of lateral acceleration.[13]

Understeer - The condition where the steering angle has to increase in order to maintain a constant radius turn at increasing levels of lateral acceleration. This is often described as "plowing" or "pushing" when referring to the handling characteristics of a vehicle.

Oversteer - The condition where the steering angle has to decrease in order to maintain a constant radius turn at increasing levels of lateral acceleration. This is the feeling that a vehicle's rear end is "swinging" or "coming around" during extreme cornering maneuvers.

Note: For this paper the phrase "vehicle's center of gravity" has been substituted for the phrase "a point in the vehicle" in the SAE definitions. This was possible due to the limited scope of this investigation. For other investigations the more general definitions would be proper.

GLOBAL AND VEHICLE COORDINATE SYSTEMS

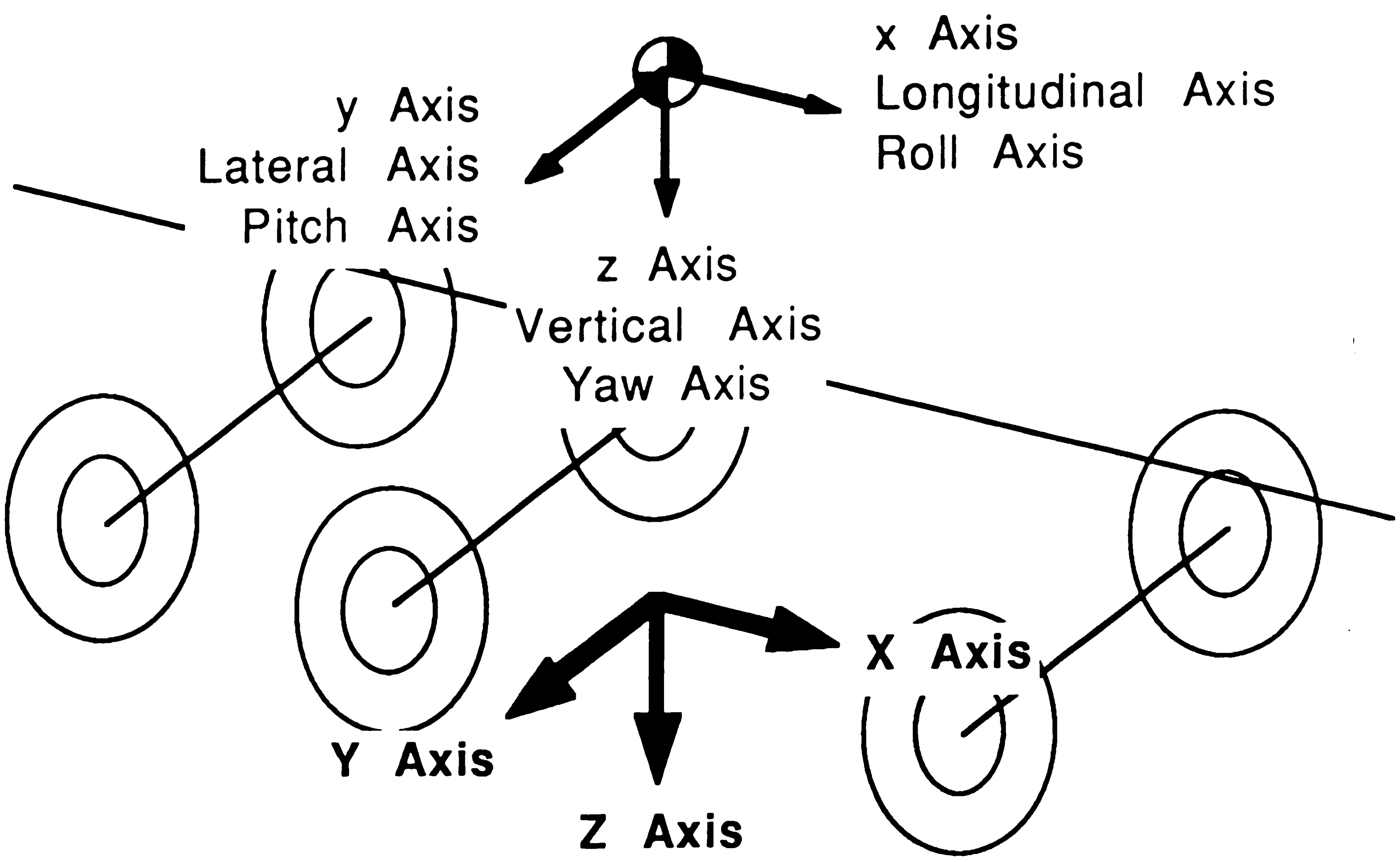


Figure 22

APPENDIX B - MODEL DESCRIPTION

The model used for this study is based on a model developed at the University of Michigan Transportation Research Institute (UMTRI). The original model was created by Yoram Guy, to examine the rollover characteristics of tractor/semi-trailer/trailer combination. This model is used for demonstration purposes by Mechanical Dynamics Incorporated (MDI). MDI supplied a listing of the model's ADAMS code for use in this study.

The original model consists of four separate units: a 4x2 tractor, semi-trailer #1, an intermediate dolly, and semi-trailer #2. Each unit is modeled as rigid. For short, box type trailers this is a reasonable assumption. The 4x2 tractor is connected to the first by a conventional "fifth wheel" to a 27 ft. semi-trailer. A single axle dolly is connected to the rear of this first trailer. The dolly contains another "fifth wheel", which supports the front end of a second 27 ft. trailer. The entire combination has a gross vehicle weight of 80,000 pounds. The tractor wheelbase is 120 inches. This wheelbase is short enough that the simplification of a rigid tractor chassis should be valid. Longer wheelbases require that frame flexibility be taken into account. The tractor steers by pivoting the front axle (AXLE 1) about its center, as viewed in the X-Y plane. This is a simplification of the actual steering linkages found on prototype trucks, but was sufficient for the study.

The new model is a 6x4 tractor connected to a single 27 ft. semi-trailer. It was created from the original by first removing the rear trailer and the dolly. A second drive axle (AXLE 3) was created by copying the original drive axle (AXLE 2). The distance between the two drive axles (also called the "bogie wheelbase") was set to 50 inches. The chassis wheelbase, as measured to the center of the bogie, was kept at 120 inches. This maintains the simplification of a rigid frame, but created a 6x4 tractor that may be too short to be practical (due to the extreme variation in driveshaft universal joint angles and shaft lengths that would be encountered during operation).

Each axle was located by three links. A center link was constrained to move only in a vertical plane, passing through the vehicle centerline. This restrained the axle from moving laterally. Two additional links were then oriented parallel to the center link, and displaced in the horizontal and vertical directions. These three links formed a parallelogram mechanism, which allowed the axle to move vertically and twist about its center.The complete rear suspension is shown in Figure 10. The vertical distance between the two ends of the rear suspension links was used as the variable for the parametric study ("h" in Figures 1 and 2). For the roll steer model the chassis mounting points for the links are towards the center of the bogie. For the non-roll steer model the chassis mounting points are towards the front of the chassis.

Spring-damper elements were located at the four axle mounting locations of the outboard links. Since these elements were no longer along the chassis centerline and were now able to produce a moment about the chassis roll axis, the rotational springs and dampers from the original model were removed. The damping coefficients were increased from the original model to reduce the oscillation of the vehicle about its roll axis.

The model of the tire forces were used directly from the original model. The suspension forces for the rear suspension were scaled to give similar deflection under normal load.

The next section discusses the details of one of the ADAMS models used for the parametric study. A block of ADAMS commands and keywords will be shown in uppercase characters. A description of the block will then start on the next line.[14]

To understand the model there are several important ADAMS terms, or "keywords", and concepts that should be understood. An ADAMS model consists of a series of PARTS that are connected kinematically by constraints, or dynamically by forces. Each part has six global degrees of freedom (DOF). The constraints and forces used to remove or limit these DOF's. The ADAMS parts location is defined in terms of a global coordinate system. Only a part may have mass properties (i.e. mass, three moments of inertia, and three products of inertia). If a part is fully kinematically constrained, with all six DOF's removed, then it is not necessary for it to have any mass properties. This was done for the suspension links, since their mass properties would be very small when compared to those of the other parts. Each part also has its own local coordinate system, or "local part reference system" (LPRF). These are useful when trying to visualize the motion of the model.

Specific points (such as constraint locations and mass centers) are defined by MARKERS. A marker must be defined as belonging to a specific part. It moves with the part during the analysis.

Since one part is always defined to be GROUND (the inertial frame of reference) any markers that are desired at a fixed location are attached to that "ground" part. Each marker may also have its own local coordinate system. These are usually used when defining the kinematic constraints. Additionally, any information requested from the ADAMS analysis is reported in terms of a reference marker's local coordinate system. Additional keywords will be described in the following text, where appropriate.

```
!6X4/SEMI - ROLLSTEER REAR SUSP H=0
```

This is the title of the particular ADAMS file. It is echoed in all the ADAMS output files. The "!" indicates to ADAMS that the text following the exclamation mark is not to be executed during the analysis.

```
PART/9999, GROUND    ! ORIGIN OF GROUND REFERENCE FRAME IS @ ROAD
                     ! LEVEL, RIGHT BELOW CENTERLINE OF AXLE 1 @ T=0.
MARKER/9999, QP=0,0,0              !("SAE" AXIS SYSTEM)
MARKER/999,  QP=0,0,20, REULER=0,180D,0  !(Z-AXIS UPWARDS)
MARKER/99,   QP=0,0,0, REULER=180D,90D,0 !(Z-AXIS SIDEWARDS)
```

This block of ADAMS commands sets up the reference part for the system, in this case the road surface. The GROUND keyword indicates that this part will remain fixed in space during the run. Marker 999 is oriented differently from the global coordinate system, and will be used for the tire force definitions.

```
ACCGRAV/ GC=386.2, KGRAV=386.2
```

This command sets up the system of units to be used for the analysis.By using 386.2 for the gravitational constant it is implied that the following units will be used: inches, seconds, lbm and lbf. The KGRAV keyword indicates that the acceleration due to gravity acts in the positive Z direction.

```
PART/01, MASS=10000, CM=01, VX=1093.6,
, IP=5793000,28965000,28965000     ! TRACTOR (LPRF=GROUND @ T=0)
MARKER/01,  QP=-25,0,-44         ! C.G.
```

```
MARKER/0111, QP=-25,0,-44, REULER=0,-90D,0 ! C.G.
MARKER/0155, QP=-106,0,-40, REULER=180D,90D,0 ! 5thWHL TRUNNIONS
MARKER/0110, QP=0,0,-23          !@ AXLE 1 R.C.
MARKER/0120, QP=-95,0,-29        !@ AXLE 2 R.C.
MARKER/0130, QP=-145,0,-29       !@ AXLE 3 R.C.
MARKER/2290, QP=-120,0,0, REULER=0,-90D,0 !@ BOGIE CENTER PAD
```

This block defines the tractor. It has a mass of 10,000 lbm, and an initial velocity of 1093.6 in/s**2 (60 MPH). The center of mass is defined by MARKER 01. The moments of inertia are given by the values after the IP keyword. Note that only three values are given (Ixx, Iyy, Izz). Because the three products of inertia (Pxy, Pyz, Pzx) are not explicitly stated they default to zero. Additional markers are created to serve as portions of later JOINT commands, or at locations of interest. Some have rotated local reference coordinate frames to allow proper joint creation or sign convention. This is accomplished by the REULER keyword, which indicates the values for an Euler angle rotation. Again note the use of exclamation marks to include descriptions as part of the individual command line. MARKER 0155 is used to define the location of the trailer connection. The Euler angles are again used rotate the marker's local reference system, in this case the marker's z axis is lying in the tractor's x-y plane and is perpendicular to the tractor's x axis. This will be used for the joint which will be discussed later.

```
PART/9901, QG=-25,0,-44               ! CG PAD
MARKER/9901, QP=0,0,0
MARKER/9902, QP=0,0,0, REULER=90D,90D,0
```

This block defines a part that will move along parallel to the ground plane, acting as the projection on the tractor's center of gravity (and its coordinate system) onto the ground plane. The motion of the part is fully kinematically constrained by the following JOINT and JPRIM statements, and thus does not require any mass properties. The part's initial velocity was omitted for the same reason.

```
JOINT/9901, I=9902, J=0111, UNIVERSAL
JPRIM/109, I=9901, J=9999, PARALLEL_AXES
```

This block contains the statements necessary to have the CG PAD to have the motion described above. The UNIVERSAL JOINT command forces the z axis of MARKER 9902 to remain perpendicular to that of MARKER 0111, with their origins remaining coincident. MARKER 0111 is oriented such that its z axis is initially parallel to the global y axis. MARKER 9902 is oriented such that its z axis is initially parallel to the global x axis. The universal joint thus maintains the z axis of MARKER 9902 pointing in the direction of tractor's motion. The PARALLEL_AXES JPRIM command forces the z axis of MARKER 9901 to remain parallel to MARKER 9999 (the global origin and coordinate system). (JPRIM's are the "building blocks for JOINTs. A JPRIM may have one to three constraints. Multiple JPRIMS may be combined to create a constraint between two parts that is not otherwise possible through the JOINT command.) Since markers 9901 and 9902 are contained in the same part that part must stay upright with respect to the ground, and point in the direction of travel. This is used to give the lateral acceleration of the tractor for all the analysis.

```
PART/9905, QG=-120,0,0                ! BOGIE CENTER PAD
MARKER/9905, QP=0,0,0
MARKER/9906, QP=0,0,0, REULER=90D,90D,0
JOINT/9905, I=9906, J=2290, UNIVERSAL
JPRIM/509, I=9905, J=9999, PARALLEL_AXES
```

The part and constraints defined in this block are very similar to those described above. The differences are that the part is attached to the tractor at the bogie center, and its origin is in the ground plane (not at the CG). This part was used to determine the slip angle of the bogie center. The ratio of the part's local y axis velocity to its local x axis velocity is the tangent of the slip angle.

```
PART/11, MASS=1200, CM=11, VX=1093.6,
, IP=1436278,100,1436278        ! AXLE 1 (LPRF COINCIDES W/ GROUND)
MARKER/11,  QP=0,0,-20          ! C.G.
MARKER/1100, QP=0,0,-23, REULER=90D,90D,0   ! R.C.
MARKER/3191, QP=0,-40,0         ! LEFT TIRE CONTACT PATCH
MARKER/3292, QP=0,40,0          ! RIGHT TIRE CONTACT PATCH
```

This block defines the tractor's front axle. It has mass properties and initial velocity. The markers defining the tire contact patches are located at the wheel centers. They will be used with the axle pads to determine the tire forces. MARKER 1100 is the roll center of the axle. It is the midpoint of a line between the wheel centers. It is not coincident with the center of gravity due to the shape of a typical heavy duty truck front axle, they dip in the center to provide additional clearance to the engine oil pan.

```
PART/12, MASS=2300, QG=-95,0,0, CM=12, VX=1093.6,
```

```
, IP=1721680,100,1721680        ! AXLE 2 (LPRF @ AXLE C.L. & ROAD)
MARKER/12,  QP=0,0,-20          ! C.G.
MARKER/1200, QP=0,0,-29, REULER=90D,90D,0   ! R.C.
MARKER/1291, QP=0,-42,0          ! LEFT (OUTER) TIRE CONTACT PATCH
MARKER/1292, QP=0,42,0           ! RIGHT (OUTER) TIRE CONTACT PATCH
```

This block defines the tractor's front rear axle. It is similar to the block defining the front axle. Again the center of gravity is not located at the midpoint of the axle, due to the prototype's geometry.

```
PART/121, QG=-120,0,0, CM=1211, VX=1093.6,
, IP=0,0,0                  ! CENTER LINK FOR FRONT REAR AXLE
MARKER/1211,  QP=0,0,-29, ZP=0,0,-50   ! FRAME SIDE - REV JOINT
MARKER/1212,  QP=25,0,-29        ! AXLE PIN
```

This is the first of six rear axle locating links. It is fully constrained and its mass would be insignificant when compared to the tractor and axles, so no mass properties were assigned. The initial velocity could have also been left off, but by including it the simulation reached equilibrium in fewer iterations. Only two markers are required for each link. The first is for the link's connection to the tractor. The second is for the connection to axle. The remaining links are essentially identical, only the marker numbers and locations change.

```
PART/122, QG=-120,0,0, CM=1221, VX=1093.6,
, IP=0,0,0                  ! LEFT LINK FOR FRONT REAR AXLE
MARKER/1221,  QP=0,-17,-20, ZP=0,-7,-20  ! FRAME SIDE - CYL
MARKER/1222,  QP=25,-17,-20       ! AXLE PIN
```

This block defines the second suspension link. See the block describing the first link for more details.

```
PART/123, QG=-120,0,0, CM=1231, VX=1093.6,
, IP=0,0,0                  ! RIGHT LINK FOR FRONT REAR AXLE
MARKER/1231,  QP=0,17,-20, ZP=0,7,-20   ! FRAME SIDE - CYL
MARKER/1232,  QP=25,17,-20        ! AXLE PIN
```

This block defines the third suspension link. See the block describing the first link for more details.

```
PART/13, MASS=2300, QG=-145,0,0, CM=13, VX=1093.6,
, IP=1721680,100,1721680        ! AXLE 3 (LPRF @ AXLE C.L. & ROAD)
MARKER/13,  QP=0,0,-20          ! C.G.
MARKER/1300, QP=0,0,-29, REULER=90D,90D,0   ! R.C.
MARKER/1391, QP=0,-42,0          ! LEFT (OUTER) TIRE CONTACT PATCH
MARKER/1392, QP=0,42,0           ! RIGHT (OUTER) TIRE CONTACT PATCH
```

This block defines the tractor's rear rear axle. It was duplicated from Axle 2. The marker numbers were changed to prevent duplication, and their locations were moved 50 inches rearward.

```
PART/131, QG=-120,0,0, CM=1311, VX=1093.6,
, IP=0,0,0                  ! CENTER LINK FOR REAR REAR AXLE
MARKER/1311,  QP=0,0,-29, ZP=0,0,-50  ! FRAME SIDE - REV JOINT
MARKER/1312,  QP=-25,0,-29       ! AXLE PIN
```

This block defines the fourth suspension link. See the block describing the first link for more details.

```
PART/132, QG=-120,0,0, CM=1321, VX=1093.6,
, IP=0,0,0                  ! LEFT LINK FOR REAR REAR AXLE
MARKER/1321,  QP=0,-17,-20, ZP=0,-7,-20  ! FRAME SIDE - CYL
MARKER/1322,  QP=-25,-17,-20      ! AXLE PIN
```

This block defines the fifth suspension link. See the block describing the first link for more details.

```
PART/133, QG=-120,0,0, CM=1331, VX=1093.6,
, IP=0,0,0                  ! RIGHT LINK FOR REAR REAR AXLE
MARKER/1331,  QP=0,17,-20, ZP=0,7,-20  ! FRAME SIDE - CYL
MARKER/1332,  QP=-25,17,-20       ! AXLE PIN
```

This block defines the sixth suspension link. See the block describing the first link for more details.

```
PART/02, MASS=30500, QG=-106,0,0, CM=02, VX=1093.6,
, IP=4.055E7,1.873E8,1.873E8     ! SEMI 1 (LPRF @ KINGPIN & ROAD)
MARKER/02,  QP=-138,0,-81        ! C.G.
MARKER/0255, QP=0,0,-40          ! KINGPIN
MARKER/0230, QP=-270,0,-29        ! @ AXLE 3 R.C.
```

This block defines the trailer. The trailer has mass properties and velocity. Markers are used to define its center of gravity, tractor connection (kingpin) and the axle reference center.

```
PART/23, MASS=1500, QG=-376,0,0, CM=23, VX=1093.6,
, IP=1583420,100,1583420        ! AXLE 4 (LPRF @ AXLE C.L. & ROAD)
MARKER/23, QP=0,0,-20           ! C.G.
MARKER/2300, QP=0,0,-29, REULER=90D,90D,0   ! R.C.
MARKER/2391, QP=0,-42,0         ! LEFT TIRE CONTACT PATCH
MARKER/2392, QP=0,42,0          ! RIGHT TIRE CONTACT PATCH
```

This block defines the trailer axle. It is the same as the other axle definitions.

```
!              HITCH CONSTRAINTS
!        1) TRACTOR-SEMI#1 FIFTH-WHEEL:
JOINT/55, I=0155, J=0255, UNIVERSAL
```

This block defines the tractor/trailer connection. A UNIVERSAL joint constrains the two markers (one in the tractor, one in the trailer) to remain coincident, but allows each marker to rotate about the others z axis. This allows the trailer to pitch and yaw relative to the tractor.

```
!            AXLE CONSTRAINTS
!        Z-TRANSLATIONAL + X-REVOLUTE @ R.C. MARKERS:
PART/10, QG=0,0,-23             ! AXLE 1 - R.C. SLIDER/PIVOT
MARKER/1011, QP=0,0,0, REULER=90D,90D,0 ! PIVOT
MARKER/1001, QP=0,0,0           ! SLIDER
MARKER/1000, QP=0,0,10          ! SPRING SEAT
JOINT/1011, I=1011, J=1100, REVOLUTE   ! FOR ROLL
JOINT/1001, I=1001, J=0110, CYLINDRICAL ! FOR BOUNCE & STEER
```

This block defines the front axle/tractor connections. A phantom part is defined at the axle's roll center. This phantom part is fully kinematically constrained and therefore requires no mass properties, and has no dynamic effect on the remainder of the system. The REVOLUTE joint between the phantom part and the front axle constrain the axle to rotate about an axis that is (initially) parallel to the chassis centerline, without any translation with respect to the phantom part. The CYLINDRICAL joint then constrains the phantom part to rotate about a vertical axis to provide steering motion, and translate along that same axis to allow for suspension compliance.

```
! AXLE 2 CONNECTIONS
MARKER/1021, PART=01, QP=-120,0,-29,  ZP=-120,0,-30
MARKER/1022, PART=01, QP=-120,-17,-20, ZP=-120,-17,-21
MARKER/1023, PART=01, QP=-120,17,-20,  ZP=-120,17,-21
MARKER/2012, PART=12, QP=0,0,-29
MARKER/2022, PART=12, QP=0,-17,-20
MARKER/2032, PART=12, QP=0,17,-20
JOINT/2011,  I=1021, J=1211, REVOLUTE
JOINT/2021,  I=1022, J=1221, UNIVERSAL
JOINT/2031,  I=1023, J=1231, UNIVERSAL
JOINT/2012,  I=1212, J=2012, SPHERICAL
JOINT/2022,  I=1222, J=2022, SPHERICAL
JOINT/2032,  I=1232, J=2032, SPHERICAL
```

The connections for both of the tractor's rear axles are identical in form, differing only in the numberings and locations. The markers defined here are attached to specific parts (by the PART= keyword), instead of defaulting to the previous PART statement. This was done to make changing the tractors suspension height less cumbersome, most of the markers that need to be modified are located in these sections. A REVOLUTE joint is used between the tractor and the center link, with the axis of rotation perpendicular to the tractors local x-z plane. A SPHERICAL joint is used between the center link and the axle. This combination of joints constrains the center link, and thus the center of the axle to remain in that plane. UNIVERSAL joints are used between the frame and the outboard links. They allow the links to rotate out of planes parallel to that of the center link. This permits the axle to twist relative to the tractor as the tractor leans in turns (or if one side were jounced higher than the other). SPHERICAL joints are also used at the axle end of the outboard links. The three links are initially parallel, and non-coplanar. This is done to prevent the axles from twisting about the tractor's local y axis under braking or other tire induced torques.

```
! AXLE 3 CONNECTIONS
MARKER/1031, PART=01, QP=-120,0,-29,  ZP=-120,0,-30
MARKER/1032, PART=01, QP=-120,-17,-20, ZP=-120,-17,-21
MARKER/1033, PART=01, QP=-120,17,-20,  ZP=-120,17,-21
MARKER/3012, PART=13, QP=0,0,-29
MARKER/3022, PART=13, QP=0,-17,-20
MARKER/3032, PART=13, QP=0,17,-20
JOINT/3011,  I=1031, J=1311, REVOLUTE
JOINT/3021,  I=1032, J=1321, UNIVERSAL
JOINT/3031,  I=1033, J=1331, UNIVERSAL
```

```
JOINT/3012, I=1312, J=3012, SPHERICAL
JOINT/3022, I=1322, J=3022, SPHERICAL
JOINT/3032, I=1332, J=3032, SPHERICAL
```

The connections defined in this block are functionally identical to those described in the previous block. See that block for further description.

```
PART/30, QG=-376,0,-29            ! AXLE 4 - R.C. SLIDER/PIVOT
MARKER/3023, QP=0,0,0, REULER=90D,90D,0 ! PIVOT
MARKER/3002, QP=0,0,0             ! SLIDER
MARKER/3000, QP=0,0,10            ! SPRING SEAT
JOINT/3023, I=3023, J=2300, REVOLUTE   ! FOR ROLL
JOINT/3002, I=3002, J=0230, TRANSLATIONAL ! FOR BOUNCE
```

The connections for the trailer axle are defined in this block. A phantom block is again used to get the proper constraints. The axle is connected to the phantom part by the REVOLUTE joint, and can pivot about an axis parallel to the trailer's centerline. The phantom part may then move along a line perpendicular to the ground plane, to give the axle the capability for vertical travel. MARKER 3000 will be used for the trailer suspension definitions.

```
!          SUSPENSIONS - BOUNCE STIFFNESS
MARKER/1051, PART=01, QP=-95,-17,-29
MARKER/1052, PART=01, QP=-95,17,-29
MARKER/1053, PART=01, QP=-145,-17,-29
MARKER/1054, PART=01, QP=-145,17,-29
SPRING/11, I=1000, J=0110, TRANSL, C=20, K=2400, L=14.01 ! AXLE 1
SPRING/121, I=1222, J=1051, TRANSL, C=20, K=8000, L=9    ! AXLE 2
SPRING/122, I=1232, J=1052, TRANSL, C=20, K=8000, L=9    ! AXLE 2
SPRING/131, I=1322, J=1053, TRANSL, C=20, K=8000, L=9    ! AXLE 3
SPRING/132, I=1332, J=1054, TRANSL, C=20, K=8000, L=9    ! AXLE 3
SPRING/23, I=3000, J=0230, TRANSL, C=20, K=20000, L=10.79 ! AXLE 4
```

This block defines the vertical stiffness of the tractor and trailer suspensions. The SPRING statements are specialized forms of ADAMS FORCE statements. Springs apply action/reaction forces between two parts (as opposed to action only forces such as body forces) of the form:

$$F = -K(x1-x2)-C(v1-v2)$$

where the sign convention is such that a spring force resists the motion of the markers towards each other. ADAMS allows (spring) force statements to have a free length, be compression or extension only, or to act only on a single part (as a body force) instead of between two parts. These springs take the defaults for these options. The TRANSLation keyword defines the force to be translational instead of rotational.

```
!           SUSPENSIONS - ROLL STIFFNESS
!INCREASING DAMPING BY A FACTOR OF 10 - JES 4/23/90
SPRING/1, I=1011, J=1100, ROTATION, CT=286478, KT=699008.4  ! AXLE 1
SPRING/3, I=3023, J=2300, ROTATION, CT=286478, KT=7792226.0 ! AXLE 4
```

This block defines the roll stiffness for the tractor front axle and the trailer axle. ROTATIONal springs are used to provide restoring moments at the axle pivot locations. These statements are functionally identical to those in the preceding block. No corresponding springs are used for the tractors rear axles. Since the vertical springs are located outboard of the tractor centerline they also create a restoring moment when the tractor leans.

```
!           TIRES - ROAD PADS

PART/9911, QG=0,0,0                      ! AXLE 1 PAD
MARKER/9111, QP=0,-40,0
MARKER/9112, QP=0,40,0, REULER=90D,90D,0     ! Z AXIS - FORWARDS
MARKER/9110, QP=0,0,0, REULER=180D,90D,0     ! Z AXIS - SIDEWAYS
JOINT/9111, I=9111, J=9999, PLANAR
JPRIM/1119, I=3191, J=9111, INLINE
JPRIM/2119, I=3292, J=9112, INPLANE
```

This block, and the following three, define the axle pads. These phantom parts act as the projection of the individual axles onto the ground plane. The PLANAR joint constrains the axle pad to always remain with its local origin in the ground plane (the global X-Y plane) and its local z axis parallel to the global Z axis. The INLINE jprim constrains marker 3191 (the front axle's left tire patch) to stay inline with local z axis of marker 9111. The final INPLANE jprim constrains marker 3292 to remain in the local x-y plane 9112, without imposing any restrictions on its orientation. This combination of joints and jprims gives the following result: The pad translates (in the ground plane) with the front axle. Since markers 9112 and 9111 are always directly below markers 3191 and 3292 the axle pivots with the axle. The pad must remain in the ground plane, so the angle of the pivot is the projection on the axle's rotation onto the ground plane. Marker 9110 is used by a later block, to define the tire forces.

```
PART/9912, QG=-95,0,0                    ! AXLE 2 PAD
MARKER/9121, QP=0,-42,0
MARKER/9122, QP=0,42,0, REULER=90D,90D,0    ! Z AXIS - FORWARDS
MARKER/9120, QP=0,0,0, REULER=180D,90D,0    ! Z AXIS - SIDEWAYS
JOINT/9121, I=9121, J=9999, PLANAR
JPRIM/1219, I=1291, J=9121, INLINE
JPRIM/2219, I=1292, J=9122, INPLANE
```

This block is functionally equivalent to that of AXLE 1 PAD.

```
PART/9913, QG=-145,0,0                   ! AXLE 3 PAD
MARKER/9131, QP=0,-42,0
MARKER/9132, QP=0,42,0, REULER=90D,90D,0    ! Z AXIS - FORWARDS
MARKER/9130, QP=0,0,0, REULER=180D,90D,0    ! Z AXIS - SIDEWAYS
JOINT/9131, I=9131, J=9999, PLANAR
JPRIM/1319, I=1391, J=9131, INLINE
JPRIM/2319, I=1392, J=9132, INPLANE
```

This block is functionally equivalent to that of AXLE 1 PAD.

```
PART/9923, QG=-376,0,0                   ! AXLE 4 PAD
MARKER/9231, QP=0,-42,0
MARKER/9232, QP=0,42,0, REULER=90D,90D,0    ! Z AXIS - FORWARDS
MARKER/9230, QP=0,0,0, REULER=180D,90D,0    ! Z AXIS - SIDEWAYS
JOINT/9231, I=9231, J=9999, PLANAR
JPRIM/1329, I=2391, J=9231, INLINE
JPRIM/2329, I=2392, J=9232, INPLANE
```

This block is functionally equivalent to that of AXLE 1 PAD.

```
! NOTE: PADS ARE CONSTRAINED TO YAW WITH SPINDLES IN ROAD PLANE
!    - DEFINING PROJECTION OF SPINDLE X-Y AXES ONTO ROAD PLANE
```

The following blocks are used to define the three tire forces and moments used in this model. ADAMS also has an optional ADAMS/TIRE package that calculates six tire forces and moments. It uses much more complex algorithms to determine the magnitudes than is applied here. The ADAMS/TIRE package is not currently available at Lehigh University, so the original tire force models were retained.

! TIRES - VERTICAL STIFFNESS

The following eight blocks define the vertical tire forces for the model. All of the blocks are functionally identical, with only the parameters changed (force numbers, marker numbers and the tire stiffnesses). SFORCEs are used for the tire forces. These are generalized forces. The TRANSLATION keyword is used to specify that a force is the result, not a torque. The ACTIONONLY keyword means that the force acts on the tire only, not on the tire and the ground. A FUNCTION statement is used to define the actual magnitude of the force. The function chosen from the ADAMS library is the IMPACT function. The parameters involved in the IMPACT function are: a distance variable, a velocity variable, a free length, a spring constant, an exponent constant, a damping coefficient constant, and a damping penetration constant. The value of the IMPACT function is always positive. Therefore the force is always directed upwards from the ground plane. For the vertical tire stiffness the vertical separation (DZ) between two markers (in the frame of reference of the second marker) is used for the distance variable. The rate of change of this distance (VZ) is used for the velocity variable. The free length is the tire's loaded radius. The spring constant is the tire's (or pair of tires for the tractor rear axles and the trailer axle) radial stiffness. Unity was used for the exponent value. The maximum damping coefficient is two for all tires. This maximum damping is fully applied at penetration equal to one times the free length. This results in the following force equation:

SFORCE = 0, for DZ > Tire Loaded Radius

SFORCE = Radial Stiffness * (DZ - Tire Loaded Radius) ** 1 - (VZ * 2)

The (VZ * 2) term is applied in at a non-linearly increasing rate as the tire deformation approaches the loaded radius. A more complete description of the IMPACT function is contained in the ADAMS User's Manual.[15]

The following two blocks define the vertical tire stiffness for the tractor's front tires. The vertical separation is measured between the tire contact patches (markers 3191 and 3292). The tire loaded radii are 21.203 inches. The radial stiffnesses are 4500 lbf per inch.

```
!       AXLE 1 - LEFT TIRE
SFORCE/191, I=3191, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(3191,999,999),VZ(3191,999,999),21.203,4500,
, 1,2,1)

!       AXLE 1 - RIGHT TIRE
SFORCE/192, I=3292, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(3292,999,999),VZ(3292,999,999),21.203,4500,
, 1,2,1)
```

The next four blocks define the vertical stiffnesses for the tractor's rear tire pairs. The loaded radii are slightly less than those of the front tires. The stiffnesses are doubled.

```
!       AXLE 2 - LEFT TIRE
SFORCE/291, I=1291, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(1291,999,999),VZ(1291,999,999),20.964,9000,
, 1,2,1)

!       AXLE 2 - RIGHT TIRE
SFORCE/292, I=1292, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(1292,999,999),VZ(1292,999,999),20.964,9000,
, 1,2,1)

!       AXLE 3 - LEFT TIRE
SFORCE/391, I=1391, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(1391,999,999),VZ(1391,999,999),20.964,9000,
, 1,2,1)

!       AXLE 3 - RIGHT TIRE
SFORCE/392, I=1392, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(1392,999,999),VZ(1392,999,999),20.964,9000,
, 1,2,1)
```

The trailer axle tires' vertical stiffnesses are defined in the next two blocks. The force parameters are identical to those of the tractor rear axles.

```
!       AXLE 4 - LEFT TIRE
SFORCE/491, I=2391, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(2391,999,999),VZ(2391,999,999),20.962,9000,
, 1,2,1)
```

```
!       AXLE 4 - RIGHT TIRE
SFORCE/492, I=2392, J=999, TRANSLATION, ACTIONONLY,
, FUNCTION=IMPACT(DZ(2392,999,999),VZ(2392,999,999),20.962,9000,
, 1,2,1)
```

The tire cornering stiffnesses (lateral stiffnesses) are defined in the following eight blocks. The definitions are very similar to those of the vertical tire stiffnesses, with the exception of an ADAMS ARITHMETIC IF[16] function replacing the IMPACT function. The IF function has four expressions as parameters. The first expression controls which of the other three are used to determine the magnitude of the force. If the first expression is less than zero the ARITHMETIC IF equals the second expression. If the first expression equals zero, then the IF equals the third expression. IF the first expression is greater than zero, then the IF equals the final expression. The value of the IF is multiplied by a negative coefficient. This negative value serves to attract the first marker instead of repel it, as in the case of the vertical stiffness forces.

These IF functions all use the simulation time as the first expression. For time less than or equal to zero the IF equals zero. For time greater than zero it interprets the following expression: The absolute value of the tire's vertical force is divided by a value. The square root of the result is then multiplied by the tire's slip angle. The force is equal to this result multiplied by another parameter. Note that the tire's slip angle is defined as the arctangent of the ratio of the tire contact patch's lateral velocity to its longitudinal velocity. The RTOD keyword converts the arctangent result from degrees to radians.

```
!       TIRES - CORNERING STIFFNESS
```

The following two blocks define the cornering stiffnesses of the tractor's front tires. The two force parameters used here are the -800 lbf per radian, and the 6000, which is used to scale the tires' vertical forces.

```
!       AXLE 1 - LEFT TIRE
SFORCE/161, I=3191, J=9110, TRANSLATION, ACTIONONLY,
, FUNCTION=-800*IF(TIME:0,0,SQRT(ABS(FZ(3191,0,999))/6000)*
,RTOD*ATAN2(VY(3191,9999,9111),VX(3191,9999,9111)))
```

```
!       AXLE 1 - RIGHT TIRE
SFORCE/162, I=3292, J=9110, TRANSLATION, ACTIONONLY,
, FUNCTION=-800*IF(TIME:0,0,SQRT(ABS(FZ(3292,0,999))/6000)*
,RTOD*ATAN2(VY(3292,9999,9111),VX(3292,9999,9111)))
```

The next six blocks define the cornering stiffnesses for the tractor's rear tires and the trailer's tires. The values of the two force parameters are 1600 lbf per radian and a scaling factor of 12000.

```
!       AXLE 2 - LEFT TIRE
SFORCE/261, I=1291, J=9120, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(1291,0,999))/12000)*
,RTOD*ATAN2(VY(1291,9999,9121),VX(1291,9999,9121)))

!       AXLE 2 - RIGHT TIRE
SFORCE/262, I=1292, J=9120, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(1292,0,999))/12000)*
,RTOD*ATAN2(VY(1292,9999,9121),VX(1292,9999,9121)))

!       AXLE 3 - LEFT TIRE
SFORCE/361, I=1391, J=9130, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(1391,0,999))/12000)*
,RTOD*ATAN2(VY(1391,9999,9131),VX(1391,9999,9131)))

!       AXLE 3 - RIGHT TIRE
SFORCE/362, I=1392, J=9130, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(1392,0,999))/12000)*
,RTOD*ATAN2(VY(1392,9999,9131),VX(1392,9999,9131)))

!       AXLE 4 - LEFT TIRE
SFORCE/461, I=2391, J=9230, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(2391,0,999))/12000)*
,RTOD*ATAN2(VY(2391,9999,9231),VX(2391,9999,9231)))

!       AXLE 4 - RIGHT TIRE
SFORCE/462, I=2392, J=9230, TRANSLATION, ACTIONONLY,
, FUNCTION=-1600*IF(TIME:0,0,SQRT(ABS(FZ(2392,0,999))/12000)*
,RTOD*ATAN2(VY(2392,9999,9231),VX(2392,9999,9231)))

!       TIRES - ALIGNING TORQUE STIFFNESS
```

The tire aligning torque stiffnesses are moments that tend to restore the tires' direction of

travel to straight ahead. They are almost identical in form to the cornering stiffnesses. The differences are that the ROTATION keyword defines the SFORCES as torques, and the fourth expression in the ARITHMETIC IF function has the magnitude raised to the 1.3 power and a constant of 0.0001 added.

The tractor's front tires use parameter values of 1400 inch-lbf per degree ~~radian~~, and a scaling value of 6000.

```
!        AXLE 1 - LEFT TIRE
SFORCE/171, I=3191, J=9111, ROTATION, ACTIONONLY,
, FUNCTION=-1400*IF(TIME:0,0,(.0001+ABS(FZ(3191,0,999))/6000)**1.3*
, RTOD*ATAN2(VY(3191,9999,9111),VX(3191,9999,9111)))

!        AXLE 1 - RIGHT TIRE
SFORCE/172, I=3292, J=9111, ROTATION, ACTIONONLY,
, FUNCTION=-1400*IF(TIME:0,0,(.0001+ABS(FZ(3292,0,999))/6000)**1.3*
, RTOD*ATAN2(VY(3292,9999,9111),VX(3292,9999,9111)))
```

The tractor's rear tires and the trailer tires use parameters of 2800 inch-lbf per degree ~~radian~~, and a scaling value of 12000.

```
\!        AXLE 2 - LEFT TIRE
SFORCE/271, I=1291, J=9121, ROTATION, ACTIONONLY,
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(1291,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(1291,9999,9121),VX(1291,9999,9121)))

!        AXLE 2 - RIGHT TIRE
SFORCE/272, I=1292, J=9121, ROTATION, ACTIONONLY,
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(1292,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(1292,9999,9121),VX(1292,9999,9121)))

!        AXLE 3 - LEFT TIRE
SFORCE/371, I=1391, J=9131, ROTATION, ACTIONONLY,
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(1391,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(1391,9999,9131),VX(1391,9999,9131)))

!        AXLE 3 - RIGHT TIRE
SFORCE/372, I=1392, J=9131, ROTATION, ACTIONONLY,
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(1392,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(1392,9999,9131),VX(1392,9999,9131)))

!        AXLE 4 - LEFT TIRE
SFORCE/471, I=2391, J=9231, ROTATION, ACTIONONLY,
```

```
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(2391,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(2391,9999,9231),VX(2391,9999,9231)))

!       AXLE 4 - RIGHT TIRE
SFORCE/472, I=2392, J=9231, ROTATION, ACTIONONLY,
, FUNCTION=-2800*IF(TIME:0,0,(.0001+ABS(FZ(2392,0,999))/12000)**1.3*
, RTOD*ATAN2(VY(2392,9999,9231),VX(2392,9999,9231)))

!        LATERAL STATIC EQUILIBRIUM AT T=0
```

When the model is reaching static equilibrium the various parts may undergo displacements. The tire forces will then produce forces which cause the model to behave improperly at initial conditions. The following SFORCE statements create forces that are directly proportional to the lateral distance between each axle and a marker at the origin. These forces restore the axles from any lateral displacement during static equilibrium calculations. The forces are only active for time less than or equal to zero.

```
SFORCE/1111, I=11, J=99, TRANSLATION, ACTIONONLY,
, FUNCTION=IF(TIME:-DZ(11,99,99),-DZ(11,99,99),0)
SFORCE/2222, I=12, J=99, TRANSLATION, ACTIONONLY,
, FUNCTION=IF(TIME:-DZ(12,99,99),-DZ(12,99,99),0)
SFORCE/3333, I=13, J=99, TRANSLATION, ACTIONONLY,
, FUNCTION=IF(TIME:-DZ(13,99,99),-DZ(13,99,99),0)
SFORCE/4444, I=23, J=99, TRANSLATION, ACTIONONLY,
, FUNCTION=IF(TIME:-DZ(23,99,99),-DZ(23,99,99),0)

! GRAPHICS INFORMATION DELETED
```

The graphics statements have been edited from this text. The graphics statements produce a wireframe image of the model for display on graphics style terminals used to visually check the motion of the model.

```
!    STEER EXCITATION - CYCLOIDAL
MOTION/1, JOINT=1001, ROTATION,
, FUNCTION=IF(TIME-0.5:0,0,
,(DTOR*30*(((TIME-0.5)/30)-SIN(2*PI*(TIME-0.5)/30)/2/PI)))
```

This block defines the steering input to the front axle during simulation. The MOTION statement defines a displacement (in this case it is rotational) for the degree of freedom of a joint. An ARITHMETIC IF function is used to start the displacement at a simulation time of 0.5 seconds. The steering input is a cycloidal cam follower motion function, with time used as the independent variable in place of cam displacement.

```
!       OUTPUT REQUESTS
```

The following blocks specify what variables ADAMS will tabulate in the output files. The parameters for the REQUEST statements are: the type of relative information requested, the two markers that define the relationship, and the reference marker (whose local coordinate system will be used to reconcile the information). The information is given as six components, usually six degrees of freedom for the requested vector.

```
!       SPRUNG MASS VELOCITIES
REQUEST/2, VELOCITY, I=01, J=9999, RM=9121,
, COMMENT=TRACTOR SPRUNG-MASS VELOCITIES (FT/SEC - DEG/SEC)
REQUEST/3, VELOCITY, I=02, J=9999, RM=9231,
, COMMENT=SEMI #1 SPRUNG-MASS VELOCITIES (FT/SEC - DEG/SEC)
```

The REQUEST statements in this block give the velocity vectors of the tractor and trailer.

```
!       SPRUNG MASS ACCELERATIONS
REQUEST/6, ACCELERATION, I=01, J=9999, RM=9121,
, COMMENT=TRACTOR SPRUNG-MASS ACCELERATIONS (G'S - DEG/SEC**2)
REQUEST/7, ACCELERATION, I=02, J=9999, RM=9231,
, COMMENT=SEMI #1 SPRUNG-MASS ACCELERATIONS (G'S - DEG/SEC**2)
```

The REQUEST statements in this block give the acceleration vectors of the tractor and trailer.

```
!       TIRE FORCES
MREQUEST/11, FORCE, APPFORS=161,191, RM=9111,
, COMMENT=AXLE 1 - LEFT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/12, FORCE, APPFORS=162,192, RM=9111,
, COMMENT=AXLE 1 - RIGHT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/21, FORCE, APPFORS=261,291, RM=9121,
, COMMENT=AXLE 2 - LEFT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/22, FORCE, APPFORS=262,292, RM=9121,
```

```
, COMMENT=AXLE 2 - RIGHT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/31, FORCE, APPFORS=361,391, RM=9131,
, COMMENT=AXLE 3 - LEFT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/32, FORCE, APPFORS=362,392, RM=9131,
, COMMENT=AXLE 3 - RIGHT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/41, FORCE, APPFORS=361,391, RM=9231,
, COMMENT=AXLE 4 - LEFT TIRE - LATERAL AND VERTICAL FORCES (LBS)
MREQUEST/42, FORCE, APPFORS=362,392, RM=9231,
, COMMENT=AXLE 4 - RIGHT TIRE - LATERAL AND VERTICAL FORCES (LBS)
```

This block requests that the tire force values be included in the output.

```
REQUEST/1111, FUNCTION=USER(9901,9111,9905,9999)\
,TITLE=HANDLING
, COMMENT=HANDLING DIAGRAM INFORMATION
```

This block requests that information developed in a user written subroutine be included in the output. The subroutine uses markers 9901, 9111, 9905 and 9999 to develop the information that is plotted as the handling diagram. This subroutine is included at the end of the main model (after the END statement). During operation it is stored in a separate FORTRAN file, and its executable code is linked to that of the ADAMS main program. Together they form a "USER_WRITTEN EXECUTABLE CODE".

```
OUTPUT/ REQSAVE, GRSAVE, FIXED, YPR, NOPRINT,
, DSCALE=0.0833333, VSCALE=0.0833333, ASCALE=0.002589332
```

This block scales the information given as output. The distance and velocity information is scaled to have units of feet, rather than inches. The accelerations are scaled to have units of "g's", instead of inches per second per second.

```
END
```

The END statement terminates the model code.

APPENDIX C - HANDLING DIAGRAM SUBROUTINE - REQSUB

The following section describes the REQSUB FORTRAN subroutine that was used to develop the handling diagram data from the raw data supplied by ADAMS. This subroutine was developed by including the necessary calculations in the body of a sample ADAMS REQSUB subroutine. The conventions used to describe the main ADAMS model in APPENDIX B will also be used in this section.

```
SUBROUTINE REQSUB (ID,TIME,PAR,NPAR,IFLAG,RESULT)
```

The subroutine call uses the following parameters: the REQUEST statement number in the main ADAMS model that calls the subroutine, a variable to keep track of simulation time, the array of constants supplied by the calling USER function (the marker numbers), the number of constants in that array, a logical variable (that is not used in this subroutine, but is required in the statement), and the array of the returned results.

```
C
C ---TYPING AND DIMENSIONING STATEMENTS --------------------------------
C
   IMPLICIT DOUBLE PRECISION (A-H, O-Z)
   INTEGER ICG,IFA,IRA,IGRND,NPAR
   DOUBLE PRECISION ACCEL(6)
   DOUBLE PRECISION V1(6)
   DOUBLE PRECISION V2(6)
   DOUBLE PRECISION RESULT(8)
   DOUBLE PRECISION PAR( * )
   DOUBLE PRECISION VEL1M
   DOUBLE PRECISION VEL2M
   LOGICAL IFLAG, ERRFLG
   CHARACTER*4 ACCINF
   CHARACTER*4 VELINF
   CHARACTER*80 MESSE
   CHARACTER*4 ACTSTP
   PARAMETER  ( ACTSTP = 'STOP' )
```

```
      SAVE
      DATA ACCINF, VELINF / 'ACC ', 'VEL ' /
      ICG = PAR(1)
      IFA = PAR(2)
      IRA = PAR(3)
      IGRND = PAR(4)
C
C SUBROUTINE INFO CALLED TO FIND VEHICLE CG LATERAL ACCELERATION
C
      CALL INFO (ACCINF, ICG, IGRND, ICG, ACCEL, ERRFLG)
      CALL ERRMES (ERRFLG, 'ERROR CALLING INFO IN REQSUB', ID, 'STOP')
```

The ADAMS subroutine INFO is used with the ACCINF keyword to return the acceleration of the tractor's CG to REQSUB. The subroutine determines the acceleration of the CG marker (ICG) with respect to the ground marker (IGRND), in the coordinate system of the CG marker. ACCEL is the resulting vector of the acceleration components. Only the y component (lateral component) will be used. The ERRMES is used to provide error messages to the output in case INFO fails.

```
C
C SUBROUTINE INFO CALLED TO FIND FRONT AXLE VELOCITY VECTOR
C
      CALL INFO (VELINF,IFA,IGRND,IFA,V1,ERRFLG)
      CALL ERRMES (ERRFLG, 'ERROR CALLING INFO IN REQSUB', ID, 'STOP')
```

In this block the VELINF keyword is used with the INFO subroutine to get the velocity vector for the tractor's front axle with respect to the ground marker, in the front axle's coordinate system. The results are stored in the vector V1.

```
C
C SUBROUTINE INFO CALLED TO FIND REAR AXLE VELOCITY VECTOR
C                         (OR BOGIE CENTER VELOCITY VECTOR)
C
      CALL INFO (VELINF,IRA,IGRND,IRA,V2,ERRFLG)
      CALL ERRMES (ERRFLG, 'ERROR CALLING INFO IN REQSUB', ID, 'STOP')
```

This block is used to get the velocity information for the tractor's rear axle, in a

manner identical to the previous block. For the 6x4 type of tractor the bogie center was used instead of an individual axle.

```
C
C PAR(1) - VEHICLE CG PAD MARKER ID
C PAR(2) - FRONT AXLE TIRE PAD MARKER ID
C PAR(3) - REAR AXLE TIRE PAD MARKER ID
C PAR(4) - GROUND MARKER ID
C NOTE: THE TIRE PADS MUST BE THE PROJECTION OF THE SPINDLES ONTO
C        THE GROUND PLANE. THESE MARKERS MUST FOLLOW STANDARD SAE
C        AXIS ORIENTATION CONVENTION.
C
  VEL1M = SQRT(V1(1)**2+V1(2)**2)
  VEL2M = SQRT(V2(1)**2+V2(2)**2)
  IF(VEL1M.EQ.0.OR.VEL2M.EQ.0) RETURN
```

This block calculates the magnitude of the axles' horizontal velocities. If either has zero magnitude the subroutine returns execution to the main program. This avoids a possible arithmetic fault on later arctangent calculations.

```
PI=3.14159
RESULT(2) = ABS(ACCEL(2))
RESULT(2) = RESULT(2)/386.2
RESULT(6) = ASIN(V1(1)/VEL1M)
RESULT(7) = ASIN(V2(1)/VEL2M)
RESULT(3) = 180*(RESULT(6) - RESULT(7))/PI
```

In this block the result vector for the subroutine is filled, and the values are scaled. The second position in the vector contains the tractor's lateral acceleration, in "g's". The third position contains the difference between the tractor's front axle and the bogie center. These values are the ordinate and abscissa of the handling diagram.

```
RETURN
END
```

These are standard FORTRAN commands to return execution to the main body of the program.

APPENDIX D

To verify the handling diagram information subroutine, REQSUB, a comparison was made between an ADAMS simulation and an analytical solution of a 4 x 2 tractor performing a steady-state turn.

The ADAMS simulation used the 4 x 2 tractor from the original model from MDI. The steering angle was specified to be a constant 2 degrees. The simulation was run for a sufficiently long period of time for the initial transient behavior to die out. The equilibrium values were 0.154 g for lateral acceleration and -0.5 degree for delta slip angle.

The following data was used for the analytical solution:

Tractor Weight = 10,000 lbf

Front Axle Weight = 1200 lbf

Rear Axle Weight = 2300 lbf

Wheelbase = 120 inches

CG to Front Axle = 25 inches

Steering Angle = 2 degrees

The following assumptions were made:

A. Chassis roll was negligable.

B. Slip angles of tires on the same axle are identical.

With these assumptions the four wheeled vehicle was "collapsed" onto its plane of symmetry along its centerline, as shown in Figure 23.

1. $\overset{\rightarrow}{+} \sum F_y = \dfrac{(W_T + W_{FA} + W_{RA})}{g} a_l = F_{A_R} \cos(\delta - \alpha_1) + F_{B_R} \cos(\alpha_2)$

2. $\curvearrowleft + M_O = I\ddot{\theta}_O = -AO \cdot F_{A_R} \cos(\delta - \alpha_1) + BO \cdot F_{B_R} \cos(\alpha_2) + M_A + M_B$

where:

W_T = weight of the tractor = 10,000 lbf

W_{FA} = weight of the front axle = 1200 lbf

W_{RA} = weight of the rear axle = 2300 lbf

g = acceleration due to earth's gravity

a_l = vehicle's lateral acceleration (in "g's")

F_{A_R} = radial component of the forces at point A

F_{B_R} = radial component of the forces at point B

δ = steering angle

α_1 = Front Axle Slip Angle (in degrees)

α_2 = Rear Axle Slip Angle (in degrees)

AO = distance from point A to point O = 25 inches

BO = distance from point B to point O = 95 inches

M_A = tire aligning torque at point A

M_B = tire aligning torque at point B

The following force and torque definitions were taken from the ADAMS model definition:

3. $F_{A_R} = 800 \cdot \left[\dfrac{F_{A_V}}{6000}\right]^{1/2} \cdot \alpha_1 \cdot 2$

4. $F_{B_R} = 1600 \cdot \left[\dfrac{F_{B_V}}{12000}\right]^{1/2} \cdot \alpha_2 \cdot 2$

5. $M_A = 1400 \cdot \left[0.0001 + \left(\frac{F_{AV}}{6000}\right)\right]^{1.3} \cdot \alpha_1 \cdot 2$

6. $M_B = 2800 \cdot \left[0.0001 + \left(\frac{F_{BV}}{12000}\right)\right]^{1.3} \cdot \alpha_2 \cdot 2$

where:

F_{AV} = vertical component of the forces at point A

F_{BV} = vertical component of the forces at point B

The following equarions were solved for F_{AV} and F_{BV}:

7. $+\uparrow \Sigma F_z = 0 = -(W_T + W_{FA} + W_{RA}) + F_{AV} + F_{BV}$

8. $\curvearrowleft + \Sigma M_A = 0 = -(W_T) \cdot AO - (W_{RA})(AO + BO) + F_{BV}(AO + BO)$

The resulting values of F_{AV} = 9947.4 lbf and F_{BV} =3552.6 lbf were used to reduce equations 3 through 6 to:

3. $F_{AR} = 2060.1\alpha_1$

4. $F_{BR} = 1741.1\alpha_2$

5. $M_A = -5402.8\alpha_1$

6. $M_B = -1151.2\alpha_2$

It was assumed that α_1, , α_2 , and δ were small. Therefore the cosine terms approximately equal 1. Equations 1 and 2 then reduced to:

1. $2079 = 2060.1\alpha_1 + 1741.1\alpha_2$

2. $0 = -56{,}905.3\alpha_1 + 164{,}253.3\alpha_2$

whichwas solved for $a_\ell = 0.154$ g to give:

$\alpha_1 = 0.78$ degrees

$\alpha_2 = 0.27$ degrees

or a delta slip angle value equal to 0.51 degrees.

The ADAMS simulation gives results that agree with the analytical solution, for a given value of lateral acceleration.

4 X 2 ANALYTICAL MODEL

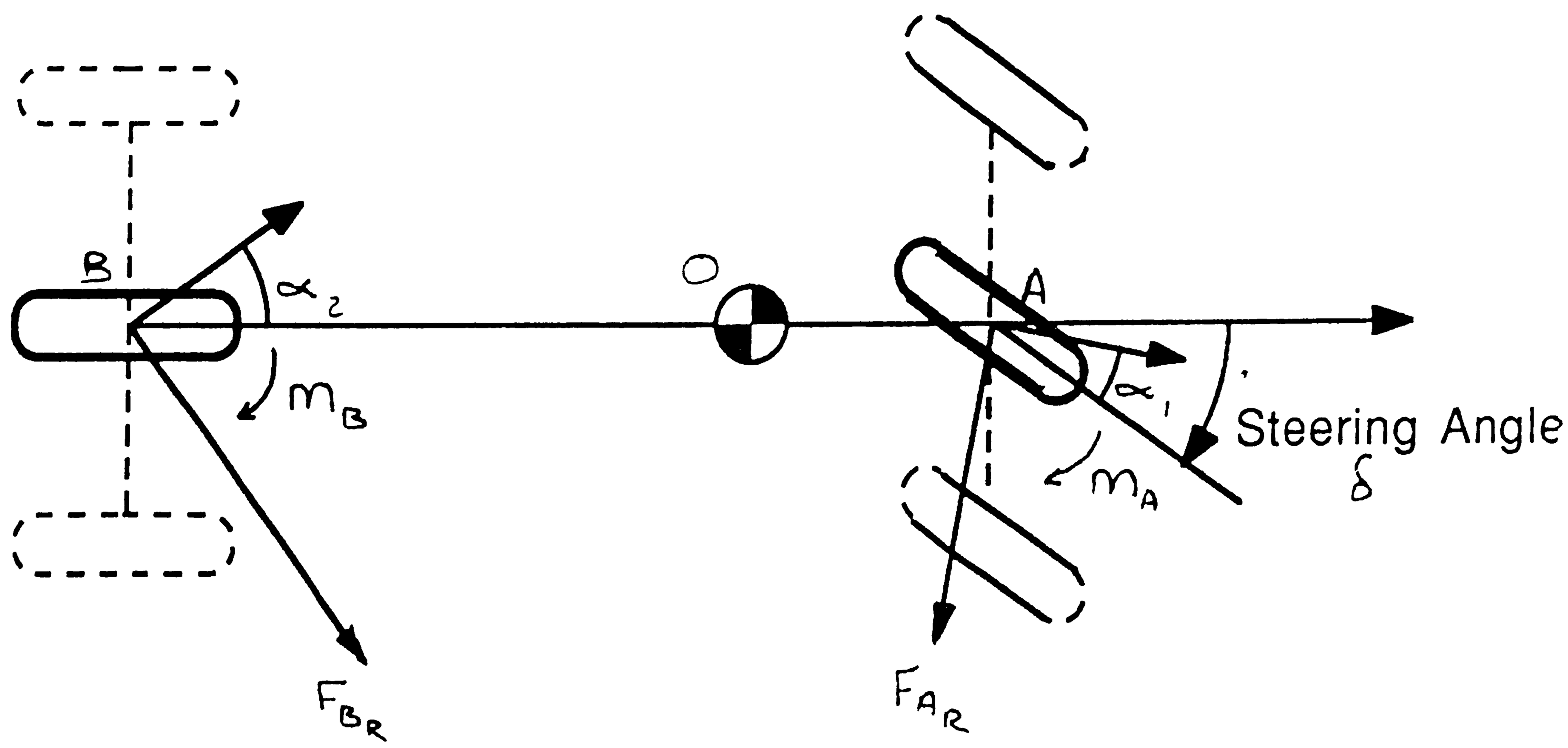


Figure 23

APPENDIX E - ADAMS COMMAND FILE

The following five lines of code, including one blank line, are the ADAMS command file for performing a dynamic analysis on the ADAMS file named TRUCK_0 (truck with roll-steer characteristics, h=0 inches). The command file allows ADAMS to be run in batch mode. A typical run of this model required about 30 CPU minutes.

The first line informs ADAMS of the model's name, in this case the model is stored in a file named "TRUCK_0.ADM". The ".ADM" file specification is not needed since it is the default.

The second line is blank to accept the default filenames for the additional files that ADAMS will create as part of the analysis. The default will use the same name, "TRUCK_0", with different file specifiers. These specifiers include: ".REQ", ".OUT", and ".LOG".

The third line contains the ADAMS command to perform a static equilibrium simulation. This allows the model to reach static equilibrium, or settle, before the dynamic analysis is performed. If this step were not included and the model was not placed in an equilibrium condition, then during the beginning of the dynamic analysis the model could undergo some large transient behaviors.

The next line contains the dynamic simulation command. The additional qualifiers indicate that the simulation should end after 15 seconds of simulation time, and that the results should be reported for 151 moments during the run. This gives the results at time equal to 0, and at 0.1 second intervals.

The final line ends the ADAMS session and the batch run.

```
TRUCK_0

SIMULATE/STATIC
SIMULATE/DYNAMIC, END=15,STEPS=151
STOP
```

REFERENCES

[1]Segal, L. et al., "Mechanics of Heavy-Duty Trucks and Truck Combinations," Proceedings of the University Of Michigan Engineering Summer Conferences, June 25-29, 1984, p.643.

[2] see note 1 above, p. 694.

[3] see note 1 above, pp.809 - 875

[4] Pacejka, H.B., "Simplified Analysis of Steady Turning Behavior of Motor Vehicles. PART 1." Vehicle System Dynamics,Vol. 2, No. 4, December 1973, pp. 161-172.

[5] see note 1 above, p. 686.

[6] see note 4 above, p. 170.

[7] see note 1 above, p. 643.

[8] see note 1 above, p. 738.

[9] Chen, F.Y. Mechanics and Design of Cam Mechanisms, New York: Pergamon Press, 1982, p. 103.

[10] see note 9 above, pp. 38-39.

[11] Durstine, J.W., "The Truck Steering System From Hand Wheel to Road Wheel", Society of Automotive Engineers SP-374, New York: The Society of Automotive Engineers, Inc.,January, 1973, p. 36.

[12] Society of Automotive Engineers, Vehicle Dynamics Terminology, Society of Automotive Engineers SAE J670e, Warrendale, PA: The Society of Automotive Engineers, Inc.

[13] see note 4 above, p. 166.

[14]Mechanical Dynamics, Inc., ADAMS User's Manual, Ann Arbor, MI, 1987.

[15] see note 14 above, pp. 4-32 - 4-35.

[16] see note 14 above, p. 4-13

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VITA

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In 1979, he graduated from the Swain School, Allentown, Pennsylvania. He enrolled at Lehigh University where he was awarded his Bachelor of Science Degree in Mechanical Engineering in 1983. Upon graduation he was employed at Mack Trucks, Incorporated of Allentown, Pennsylvania. While with Mack Trucks he started his M.S. work at Lehigh University, taking courses during the evenings. In 1990 he returned to Lehigh full-time and completed the degree requirements in that same year.