RIDE AND HANDLING ANALYSIS
OF THE AM GENERAL TRANSBUS

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1.0 INTRODUCTION

The objectives and status of AM General's ride and handling analysis are outlined below. These items are followed by a discussion of ride and handling performance measures applicable to the motor coach. In particular, the limit maneuver performance measures cited in the AM General Transbus Subcontract are reviewed with respect to their interpretation and validity as indicators of pre-crash safety quality. Following this discussion, an example set of calculations are presented to indicate the braking performance and the directional static margin that can be achieved by a motor coach possessing the running gear and suspension layout projected for the Transbus. The challenge involved in making quantitative predictions of ride quality (as a means of assessing the ride performance of the Transbus as compared with that of a passenger vehicle) is then examined.

The braking and handling computer simulation in existence at the Highway Safety Research Institute (HSRI) is described with respect to its applicability to performing the analyses required for this program. A second purpose of this description is to emphasize that a large number of design details must be known in order to obtain, or otherwise estimate, the parameter data required to predict the braking, ride, and handling performance of a motor coach. This document is concluded by summarizing the analytical work that will be conducted upon finalization of the design.

2.0 OBJECTIVE

The development of a motor coach that is responsive to the needs of present-day urban communities requires that every effort be made to insure that safety levels (as determined by braking and steering performance) are not unduly compromised by the companion goal of attracting ridership through the provision of ride comfort levels that are exhibited by the personal automobile. To this end, AM General will employ vehicle dynamic analysis tools as a means of monitoring and modifying the design process.

Given that the motor coach under design does not represent a gradual evolution from previous designs but instead represents an innovative departure from traditional design practice, it is important that analyses be employed to predict the performance that will be achieved. In certain instances, where performance standards have been promulgated by the National Highway Traffic Safety Administration for reasons of upgrading highway safety, the objective of analysis will be that of insuring that the completed coach will comply with the standards. In those cases where performance standards have yet to be developed, e.g., handling, the objective of analysis is to insure that performance will be achieved which can be judged adequate in comparison with the performance exhibited by contemporary vehicles. In either event, it is highly desirable that performance predictions be made prior to testing the prototype as a means of determining the extent to which design objectives have been achieved.

3.0 STATUS

For reasons that are clearly established in the following technical discussion, AM General cannot, at this time, make quantitative estimates of the ride and handling performance of the Transbus in terms of the specifications set down in Subcontract 9073-054. Although we are in a position to know that our design judgments and tradeoffs will produce a motor coach that will ride and maneuver in a satisfactory manner, it will be necessary to obtain (and/or determine) the mechanical properties of the selected tire before it shall be possible to predict, in quantitative terms, ride, handling, and braking performance, either in the nominal or limit performance regime. With respect to the latter performance regime, performance prediction is most difficult. Further, limit maneuver performance prediction methods have yet to be reduced to a state where it is a routine operation for a motor vehicle manufacturer to perform such calculations. The actual state of affairs is one wherein present-day motor vehicle technology does not include the means for routinely predicting and measuring a variety of limit maneuver performance measures. Accordingly, AM General approaches this task with due recognition of its advanced nature and will utilize the expertise of HSRI to predict as many nominal and limit performance measures as is reasonable within the project budget constraints that prevail.

4.0 TECHNICAL DISCUSSION

4.1 LIMIT MANEUVER MEASURES APPLIED TO THE MOTOR COACH

Measures have been developed, for application to passenger vehicles, which intend to assess the so-called limit performance of the motor car[1]. These measurement procedures are based upon the hypothesis that the performance of the vehicle near the limits of its tire-road shear force potential constitutes a first-order determinant of the vehicle's accident avoidance capability. The developed methodology examines the vehicle as an open-loop system, by which the response to specific control or disturbance inputs is viewed in terms of the performance limitation and control challenge that would likely be imposed on a driver, if he were to attempt to close the loop. As the severity of the control inputs or external disturbances are increased, the tire-vehicle system approaches a condition of seriously degraded controllability characterized by the shear force saturation of the pneumatic tire at large values of braking and/or cornering slip. As this saturation regime is encountered, the potential for loss of control rises markedly due to the obvious impotence of the steer and brake control elements.

In the case of the passenger vehicle, the areas of limit performance outlined in the subject contract have been developed and even somewhat refined. Although it is now possible to perform a battery of limit maneuvers on the passenger car, obtaining a fairly comprehensive assessment of its performance, the accurate prediction of a given vehicle's responses is not routinely achieved. The inadequacies of limit response prediction capability rest principally in the area of tire representation. Reasonable

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[1] Dugoff, H., Ervin, R.D., and Segel, L., "Vehicle Handling Test Procedures," Final Report, Contract FH-11-7297, November 1970.

predictions are utterly impossible without an accurate characterization of the entire traction field of the tire, spanning the complete range of vertical loads, longitudinal slip, and cornering slip as is experienced in limit turning and braking.

Expanding this discussion, now, to the motor coach, it is apparent that all of the limitations of the passenger car limit performance technology apply in addition to certain other factors which derive from specific design properties of the motor coach, as a breed. Although the AM General bus will exhibit a substantial improvement in roll stability, in comparison to current buses, its static c.g. height/track width relation will still put it in a class significantly removed from the passenger vehicle (Fig. 1). As a result, it is quite likely that limit turning performance on a high coefficient surface may not be a valid concept, as the vehicle likely exhibits a roll response limit rather than a yaw response limit. If a roll response limit is produced in a step-steer maneuver, the concept of a drastic steer and brake maneuver becomes moot, since this latter measure is designed to examine rollover potential in a severity regime beyond that attainable through steering inputs alone.

The <u>prediction</u> of limit response of the motor coach, moreover, becomes more difficult and more demanding of adequate tire performance data because of the same c.g. height/track width characteristic. Even in a steady-state turn, the AM General bus will have transferred its full weight to the outside tires at a lateral acceleration near 0.9g. Under dynamic conditions, especially in the proximity of bump stops, the vertical loads at the outside tires can substantially exceed the vehicle's weight, thus imposing the need for tire performance data over a tremendous vertical load range, and far exceeding rated load.

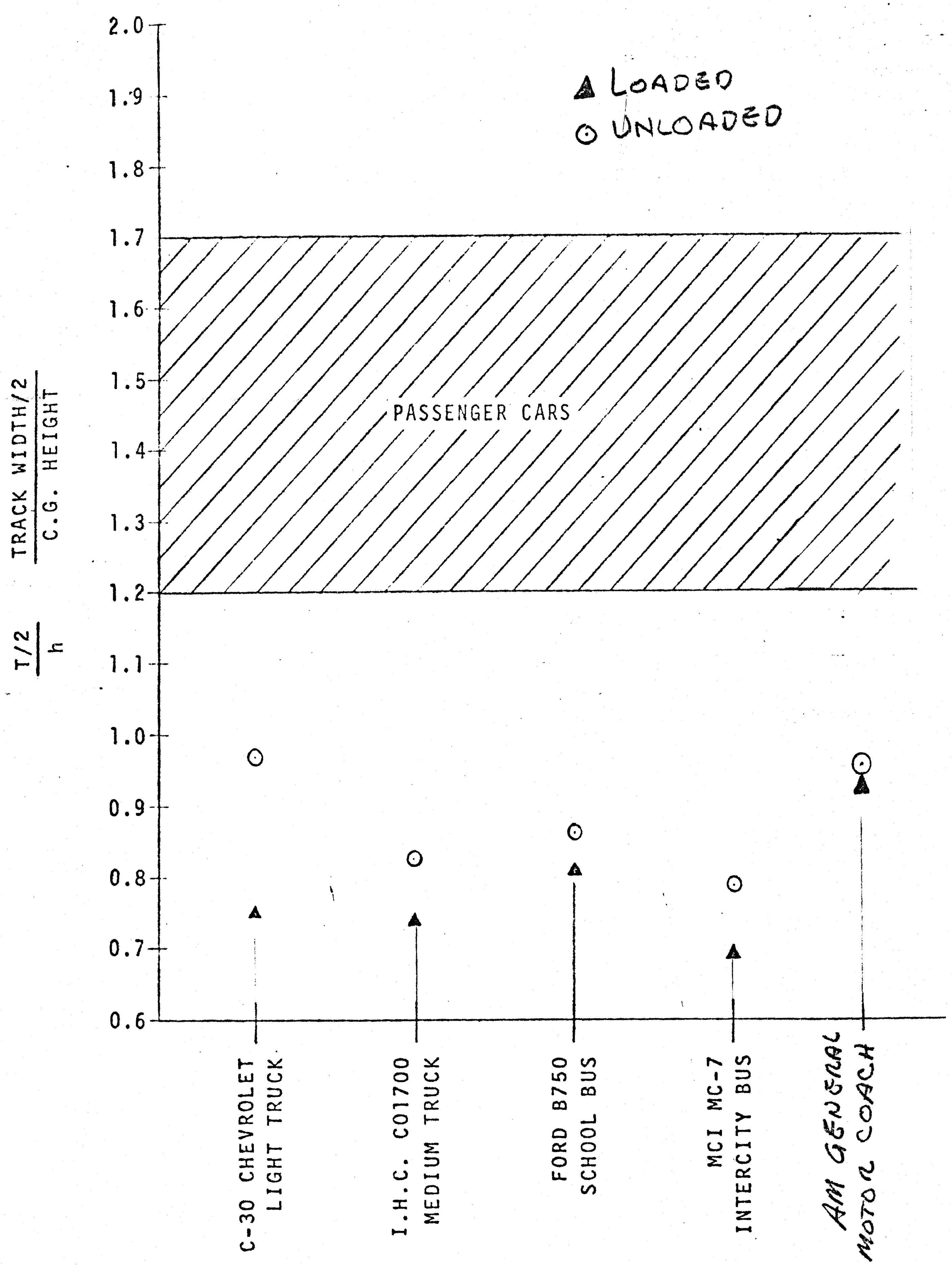

FIGURE 1. RATIOS OF HALF-TRACK WIDTH TO C.G. HEIGHTS

The suspension design factor that is of first-order importance in predicting limit turning behavior is roll stiffness distribution, the property that determines the redistribution of vertical load among inside and outside tires in a turn. As the outside tires become overloaded differentially in the front relative to the rear, a limit yawing condition (of either the spin or drift variety) will prevail. Accordingly, the prediction of limit turning performance requires an accurate determination of roll stiffness as distributed among the various axles.

In addition to the more obvious determinants of limit maneuver performance (i.e., c.g. position, tire properties, roll stiffness distribution), a host of suspension compliance and kinematic factors are significant and must be available to carry out a predictive task. These parameters are summarized in Section 5.0 of this document.

The actual conduct of limit maneuver experiments on motor coaches is without precedent in the technical literature. Thus, no direct experience can be cited that will provide insight into the applicability of passenger car limit measurement methodology to the bus. Since the predictive analysis must precede the experimental activity, however, certain assumptions will be necessary as to the likely procedure format that can be followed in limit response testing of the motor coach.

The straight-line braking experiment is sufficiently straightforward that no difficulty is anticipated in its application.

In experiments involving roadholding in a steady turn, certain practical matters may be of concern. Since the passenger vehicle test that has been developed utilizes a fixed grid of disturbances, of specified overall length, the vehicle must be placed rather accurately to cause the trajectory to properly intersect the road roughness course. Since the motor coach very likely requires more time to establish a steady turn than does a passenger vehicle, it may prove necessary to modify the existing test procedure in order to effect an accurate intersection of trajectory with the roughness course. (Note that this problem is a unique feature of open-loop experimentation, in which only the control inputs are specified.)

The step-steer test, as discussed above, will likely achieve a rollover limit on a high coefficient surface. The passenger car procedure could probably be utilized directly for the motor coach, however, as long as this likelihood is recognized and measures are taken to mechanically arrest rollover responses to prevent injury and destruction.

A sinusoidal steering procedure has been extensively applied to the passenger vehicle, but it involves a single rather than a double "lane change" as is specified in Subcontract 9073-054. This double lane-change procedure variation (involving two repetitive cycles of sinusoidal steer) has been examined for application to the passenger car under work carried out for DOT and it has been determined that the addition of a second sine wave so distorts the concept of the open-loop sine steer measure near the limit as to render it uninterpretable. Thus, it is suggested that the single sine wave version be selected with attention given to other basic changes from passenger car methodology, such as normalization of the input wave to account for gross differences in wheel base parameters and yaw transient response. It is clear that the two-second period wave used in passenger vehicle testing will be too brief for eliciting lane-change like responses from the motor coach.

Finally, the drastic steer/brake maneuver is viewed as being inappropriate for application to the motor coach if it can be shown that a significant rollover potential exists in response to steering inputs alone. It appears that "braking in a low-level turn" constitutes a reasonable substitute measure in that it provides a means for assessing the interaction between turning and braking as will occur whenever limit braking is demanded while tracking a curve.

In summary, the analysis of the limit maneuver behavior of the AM General motor coach requires the following developments:

1. A review of limit maneuver measurements and prediction as has been developed for the passenger vehicle.

2. The specification of a set of measures satisfying the (5) general performance categories as indicated in the subcontract, such that due cognizance is given to the special design properties of the motor coach which suggest a conflict with existing passenger car measurement methodology.

3. The determination of parameters characterizing the tire/bus system such that predictions of performance can be carried out.

    a) This determination must necessarily include the physical measurement of the traction field of the selected tire. It is planned that this be done, as tires become available, on HSRI's Flat Bed Tire Tester.

    b) Suspension kinematic and compliant parameters will be measured, calculated, or estimated, as necessary to complete the list given in Section 5.0.

4. The execution of the vehicle simulation activity, such that responses will be determined covering the range of performance, as will be investigated with the specified tests.

It must be recognized that, although this latter exercise can be performed, the resulting predictions may not yield a high level of agreement with experiment. Whereas excellent agreement can be expected in the linear operational range, given accurate parameter data, the prediction of the limit response of motor vehicles is far more challenging and has been met with only limited success, to date, in efforts made by HSRI and others.

4.2 BRAKING PERFORMANCE ANALYSIS

In this section, a procedure is described for predicting vehicle braking performance based upon a minimal number of vehicle, brake system, and tire-road interface parameters. Using available data on the AM General bus, an analysis was made of the braking performance of this vehicle and braking performance and braking efficiency diagrams constructed.

In the analysis, calculations were made to determine the capability of the brake system to decelerate the vehicle for a given pedal force or brake line pressure input, and the level of tire-road interface friction required to prevent wheel lock up on each axle. Braking efficiency was also calculated, which is a measure of how well the vehicle-brake system combination, on an axle-by-axle basis, takes advantage of the braking forces available in the tire-road interface.

For this preliminary analysis, HSRI was furnished brake line pressure-brake torque curves for stops from 20, 40, 50, and 60 mph, which defines the brake effectiveness characteristic. The 60 mph curves are reproduced in the braking performance diagrams shown in Figures 2 and 3. To calculate individual wheel loads, brake forces, deceleration, and friction utilization for a given brake line pressure, the following assumptions were made:

1. BRAKES. The brakes on each axle have the same line-pressure torque characteristic.

2. TIRES. The tires have an unloaded radius of 16.5", and the vertical spring constant is 4100 lb/in.

3. SUSPENSION. The spring constant measured at the wheel center at both rear axles is identical.

4. VEHICLE LOADING AND C.G. HEIGHT. These parameters are as defined in the following table:

LOADING CONDITION	STATIC LOAD, LBS.			C.G. HEIGHT INCHES	C.G. DISTANCE FROM FRONT AXLE, INCHES
	FRONT	REAR			
		LEADING	TRAILING		
Empty	6,103	7,634	7,634	44.7	202
Loaded	10,150	10,634	10,634	46.6	191.5

To calculate the dynamic load on the front axle, the following equation was used:

$$F_{z1,DYNAMIC} = [(1-\psi) + X a]W \qquad (1)$$

where

$$\psi = \frac{F_{z,\text{TOTAL REAR AXLES, STATIC}}}{W}$$

$$X = \frac{\text{CENTER OF GRAVITY HEIGHT}}{\text{WHEEL BASE}}$$

$$W = \text{Total vehicle weight}$$

$$a = \text{Vehicle deceleration in g-units} = \frac{F_{x,\text{TOTAL}}}{W}$$

$F_{x,\text{TOTAL}} = \sum_{i=1}^{3} F_{xi}$, the sum of the brake forces on each axle where F_{xi} = brake torque divided by the effective tire radius.

The dynamic load on each rear axle was calculated at each value of brake line pressure using:

$$F_{z2,\text{DYNAMIC}} = F_{z2,\text{STATIC}} - \Delta F_{z2} \tag{2}$$

$$F_{z3,\text{DYNAMIC}} = F_{z3,\text{STATIC}} - \Delta F_{z3} \tag{3}$$

where $\Delta F_{z2} = \dfrac{X\ a\ W}{1 + (\frac{b+\ell}{b-\ell})}$

b = horizontal distance from center line of rear axle to c.g. location

ℓ = half distance between rear axles

$\Delta F_{z3} = \Delta F_{z2}\ (\frac{b+\ell}{b-\ell})$

and subscripts 1, 2, and 3 refer to front, leading rear, and trailing rear axles, respectively.

The required tire-road coefficient to prevent wheel lock up on each axle was calculated at each value of brake line pressure by

$$\mu_{TIRE\text{-}ROAD,i} = \frac{F_{xi}}{F_{zi,DYNAMIC}} \qquad (4)$$

The braking efficiency is given by the following expression:

$$E_i = \frac{a}{\mu_{TIRE\text{-}ROAD,i}}$$

Results of the preliminary performance analysis are given in Figures 2, 3, and 4, for the vehicle in the empty and loaded condition. To determine the braking performance predicted at a given brake line pressure, Figures 3 and 4 are used as an example. Examination of Figure 3 shows that a pressure of 1500 psi results in a brake torque per axle of slightly less than 104,000 inch-lbs., which in turn results in a vehicle deceleration of 0.7g. To achieve this deceleration, the front axle requires a tire-road interface coefficient of 0.6, the leading rear axle a coefficient of 0.74, and the trailing rear axle a coefficient of 0.8. From the curves of Figure 4 it can be determined that, in the empty condition, the wheels on the front axle of the vehicle will lock up first for surfaces having a tire-road coefficient of 0.35 or less. For surfaces with coefficients higher than 0.35, the trailing rear axle will lock up first. In the loaded condition, the trailing rear axle tends to lock up first at coefficients of 0.1 and higher.

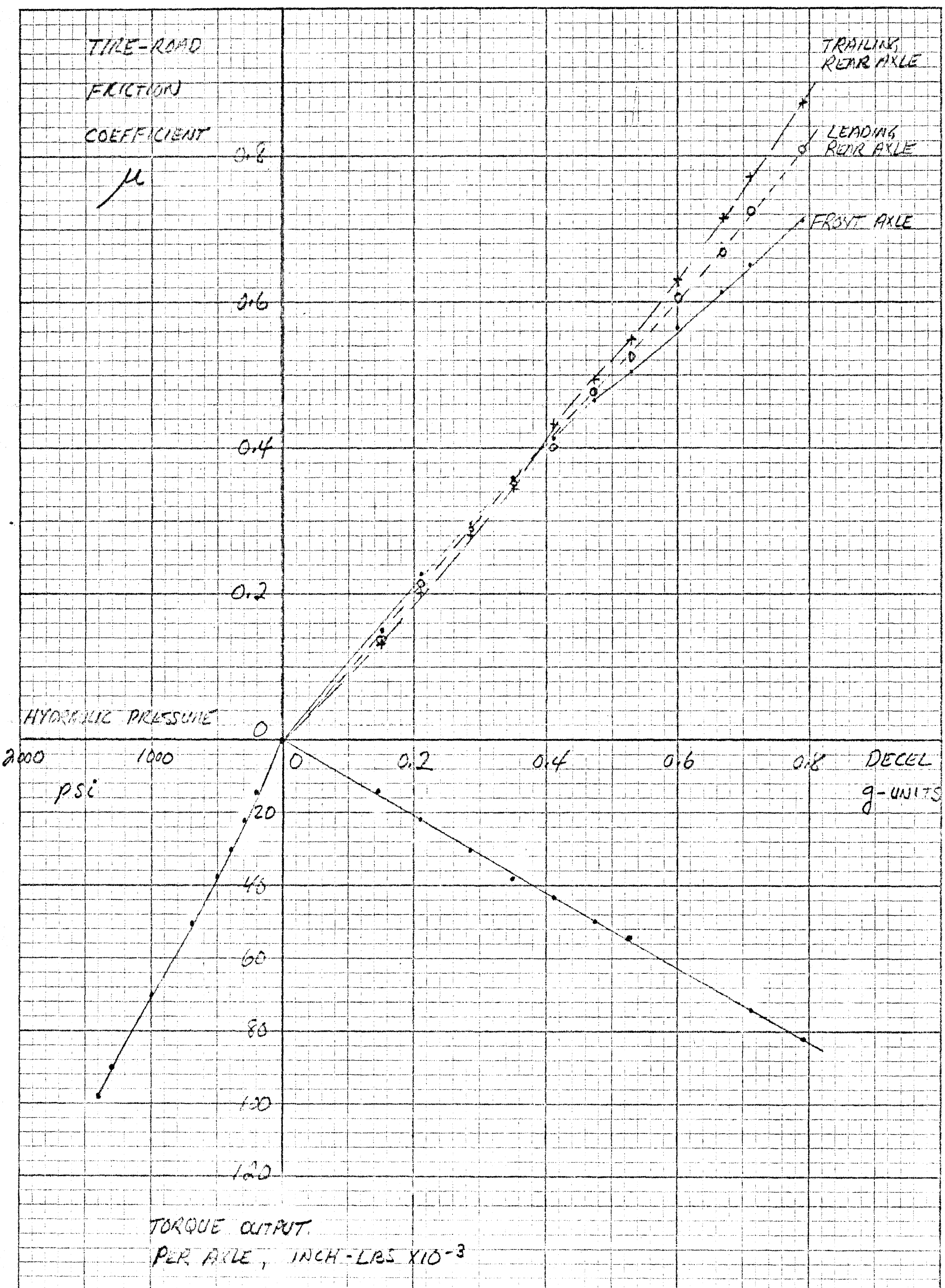

FIGURE 2. BRAKING PERFORMANCE DIAGRAM, VEHICLE EMPTY

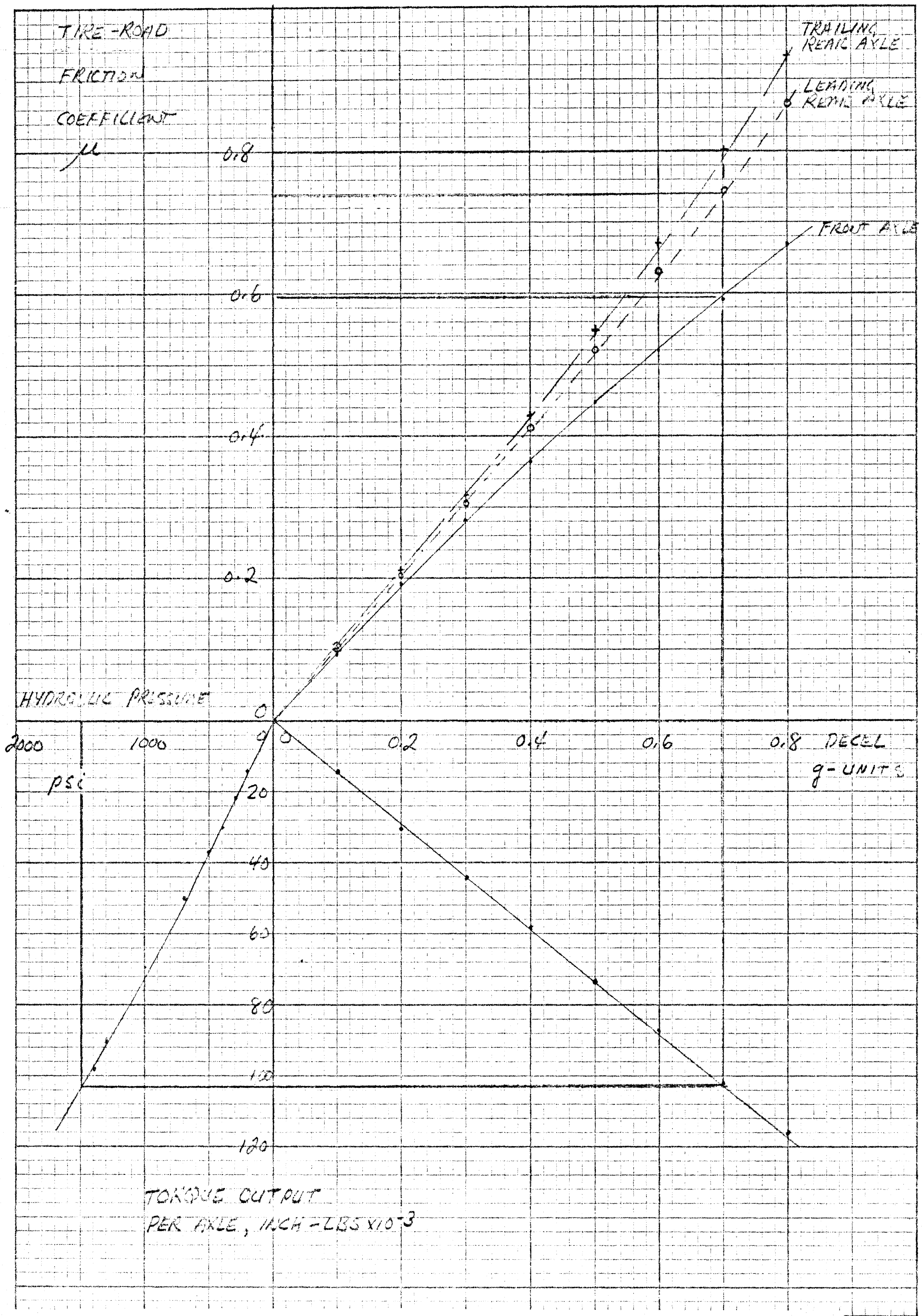

FIGURE 3. BRAKING PERFORMANCE DIAGRAMS, VEHICLE LOADED

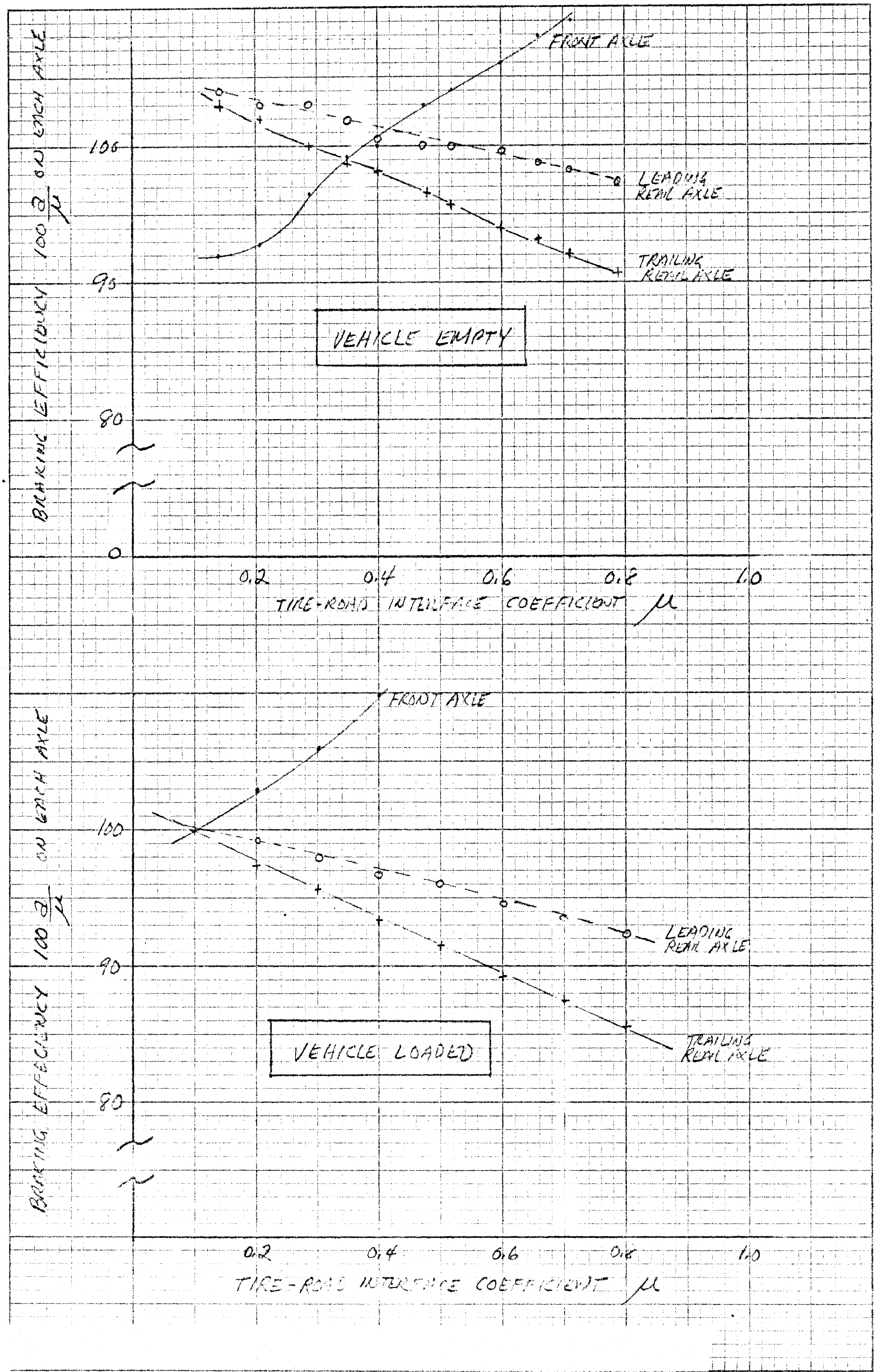

FIGURE 4. BRAKING EFFICIENCY CURVES

This preliminary analysis gives a fairly good insight into the braking performance of the vehicle as measured by braking effectiveness and braking efficiency. However, since the brake torque distribution, tire properties, and suspension properties enter into the calculations, the analysis should be repeated when the design values for these properties are finalized.

4.3 EQUILIBRIUM TURNING ANALYSIS

A preliminary analysis of the steady turning performance of the projected Transbus is presented in this section. Techniques, which have been used to examine passenger car directional performance[1,2,3], are modified and extended to apply to the 3-axle bus. First, the non-rolling vehicle is treated to determine directional performance measures for the fully loaded, partially loaded, and empty bus at low lateral acceleration conditions which are typical of normal driving. Then, the influence of vehicle roll upon those suspension characteristics which are related to steady turning performance are discussed.

4.3.1 SUMMARY OF PRELIMINARY ANALYSIS. Figure 5 shows a simplified 3-axle vehicle model which is suitable for analyzing the first order influences of center of gravity location and tire lateral forces. The two wheels on each axle are treated as an equivalent single wheel.

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[1]Segel, L., "Theoretical Prediction and Experimental Substantiation of the Response of the Automobile to Steering Control," Proceedings of Institution of Mechanical Engineers (Automobile Division), 1956-1957.

[2]Pacejka, H., "Simplified Analysis of Steady-State Turning Behavior of Motor Vehicles," Automotive Design Engineering, London, (to be published, Summer 1972).

[3]HSRI Staff, "Motor Vehicle Performance - Measurement and Prediction," Univ. of Mich. Summer Conference Course, 1972, Lecture IVB.

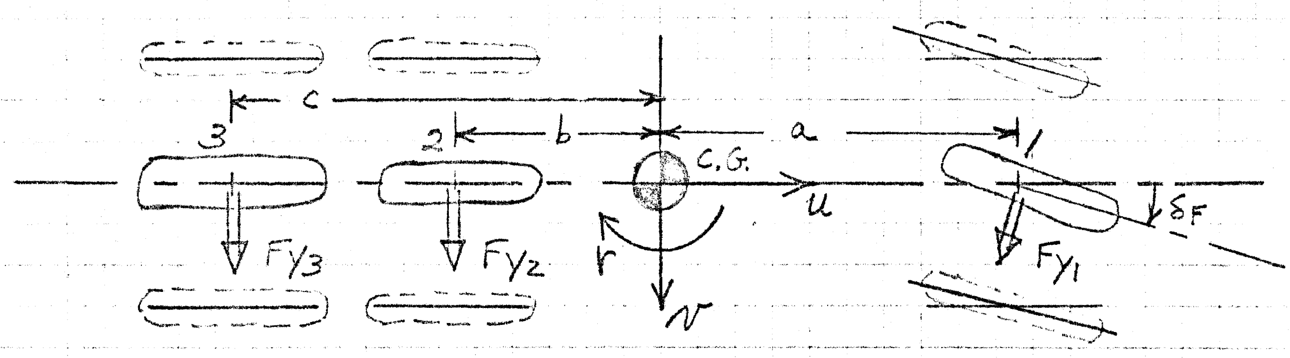

FIGURE 5. SIMPLIFIED BUS MODEL

The vehicle motion in the horizontal ground plane is described by a forward velocity, u, a lateral velocity, v, and a yaw rate, r. For small front wheel steer angle, δ_F, the side force component of F_{y1} is approximately equal to F_{y1} (i.e., $F_{y1} \cos \delta_F \doteq F_{y1}$). If it is assumed that adequate drive thrust is provided to keep the forward velocity constant (or changing very slowly), then the equations of motion become

$$m(\dot{v} + ur) = F_{y1} + F_{y2} + F_{y3} \tag{5}$$

$$I_z \dot{r} = a F_{y1} - b F_{y2} - c F_{y3} \tag{6}$$

where m is the mass of the vehicle and I_z is the yaw moment of inertia.

For a steady turn at small steering angle, $\dot{v}=\dot{r}=0$, $u \doteq V$ and $rR=V$ where V is the velocity of vehicle and R is the path radius.

For a steady turn, Equations (5) and (6) become

$$\frac{V^2}{R} = F_{y1} + F_{y2} + F_{y3} \tag{7}$$

$$0 = a\,F_{y1} - b\,F_{y2} - c\,F_{y3} \tag{8}$$

On assuming a turn restricted to small values of centripetal acceleration, the slip angles at each tire may be expressed as

$$\alpha_1 \doteq \frac{v}{u} + \frac{ar}{u} - \delta_F \doteq \beta + \frac{a}{R} - \delta_F \tag{9}$$

$$\alpha_2 \doteq \frac{v}{u} - \frac{br}{u} = \beta - \frac{b}{R} \tag{10}$$

$$\alpha_3 \doteq \frac{v}{u} - \frac{cr}{u} = \beta - \frac{c}{R} \tag{11}$$

where $\beta = v/u$, the sideslip angle of the vehicle. (The modifications in these angles caused by the influence of roll on suspension properties will be discussed later.) For slip angles assumed to be small, the tire side forces are given by

$$F_{y1} = -C_{\alpha 1}\alpha_1\,,\quad F_{y2} = -C_{\alpha 2}\alpha_2,\quad F_{y3} = -C_{\alpha 3}\alpha_3 \tag{12}$$

where $C_{\alpha 1}$, $C_{\alpha 2}$, and $C_{\alpha 3}$ are the cornering stiffnesses of both tires on the axles labeled one, two, and three, respectively.

Solving (7) through (12) for the front-wheel steer angle required to follow a given radius path at a given velocity yields:

$$\delta_F = \left(\frac{1}{R}\right)\left[\ell_b - \frac{m}{(a-s)}\,\frac{s}{C_{\alpha 1}}\,V^2\right] \tag{13}$$

where ℓ_b is "an equivalent wheel base for a 3-axle vehicle" and s is the distance from the center of gravity to the neutral steer point. Equations (7) through (12) yield the following expressions for s and ℓ_b:

$$s = \frac{a\, C_{\alpha 1} - b\, C_{\alpha 2} - c\, C_{\alpha 3}}{C_{\alpha 1} + C_{\alpha 2} + C_{\alpha 3}} \tag{14}$$

$$\left(\begin{array}{l} s > 0 \text{ implies oversteer} \\ s < 0 \text{ implies understeer} \end{array} \right)$$

$$\ell_b = \left[\frac{(a+b) + \frac{C_{\alpha 3}}{C_{\alpha 2}} \frac{(a+c)^2}{(a+b)} + \frac{C_{\alpha 3}}{C_{\alpha 1}} \frac{(c-b)^2}{(a+b)}}{1 + \frac{C_{\alpha 3}}{C_{\alpha 2}} \frac{(a+c)}{(a+b)}} \right] \tag{15}$$

Note that the static margin, as commonly defined, is equal to $-s/\ell_b$.

Equation (13) can be used to relate path curvature, 1/R, and centripetal acceleration, $\frac{V^2}{Rg}$, (in g units) to front wheel angle, δ_F, viz.:

$$\ell_b \left(\frac{1}{R}\right) - \left(\frac{mg\ s}{(a-s)C_{\alpha 1}}\right) \left(\frac{V^2}{Rg}\right) = \delta_F \tag{16}$$

Note that Equation (16) can be used to predict steady turning performance of a motor vehicle in constant radius, constant velocity, or constant steer angle tests.

For this analysis it is convenient to define an understeer/oversteer coefficient, uoc, as:

$$uoc = \left[\frac{-\ (W\ lbs)}{(C_{\alpha 1}\ \frac{lbs}{rad})} \quad (\frac{s}{a-s}) \right] \quad rad/g \tag{17}$$

where W is the weight of the vehicle. In general, uoc is equal to the rate of change of steer angle with centripetal acceleration. On a constant radius path an increase in velocity is accompanied by an increase in steer angle for an understeer vehicle and an increase in velocity is accompanied by a decrease in steer angle for an oversteer vehicle.

The solution of Equations (7) through (12) for vehicle sideslip angle β yields

$$\beta = \left[\frac{C_\alpha \ C_{\alpha 1}(q^2 - as) - C_{\alpha 1}\ a\ mV^2}{C_\alpha{}^2 (q^2 - s^2) - C_\alpha\ s\ mV^2} \right] \delta_F \tag{18}$$

where

$$C_\alpha = C_{\alpha 1} + C_{\alpha 2} + C_{\alpha 3} \tag{19}$$

$$q^2 = \left[\frac{a^2\ C_{\alpha 1} + b^2\ C_{\alpha 2} + c^2\ C_{\alpha 3}}{C_\alpha} \right] \tag{20}$$

(It may be noted that the so-called "damping-in-yaw" or "yaw stiffness" is given by $\frac{C_\alpha q^2}{V}$.)

In order to predict the steady turning performance of the Transbus, tire cornering stiffness data and their dependence on load and inflation pressure are required. Since the tires to be used on this bus do not exist as yet, the numerical results presented below are based on the tire properties summarized in Figure 6. (It should be noted that these data were obtained from tests performed on truck tires.)

EXAMPLE CALCULATIONS FOR A NON-ROLLING VEHICLE:

Case 1: An almost fully loaded bus - 10,000 lbs per axle.

The c.g. location is such that a = 188", b = 72", and c = 116". Figure 6 shows that

$$C_{\alpha 1} = C_{\alpha 2} = C_{\alpha 3} = 2(485) \text{ lbs/deg} = 970 \text{ lbs/deg} \equiv 55{,}600 \text{ lbs/rad}$$

The above equations yield:

$$\ell_b = 287'' \equiv 23.9'$$

$$s = 0$$

$$uoc = 0$$

Thus, the fully loaded bus is neutral steer (neglecting roll influences on suspension properties which can be designed to make the vehicle understeer).

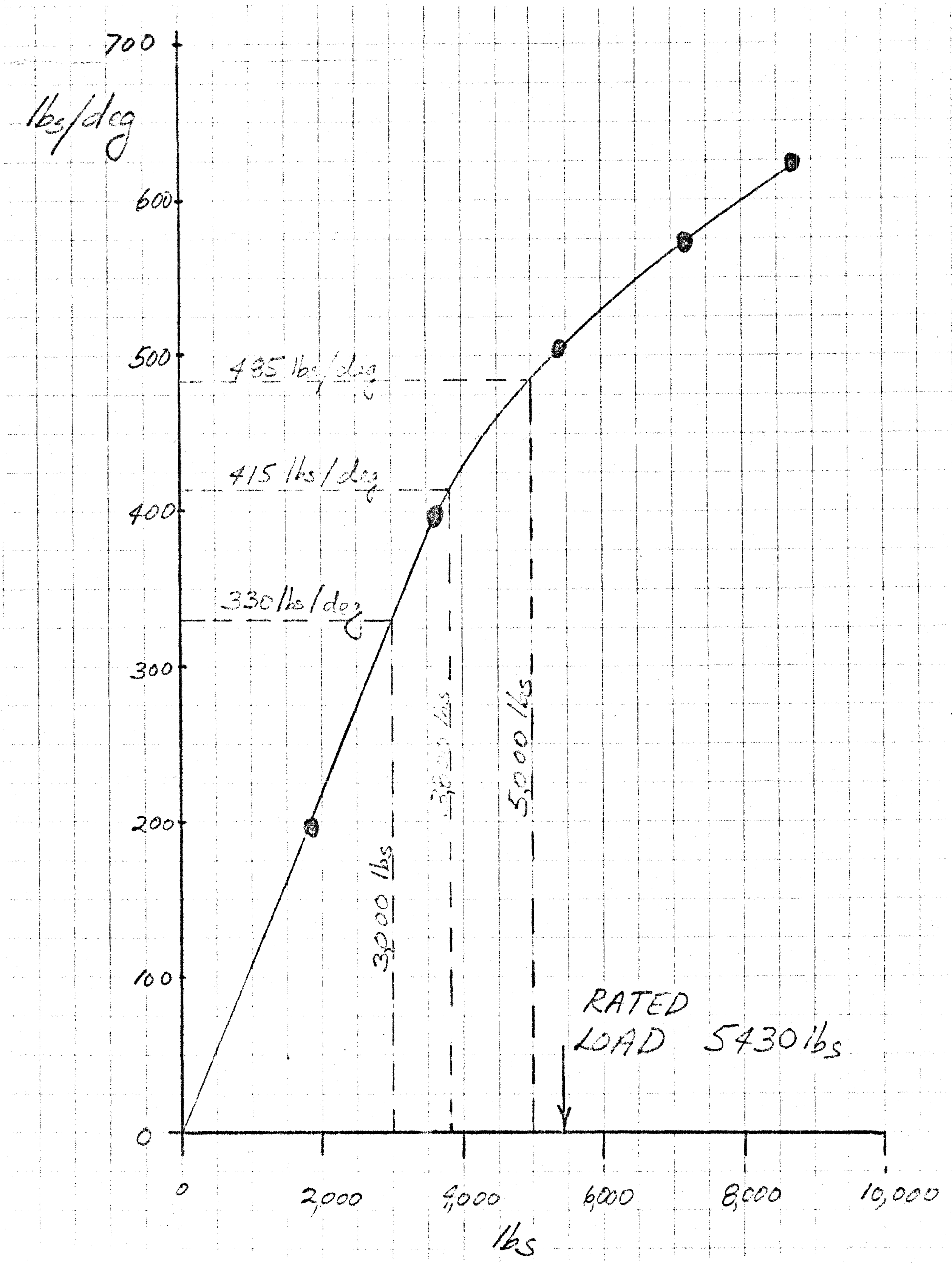

FIGURE 6. CORNERING STIFFNESS lbs/deg
vs.
LOAD lbs.

11 x 22.5 Truck Tire Load Range F
85 psi

Case 2: Empty bus

Front axle:	6000 lbs load
Leading rear axle:	7600 lbs load
Trailing rear axle:	7600 lbs load

The c.g. location is such that a = 202", b = 58", c = 102". Figure 6 shows that

$$C_{\alpha 1} = 2(330) \text{ lbs/deg} = 660 \text{ lbs/deg} \equiv 37{,}800 \text{ lb/rad}$$

$$C_{\alpha 2} = C_{\alpha 3} = 2(415 \text{ lbs/deg}) = 830 \text{ lbs/deg} \equiv 47{,}600 \text{ lbs/rad}$$

The above equations predict that

$s = 0.22"$ (very slightly oversteer)

$uoc = -\ 0.0009$

$\ell_b = 288" \equiv 24'$

$-\ s/\ell = -\ 0.0008$

Thus, the empty bus is nearly neutral steer. Note that the tire cornering stiffness versus vertical load characteristic tends to compensate for changes in c.g. location.

Case 3: Extremely poor distribution of load

Front axle:	6,000 lbs load
Both rear axles:	10,000 lbs load

The c.g. location is such that a = 215", b = 45", c = 89". Figure 6 shows that:

$$C_{\alpha 1} = 660 \text{ lbs/deg} \equiv 37{,}800 \text{ lb/rad}$$

$$C_{\alpha 2} = C_{\alpha 3} = 970 \text{ lbs/deg} \equiv 55{,}600 \text{ lb/rad}$$

The above equations predict that

$$s = 4.6\text{"} \quad \text{(oversteer)}$$

$$uoc = -\ 0.015 \text{ rad/g}$$

$$\ell_b = 289\text{"} \equiv 24.1'$$

$$-\ s/\ell = -\ 0.0159$$

Thus, with a large load in the rear of the bus, and empty load on the front axle, the vehicle is moderately oversteer.

SUMMARY OF SAMPLE CALCULATIONS
FOR THE NON-ROLLING BUS

	Fully Loaded	Empty	Extremely Poor Loading
s (inches)	0	0.22	4.6
uoc (rad/g)	0	-0.0009	-0.015
ℓ_b (ft)	23.9	24.0	24.1
$-s/\ell_b$	0	-0.0008	-0.0159
$C_\alpha q^2$ $(\frac{ft^2 lb}{rad})$	$(2.09)10^7$	$(1.53)10^7$	$(1.60)10^7$
δ_F in degrees for a 200' radius turn at 30 mph $(\frac{V^2}{Rg} = .3)$	6.85	6.88	6.65

4.3.2 INFLUENCE OF VEHICLE ROLL. The primary influences of vehicle roll in a low lateral acceleration steady turn are (1) to change the steer angle of the front wheels due to interaction between the steering and suspension system linkages, (2) to change the camber angles of the front wheels with respect to the road, and (3) to steer the rear axles. On including these effects into Equations (9), (10), and (11), the tire slip angle expressions can be written as

$$\alpha_1 = \frac{v}{u} + \frac{ar}{u} - \delta_{FS} - \frac{\partial \delta_F}{\partial \phi} \phi - \frac{C_{\gamma 1}}{C_{\alpha 1}} \frac{\partial \gamma}{\partial \phi} \phi \qquad (21)$$

$$\alpha_2 = \frac{v}{u} - \frac{br}{u} - \frac{\partial \delta_2}{\partial \phi} \phi \tag{22}$$

$$\alpha_3 = \frac{v}{u} - \frac{cr}{u} - \frac{\partial \delta_3}{\partial \phi} \phi \tag{23}$$

where ϕ is the roll angle

$$\delta_F = \delta_{FS} + \left(\frac{\partial \delta_F}{\partial \phi} + \frac{C_{\gamma 1}}{C_{\alpha 1}} \frac{\partial \gamma}{\partial \phi}\right) \phi \tag{24}$$

$$\delta_{FS} = \frac{\delta_{sw}}{NG}$$

δ_{sw} = steering wheel angle

NG = steering ratio

$C_{\gamma 1}$ = camber stiffness of the front tires

γ = the camber angle of the front wheels

δ_2 = roll steer of the second axle

δ_3 = roll steer of the third axle.

In the present design of the AM General Transbus, the trailing arms used in the rear suspensions are nearly parallel to the ground and to the longitudinal axis of the vehicle. Thus on a single rear axle as one wheel moves up in jounce and the other moves down in rebound they both move forward an equal amount and, to first order, the roll steer of the rear axle is zero.

The camber force from a rolling tire acts in the direction in which the top of the tire is cambered. For example, in an independent front suspension with equal length horizontal and parallel control arms, the camber angle with respect to the ground is equal to the roll angle of the vehicle. Since the roll angle is away from the direction of turn, the influence of the camber force is to reduce the net side force at the front axle, which reduction is an understeering effect. (Note that the camber force, $C_{\gamma 1}\gamma$, has been included in the expression that defines the slip angle of the front tires.)

For the current design, we shall assume that $\frac{\partial \delta_2}{\partial \phi} \doteq 0$ and $\frac{\partial \delta_3}{\partial \phi} \doteq 0$. Equation (16) still applies to the rolling vehicle, with the exception that

$$\delta_F = \delta_{FS} + K_F \phi$$

where $$K_F = \left(\frac{\partial \delta_F}{\partial \phi} + \frac{C_{\gamma 1}}{C_{\alpha 1}} \frac{\partial \gamma}{\partial \phi}\right)$$

In a steady turn, roll equilibrium is expressed by

$$- F_y h - K_\phi \phi = 0$$

where $F_y = F_{y1} + F_{y2} + F_{y3} = \frac{V^2}{R} m$

h = c.g. height

K_ϕ = total roll stiffness

Thus

$$\phi = \frac{F_y h}{K_\phi} = - \frac{V^2}{gR} \left(\frac{mg\ h}{K_\phi}\right) \tag{25}$$

and Equation (16) may be written as

$$\ell_b \left(\frac{1}{R}\right) - \left(\frac{mg\ s}{(a-s)C_{\alpha 1}}\right) \left(\frac{V^2}{Rg}\right) = \delta_{FS} - \frac{V^2}{gR} \frac{mgh\ K_F}{K_\phi}$$

or

$$\ell_b \left(\frac{1}{R}\right) + W\left[\frac{h\ K_F}{K_\phi} - \frac{s}{(a-s)C_{\alpha 1}}\right]\left(\frac{V^2}{Rg}\right) = \delta_{FS} = \frac{\delta_{SW}}{NG} \tag{26}$$

To illustrate the influence of roll on the static margin of the Transbus which has an independent front suspension, we shall assume that there is no roll steer designed into the front suspension (i.e., $\frac{\partial \delta_F}{\partial \phi} \doteq 0$) and that

$$\frac{C_{\gamma 1}}{C_{\alpha 1}} \doteq \frac{1}{8} \text{ (a typical value for truck tires)}$$

We shall also assume that $\frac{\partial \gamma}{\partial \phi} \doteq 1.0$, whereon it follows that $K_F = 1/8$.

For the fully loaded bus, $h \doteq 47''$. Since the bus, as presently configured, does not have any roll stabilizer bars*, the roll stiffness is determined by the spring rates at the wheels, K_{s1}, K_{s2}, and K_{s3} and the track, T:

*It appears that bump stop contact will take place at about 0.5g lateral acceleration, if roll stabilizers are not employed.

$$K_\phi = \frac{T^2}{2}(K_{s1} + K_{s2} + K_{s3})$$

$$K_\phi = \frac{(87)^2}{2}(650 + 600 + 600)\ \frac{\text{in.lb.}}{\text{rad.}}$$

$$K_\phi = 7 \times 10^6\ \frac{\text{in.lb.}}{\text{rad.}}$$

For the fully loaded bus,

$$\frac{W\ h\ K_F}{K_\phi} = \frac{30{,}000\ (47)(1/8)}{7 \times 10^6} = .025,$$

and the basic steady turn relationship becomes

$$(23.9)\ (\frac{1}{R}) + (.025)\ (\frac{V^2}{Rg}) = \delta_{FS}$$

We find that uoc = 0.025, an understeer result. Further, we compute that the front wheel angle required for 0.3g turn on a 200' radius path is 0.127 rad (7.27°). If the front wheel angle is held fixed at this value and drive thrust is reduced so as to decrease speed, we find that, under static turning conditions, the radius of the path will decrease linearly from 200 ft. at 30 mph to 188 ft. as the velocity approaches zero.

In this discussion of equilibrium turning, two other phenomena should be mentioned. First, compliance in the steering system can reduce the front wheel steer angles. The tire aligning torque and the moment of the tire lateral force about the steering pivot (as influenced by caster) are important in this case. On including steering system compliance, the equation for α_1 becomes

$$\alpha_1 = \frac{v}{u} + \frac{ar}{u} - \delta_F$$

where $\delta_F = \delta_{FS} + K_F \phi - ((AT) + F_{y1} x)/K_{SS}$

AT = the aligning torque for both front tires

x = the mechanical trail due to caster, and

K_{SS} = the steering system stiffness.

It should be noted that steering system compliance acts to make the vehicle more understeer.

Second, at moderate levels of lateral acceleration (above 0.3g on a dry surface), the load transfer from the inside to the outside tire causes a net loss in the side force produced by both wheels on a single axle. On passenger cars, it is common practice to make the front roll stiffness greater than the rear roll stiffness and thus obtain a greater loss in side force at the front wheels than at the rear wheels.

For the 3-axle bus as currently configured, the extent to which the understeer will be modified at higher levels of turning acceleration as a result of load transfer effects are not known. Many factors come into play at higher lateral accelerations and computer techniques are needed to obtain precise results. Nevertheless, these preliminary calculations show that a 3-axle Transbus can be designed to remain understeer at all loading conditions by proper design of the suspension and steering system.

4.4 RIDE ANALYSIS

There are many levels of completeness and rigor applicable to the prediction of vehicle response to an uneven road profile.

From a design point of view, however, the task of minimizing the ride response of the sprung mass to roadway excitation is reasonably straightforward. The softer the ride rate of the suspension, the smaller the accelerations experienced by the suspended mass, at the expense of increased pitch and roll displacements being caused by braking and turning accelerations. Because there are practical constraints on wheel travel and in view of the general design objective of limiting roll displacements so as to not produce bump-stop contact during hard turning, the designer finds that, for passenger vehicles, he is generally restricted to a static deflection that produces a "ride frequency" in the neighborhood of one cycle per second. In the case of vehicles that have a large payload relative to their empty weight, the designer is usually forced to accept a smaller static deflection (and, consequently, a higher ride frequency) in order to provide for adequate wheel travel in both the loaded and empty conditions. This basic compromise is mitigated on the Transbus by using a suspension design that provides a low ride rate while maintaining a constant design height through varying the spring preload.

To the degree that the designer is able to match the ride frequency of a motor coach with that of a passenger vehicle, he is assured that, <u>to first order</u>, he is attaining equivalent levels of vibration isolation. However, it does not automatically follow that the ride quality of the coach will be equal to that exhibited by the passenger car. This statement is particularly true if the ride quality of the passenger car is significantly influenced by the vibration isolation provided by its compliant seats. In addition, there are a host of other design variables (such as the radial stiffness of the tires, the undamped natural frequency of the wheel masses supported on the tire springs, the ratio of unsprung to sprung mass, the shock absorber damping level, the wheelbase, and the ratio of the [radius of gyration in pitch]^2 to the product of the distances from the c.g. to the forward and rearward wheel centers that influence the response of a two-axled motor vehicle to road roughness. The Transbus is not only different from a passenger vehicle by virtue of having three axle locations at which disturbances are imposed on the sprung chassis, it also possesses mechanical and inertial properties in its running gear and chassis that are not necessarily scaled proportionally to that of the passenger vehicle even though both vehicles may exhibit equal levels of ride frequency as defined by the static spring deflection at each wheel location.

Consequently, in order to compare the ride quality achieved by two different vehicles, as measured by the vertical and fore and aft accelerations seen by a seated passenger, one must resort to a dynamic simulation, or to direct comparisons of experimental data obtained over the same roadway at comparable speeds. For a nonexistent vehicle, simulation is the only alternative. However, once the geometry and mechanical properties of tires and suspension are established, it is straightforward to conduct a ride study by assuming the vehicle to traverse a symmetrical road profile causing the sprung mass to respond only in pitch and bounce. The task remains to select a representative disturbance for use in a simulation study and it appears reasonable to assume a random profile whose spectral density is typical of that possessed by urban streets. Further, it appears pertinent to examine the ride response caused by traversal of a profile that typically exists at the intersection of two highly crowned urban streets. In this instance, it is primarily the different ratio of wheelbase to length of road wave that exists for a coach and passenger vehicle which would be expected to cause a differing ride response. To the degree that this latter profile may represent the greatest likelihood for causing the suspension to strike through, it may be advisable to assume such a profile to provide a meaningful check of the ability of the suspension to accomplish its isolation task. The vehicle model and simulation to be described below can, and will, be modified to accomplish this task.

5.0 HSRI BRAKING, RIDE, AND HANDLING SIMULATIONS

The simulation programs currently available at HSRI which may be used to predict the braking, ride, and handling performance of the AM General bus are described in this section. Also given are the vehicle and vehicle-system parameters necessary to utilize the simulation programs.

5.1 AVAILABLE PROGRAMS

Two separate programs are available for use in predicting the performance of the Transbus--a Braking (and ride) Performance Program and a Braking and Handling Performance Program. In each program, the user may specify the vehicle geometry, brakes, suspension, tire and tire-road interface characteristics, weight, and payload distribution. Since both programs include wheel rotational degrees of freedom, and means of simulating delays and lags in brake system response, simulation of antiskid devices may be added by the user. Dynamic models of several common types of tandem suspensions are available.

5.1.1 THE UNIT VEHICLE BRAKING PERFORMANCE PROGRAM. This dynamic simulation program is based upon a mathematical model that represents a 3-axle unit vehicle. Motions are constrained to the plane of symmetry (vertical plane). Specifically, the wheels can bounce and spin, the chassis can heave and pitch, and the vehicle can accelerate (decelerate) in straight-line motion. The braking system is modeled in a manner such that the brake torque-line pressure characteristic can be specified for each brake and variable time lags and delays in torque response can be introduced. Thus, any desired brake force distribution can be specified.

The model has nine degrees of freedom, which are listed in Table 1.

TABLE 1

DEGREES OF FREEDOM—BRAKING PERFORMANCE MODEL

Variable	Description
X	vehicle forward displacement
Z	vertical displacement of c.g.
ZS1	vertical displacement of front axle
ZS2	vertical displacement of leading rear axle
ZS3	vertical displacement of trailing rear axle
Ω1	angular velocity of front wheels
Ω2	angular velocity of wheels on leading rear axle
Ω3	angular velocity of wheels on trailing rear axle

To determine the value of these variables as functions of time, nine differential equations of motion are solved simultaneously, along with ancillary equations defining intermediate variables such as suspension deflections, tire-road interface forces, normal forces on the tires, and horizontal forces acting on the sprung masses. The subroutine used to accomplish the major portion of the integration of the equations of motion is based upon Hamming's predictor-corrector method. Some optional features are listed below:

1. The user may choose brake type and lining-drum descriptive parameters rather than input dynamometer pressure-torque curves.

2. Shock absorber characteristics are generally characterized by C, the slope of the force-velocity curve. To make the model more accurate, a two-slope shock absorber can be specified, characterized by jounce slope, CJ, and rebound slope, CR.

3. The spring force-deflection relation may be characterized by the slope K of the force-deflection curve, or by table lookup of deflection-force points.

4. Rough road coordinate points or a user-supplied road algorithm may be conveniently entered as input.

5.1.2 THE BRAKING AND HANDLING PERFORMANCE MODEL. This dynamic simulation contains eighteen degrees of freedom, which are listed in Table 2. (Note X, Y and Z are fixed axes; x, y and z are body axes.)

TABLE 2

DEGREES OF FREEDOM, UNIT VEHICLE
BRAKING AND HANDLING MODEL

Variable	Description
X	longitudinal position of sprung mass center
Y	lateral position of sprung mass center
Z	vertical position of the sprung mass center
p	sprung mass rotation rate about x axis
q	sprung mass rotation rate about y axis
r	sprung mass rotation rate about z axis
$ZS_{i,j}$ (i=1,3;j=1,2)	vertical position of wheel i on axle j
$\Omega_{i,j}$ (i=1,3;j=1,2)	rotation rate of wheel i on axle j

This model has all the features of the pitch plane unit vehicle model including:

1. Tandem axles may be specified
2. Optional table lookup for force-deflection at each suspension
3. Two-slope shock absorber
4. Brake characteristics may be specified by dynamometer curves or by specification of brake parameters
5. Optional rough road input.

5.2 REQUIRED PARAMETER DATA

A copy of the printout of the input parameters for a typical run on the Unit Vehicle Braking and Handling program is given in the following pages. Steering inputs may be specified by a table of up to 25 steering angle-time pairs. Braking inputs are specified by a similar table for the pressure output of the treadle valve as a function of time. The torque-line pressure characteristic may be specified for the brakes on each wheel.

INPUT PARAMETER TABLE

SYMBOL	DESCRIPTION	INITIAL VALUE
KEY	AXLE KEY: SET TO 0 FOR SINGLE AXLE 1 FOR WALKING BEAM 2 FOR ELLIPTIC LEAF	1
AA1	HORIZONTAL DIST. FROM WALKING BEAM PIN TO FRONT AXLE (IN)	24.00
AA2	HORIZONTAL DIST. FROM WALKING BEAM PIN TO REAR AXLE (IN)	26.00
AA4	VERTICAL DIST. FROM AXLE TO W.B. (IN)	8.00
AA5	VERTICAL DIST. FROM AXLE TO TORQUE ROD (IN)	18.00
A1	HORIZONTAL DISTANCE FROM CG TO MIDPOINT OF FRONT SUSPENSION (IN)	49.50
A2	HORIZONTAL DISTANCE FROM CG TO MIDPOINT OF REAR SUSPENSION (IN)	140.50
ALPHA1	STATIC DISTANCE, FRONT AXLE TO GROUND IN)	19.95
ALPHA2	STATIC DISTANCE, REAR AXLE(S) TO GROUND (IN)	20.00
C1	VISCOUS DAMPING: JOUNCE ON FRONT AXLE (LB-SEC/IN.)	8.33
C2	VISCOUS DAMPING: REBOUND ON FRONT AXLE (LB-SEC/IN.)	16.67
C3	VISCOUS DAMPING: JOUNCE ON FRONT TANDEM AXLE (LB-SEC/IN.)	8.33
C4	VISCOUS DAMPING: REBOUND ON FRONT TANDEM AXLE (LB-SEC/IN.)	16.67
CALF1	LATERAL STIFFNESS, FRONT TIRES (LBS/DEG)	350.000
CALF2	LATERAL STIFFNESS, FRONT TANDEM TIRES (LBS/DEG)	526.00
CALF3	LATERAL STIFFNESS, REAR TANDEM TIRES (LBS/DEG)	526.000
CF1	MAX. COULOMB FRICTION, FRONT SUSPENSION (LB)	1100.00
CF2	MAX. COULOMB FRICTION, REAR SUSPENSION (LB)	2200.00
CS1	LONGITUDINAL STIFFNESS, FRONT TIRES (LBS)	44000.00
CS2	LONGITUDINAL STIFFNESS, FRONT TANDEM TIRES (LBS)	87120.00
CS3	LONGITUDINAL STIFFNESS,REAR TANDEM TIRES (LBS)	87120.00
DELTA1	STATIC VERTICAL DISTANCE, FRONT AXLE TO TRACTOR CG (IN)	22.00
FA1	FRICTION REDUCTION PARAMETER ON FRONT TIRES	0.005
FA2	FRICTION REDUCTION PARAMETER ON FRONT TANDEM TIRES	0.005
FA3	FRICTION REDUCTION PARAMETER ON REAR TANDEM TIRES	0.005

J1	SPRUNG MASS ROLL MOMENT OF INERTIA (IN-LB-SEC**2)	27960.000
J2	SPRUNG MASS PITCH MOMENT OF INERTIA (IN-LB-SEC**2)	103492.000
J3	YAW MOMENT OF INERTIA (IN-LB-SEC**2)	378000.000
JA1	ROLL MOMENT OF FRONT AXLE (IN-LB-SEC**2)	4560.000
JA2	ROLL MOMENT OF FRONT TANDEM AXLE (IN-LB-SEC**2)	4080.000
JS1	POLAR MOMENT OF FRONT WHEELS (IN-LB-SEC**2)	326.00
JS2	POLAR MOMENT OF FRONT TANDEM WHEELS (IN-LB-SEC**2)	205.00
JS3	POLAR MOMENT OF REAR TANDEM WHEELS (IN-LB-SEC**2)	205.00
K1	SPRING RATE, FRONT SUSPENSION (LB/IN)	5600.00
K2	SPRING RATE, FRONT TANDEM AXLE (LB/IN	30000.000
KT1	SPRING RATE, FRONT TIRES (LB/IN)	9400.00
KT2	SPRING RATE, FRONT TANDEM TIRES (LB/IN)	18800.00
KT3	SPRING RATE, REAR TANDEM TIRES (LB/IN)	18800.00
MUZERO1	COEFFICIENT OF FRICTION, FRONT WHEELS	1.00
MUZERO2	COEFFICIENT OF FRICTION, FRONT TANDEM WHEELS	1.00
MUZERO3	COEFFICIENT OF FRICTION, REAR TANDEM WHEELS	1.00
PERCNT	PERCENT EFFECTIVENESS OF TORQUE RODS	95.00
PW	WEIGHT OF PAYLOAD (LBS)	7300.00
PJ	POLAR MOMENT OF PAYLOAD (IN-LB-SEC**2)	112051.00
PX	HORIZONTAL DISTANCE FROM MIDPOINT OF REAR SUSPENSION TO MASS CENTER (IN)	22.00
PZ	VERTICAL DISTANCE FROM GROUND TO PAYLOAD CENTER OF MASS (IN)	72.00
TIME	MAX. REAL TIME FOR SIMULATION	2.20
TRA1	HALF TRACK OF FRONT AXLE (IN)	41.000
TRA2	HALF TRACK OF FRONT TANDEM AXLE (IN)	41.000
VEL	INITIAL VELOCITY (FPS)	47.00
W	SPRUNG WEIGHT OF TRUCK (LBS)	8190.00
WS1	WEIGHT OF FRONT SUSPENSION (LBS)	1742.00
WS2	WEIGHT OF TANDEM FRONT (LBS)	2078.00
WS3	WEIGHT OF REAR TANDEM (LBS)	1972.00

```
BRAKE PARAMETERS: TQ(1,1,1) = 0.032 TQ(1,1,2) = 0.296
                  TQ(1,2,1) = 0.032 TQ(1,2,2) = 0.296
                  TQ(2,1,1) = 0.070 TQ(2,1,2) = 0.181
                  TQ(2,2,1) = 0.070 TQ(2,2,2) = 0.181
                  TQ(3,1,1) = 0.073 TQ(3,1,2) = 0.276
                  TQ(3,2,1) = 0.073 TQ(3,2,2) = 0.276

   TABLE 1 : TIME VS PRESSURE
         NO. OF POINTS: 1
      0.0          0.0


   TABLE 2 : PRESSURE VS TORQUE
         FRONT BRAKES, LEFT SIDE
         NO. OF POINTS:  2
      0.0          0.0
    100.0000    24000.0000


   TABLE 3 : PRESSURE VS TORQUE
         FRONT BRAKE, RIGHT SIDE
         NO. OF POINTS:  2
      0.0          0.0
    100.0000    24000.0000


   TABLE 4 : PRESSURE VS TORQUE
         FRONT TANDEM BRAKES, LEFT SIDE
         NO. OF POINTS:  2
      0.0          0.0
    100.0000    42000.0000


   TABLE 5 : PRESSURE VS TORQUE
         FRONT TANDEM BRAKES, RIGHT SIDE
         NO. OF POINTS:  2
      0.0          0.0
    100.0000    42000.0000


   TABLE 6 : PRESSURE VS TORQUE
         REAR TANDEM BRAKES, LEFT SIDE
         NO. OF POINTS:  2
      0.0          0.0
    100.0000    42000.0000
```

```
TABLE 7 : PRESSURE VS TORQUE
          REAR TANDEM BRAKES, RIGHT SIDE
          NO. OF POINTS:  2
       0.0            0.0
     100.0000     42000.0000

TABLE  8 : TIME VS STEER ANGLE (DEG)
          NO. OF POINTS:  1
       0.0           11.4000

*** END INPUT ***
```

6.0 FUTURE WORK

Ride, braking, and handling performance predictions will be made for the AM General Transbus as soon as the design has reached the stage wherein it is possible to estimate the needed parameter data and tires have been procured for measurement of the required mechanical properties. It is anticipated that further discussions of limit maneuver measures as they apply to a motor coach will lead to decisions on maneuver definitions that are appropriate to the task at hand. Calculations will then be made of the performance that can be expected when the prototype is completed and tested. These calculations will also serve as a guide for planning the test activity called for in the AM General Transbus subcontract.