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A FACTBOOK OF THE MECHANICAL PROPERTIES OF THE COMPONENTS FOR SINGLE-UNIT AND ARTICULATED HEAVY TRUCKS

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16. Abstract

This factbook provides a compilation of the mechanical properties of the components used in heavy trucks. It contains sections describing and discussing geometric layout, mass distribution, tires, suspensions, steering systems, brakes, frames, and hitches. Parametric data on heavy truck components are presented in a form suitable for use in analyzing the braking and steering performance of heavy trucks including combination vehicles. The influences of component properties on maneuvering performance are discussed.

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DISCLAIMER CONCERNING THE MENTIONING OF INDIVIDUAL MANUFACTURERS

The names of individual manufacturers are included in bar graphs illustrating ranges of mechanical properties for various components and vehicles. These identifications are not endorsements nor are they intended for comparisons between the products of individual manufacturers. The data have been gathered over a number of years and they are only isolated samples of typical results. Hence these data may not be the latest for any given component and may not necessarily represent any current truck or its components. Later design changes may be in effect and may change a given truck or component position in the bar charts.

The manufacturers' names have been retained because they provide identifying information concerning the type and source of data included in this Factbook. They furnish data entries that users of the Factbook may be able to associate with their own experiences.

A FACTBOOK OF THE MECHANICAL PROPERTIES OF THE COMPONENTS FOR SINGLE-UNIT AND ARTICULATED HEAVY TRUCKS

1.0 INTRODUCTION

1.1 Background

This Component Factbook was developed by The University of Michigan Transportation Research Institute (UMTRI) during a National Highway Traffic Safety Administration (NHTSA)-sponsored research study entitled "An Evaluation of Factors Influencing Heavy Truck Dynamic Performance." It contains descriptions of the components of heavy vehicles employed in trucking on highways in the United States.

These component descriptions are stated in terms of mechanical properties which can be used in analyzing the dynamics of heavy vehicles during braking and steering maneuvers that are required for roadway driving and the resolution of traffic conflicts.

A companion document [1], entitled "A Vehicle Dynamics Handbook for Single-Unit and Articulated Heavy Trucks," was also developed during this study. The Component Factbook complements the Vehicle Dynamics Handbook in that the Factbook provides parametric data on vehicle components in a form suitable for using the procedures outlined in the Handbook to evaluate the influence of component properties on the dynamic performance of typical heavy trucks and combination vehicles.

Large collections of parametric data describing the mechanical properties of heavy trucks have been assembled in previous programs [2,3]. However, these previous "factbooks" did not emphasize the relationships between mechanical properties and vehicle performance, nor did they attempt to summarize the information as is done here.

1.2 Scope

The commercial vehicle is viewed herein as an assembly of "major" units (that is, trucks, highway tractors, semitrailers, dollies, or full trailers). Each of these major units employ some or all of the following "basic" components:

1) tires

2) suspensions

3) steering systems

4) braking systems

5) frames

6) hitches

The major units are characterized by describing their geometrical layouts and mass distributions plus the pertinent mechanical properties of their basic components.

As illustrated in Figure 1.1, the information presented here is organized in a way that is intended to facilitate the analysis of braking and steering performance. Section 2 of this Factbook presents information on the basic components (see Figure 1.2). Section 3 of this Factbook describes the geometrical layouts and inertial properties of some currently employed tractors, trucks, semitrailers, full trailers, and dollies (see Figure 1.1).

The Factbook ends with an appendix presenting parametric data sets for "benchmark" (prototypical) vehicles that have been used in obtaining the results presented in the Vehicle Dynamics Handbook.

1.3 Organization of the Discussions

The discussions of components, geometric layouts, and mass distributions all contain subsections presenting the following information:

1st) Descriptions and definitions of pertinent mechanical properties

2nd) The importance of these mechanical properties to the braking and steering of heavy trucks

3rd) Ranges of values corresponding to the pertinent mechanical properties that have been measured or can be estimated.

The descriptions and definitions are aimed at explaining simplified, but generally powerful, representations of component *performance*. The relationships between pertinent mechanical properties and detailed or complex descriptions of component characteristics are discussed. The overall roles of the component or unit properties are given attention in the first subsection.

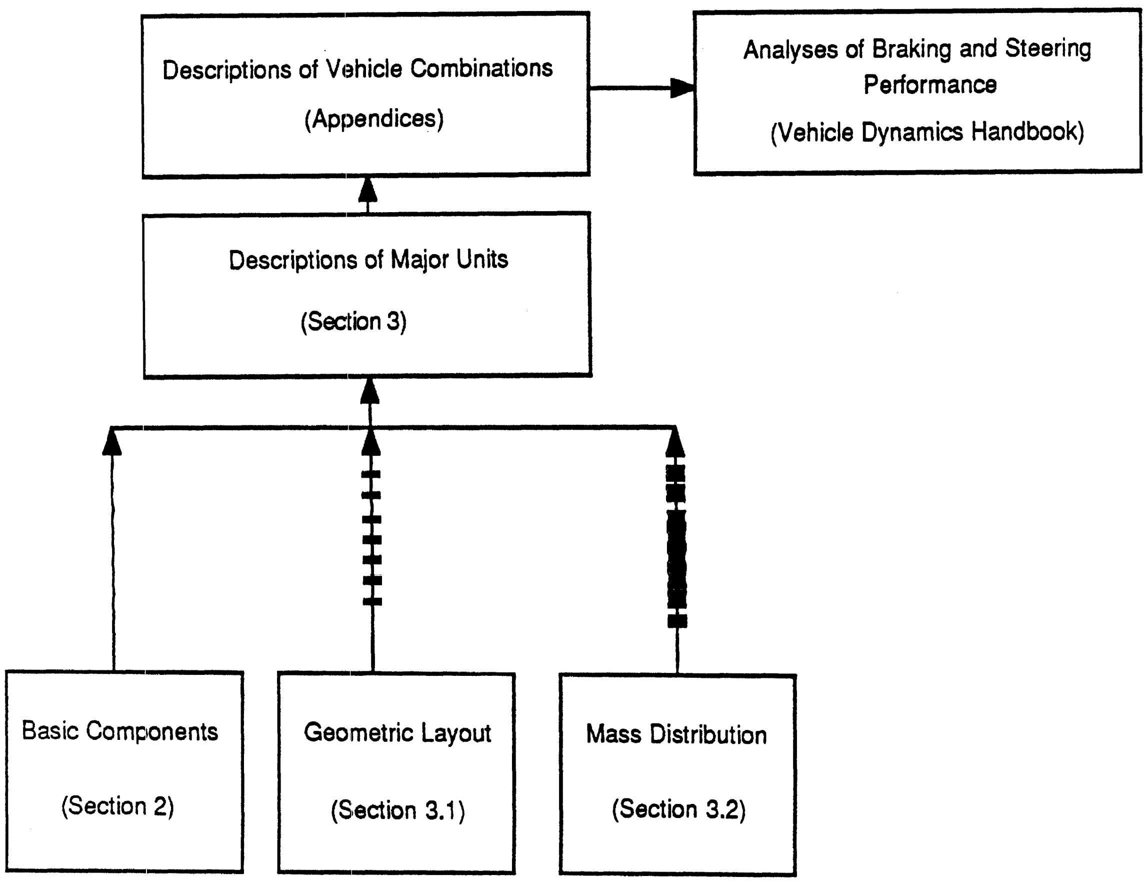


Figure 1.1 Overall information flow

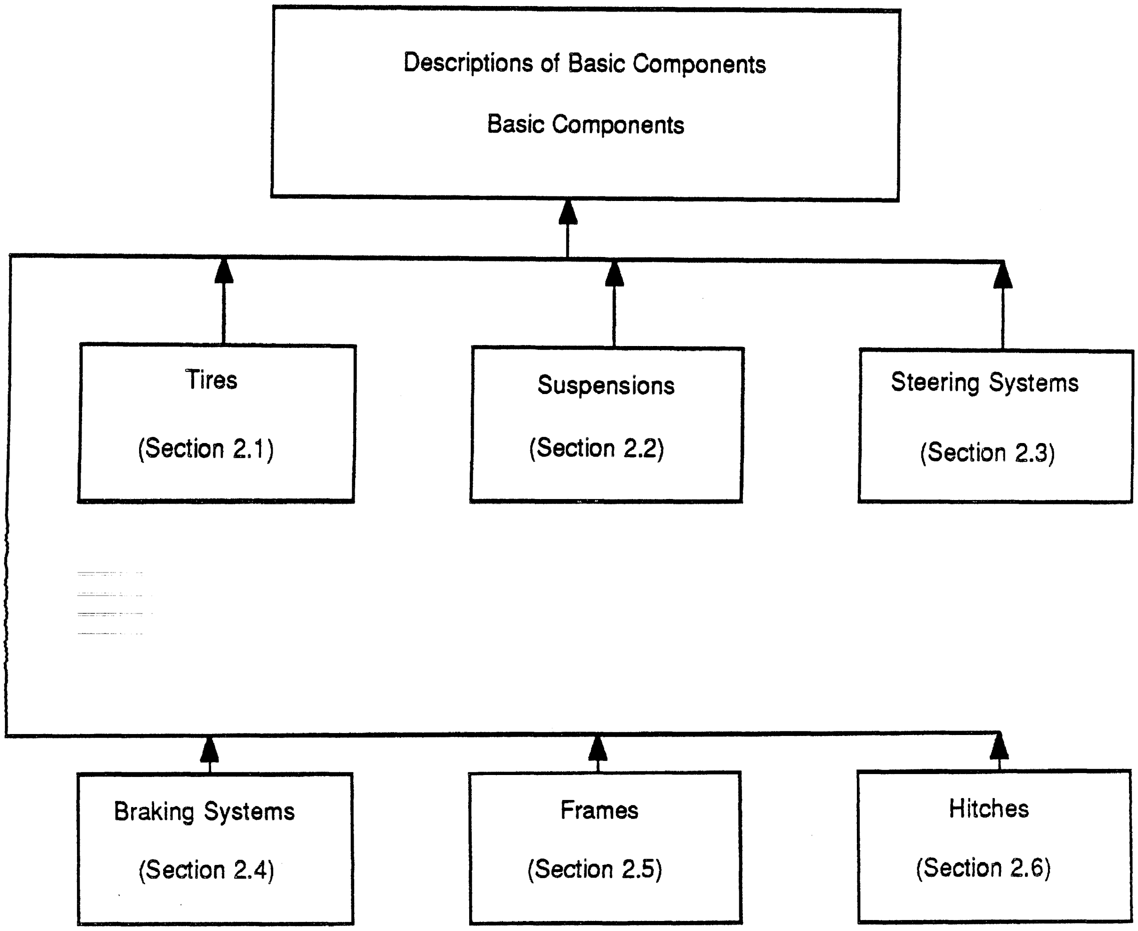


Figure 1.2. Components described in Section 2

In the second subsection on the importance of mechanical properties, specific "maneuvers" and associated performance "measures" are used in providing a quantitative assessment of the influence of mechanical properties on vehicle performance. In this regard, the "maneuvers and measures" correspond to those that are defined and utilized in the Vehicle Dynamics Handbook [1]. However, these maneuvers are easily related to driving experience, as evidenced by their names and performance measures; viz.,

Maneuver Name	Performance Measure
1. Low speed tracking (turning a corner at low speed)	offtracking of the rear with respect to the front
2. Constant deceleration braking	stopping capability (braking efficiency, friction utilization)
3. Steady turning at highway speeds	
a. Tracking	offtracking
b. Rolling	rollover threshold (the level of lateral acceleration at which the vehicle will roll over)
c. Handling	yaw stability (understeer/ oversteer factors)
4. Transient turning at highway speed	response times
5. Obstacle evasion (avoidance maneuvers such as an *emergency* lane change)	rearward amplification
6. Braking while turning on a slippery surface	directional control and stopping capability
7. Response to external disturbances	directional control
8. Mountain descent	brake fade (temperatures attained)

For the purposes of a qualitative evaluation of the types of situations where component properties are important, this Factbook relies on the readers' intrinsic understanding of the maneuver involved. Those interested in specific definitions and quantitative results are referred to the Handbook [1].

The third subsection (on ranges of values) provides bar charts illustrating the approximate spreads of mechanical properties existing in the current vehicle fleet. Specific values for typical types of components are indicated in these charts. Current differences between "generic" types of components are shown (for example, radial vs. bias tires, walking-beam vs. four-spring suspensions, etc.).

2.0 BASIC COMPONENTS

2.1 Tires

2.1.1 Mechanical Properties of Truck Tires. The tires on a truck or trailer produce the primary forces which cause the vehicle to turn, stop, or increase its speed. These forces are normally developed through elastic deformation of the tire's tread rubber and carcass structure. Most driving is done with the tire operating in this more-or-less elastic range, with the tire forces being insensitive to pavement effects. In severe maneuvers, however, the tread rubber begins to slide relative to the road and friction mechanisms limit the forces which can be developed.

The tire forces which are developed during braking and during application of engine power are determined by the longitudinal properties of the tire. The primary longitudinal properties involve the generation of the longitudinal force, F_x, as diagrammed in Figure 2.1.1. The force, F_x, acts parallel to the plane of the wheel.

The tire forces which are developed during cornering are determined by the lateral properties of the tire. The primary lateral properties involve the generation of the lateral force, F_y, and the aligning moment, M_z. As shown in Figure 2.1.1, the lateral force acts perpendicular to the wheel plane and the aligning moment constitutes a torque tending to rotate the wheel, in a steering sense, about the vertical axis.

While both the longitudinal and lateral tire forces serve to produce vehicle cornering and speed changes, the tire also serves to support the vehicle in the vertical direction. The load support function of the tire involves the vertical force, F_z, which derives simply from the deflection of the tire in the vertical direction.

Looking now at the tire in its overall role of maneuvering the vehicle by producing specific forces and moments, Figure 2.1.2 illustrates the pertinent mechanical properties, PMP, which most influence vehicle response, and the aspects of tire design and operation which most affect those properties. We see that the PMP's of truck tires can be summarized under five specific properties. Basically, the figure illustrates that the properties of most interest in the cornering and vertical support functions are not influenced by the operating variables which determine tire/pavement friction. On the other hand, all of the PMP's of the truck tire are influenced by the sum of the tire design and maintenance factors, plus the all-important vertical load level. In the subsections, below, the general response characteristics of the tire will be discussed and then the individual PMP's from each response category will be defined.

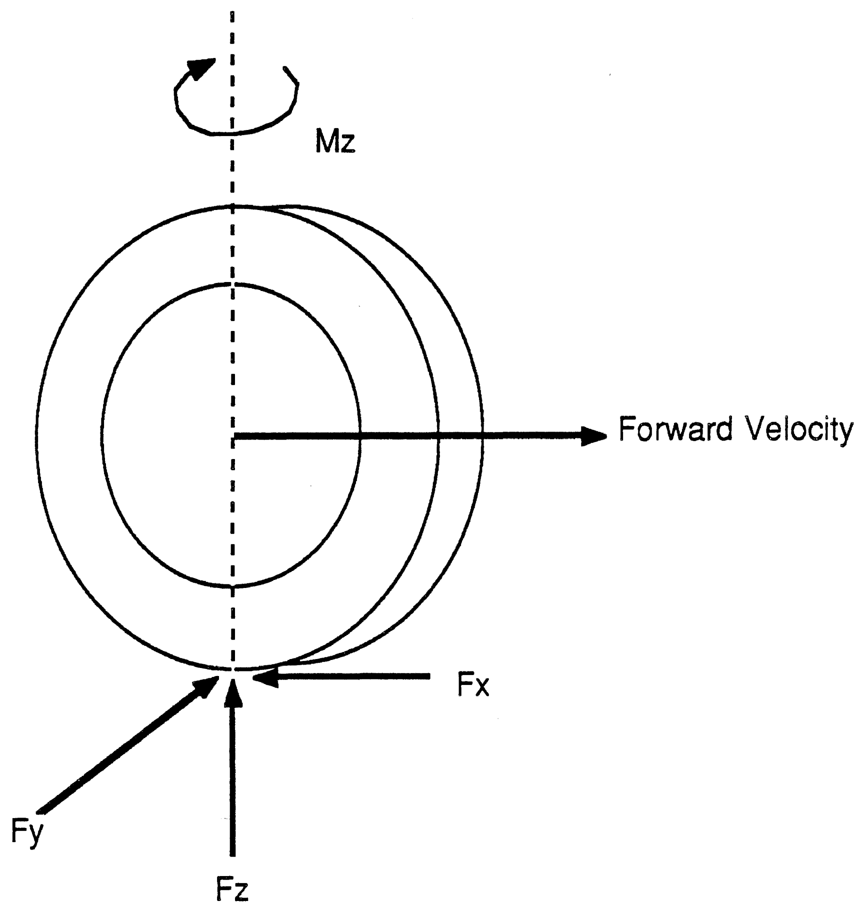


Figure 2.1.1 Definition of primary cornering and braking force and moments

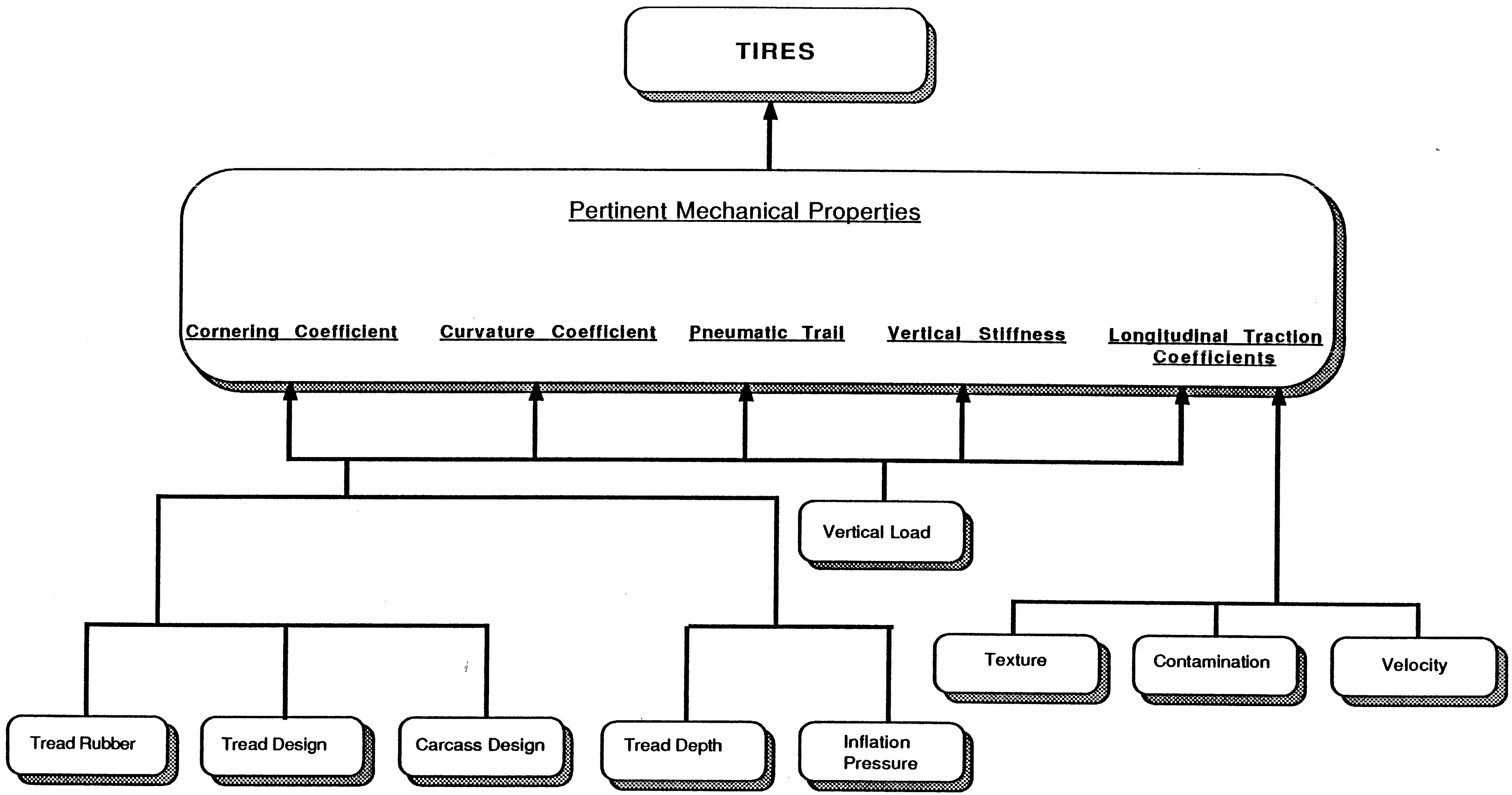


Figure 2.1.2 Factors influencing the pertinent mechanical properties of truck tires

2.1.1.1 Cornering properties. Shown in Figure 2.1.3 is a diagram of the tire moving over the road in a direction which is not exactly straight ahead. This "non-straight-ahead" condition is called "lateral slip" and implies that the tire must deform somewhat as it rolls along in such a condition. The deformations occur primarily in the vicinity of the tire's contact with the roadway and cause the development of a lateral force, F_y, generally a short distance aft of the tire center, thus also producing a so-called "aligning moment," M_z, tending to steer the tire/wheel assembly. The magnitude of the lateral force and aligning moment responses will be determined by the so-called "slip angle," alpha, which is shown on the figure. The longitudinal displacement, P_t, at which the lateral force acts is called the "pneumatic trail." Note, however, that Figure 2.1.3 addresses the tire itself, and not the steering or suspension properties which locate the tire on the vehicle.

Since all normal driving is done with low-level maneuvers, the lateral force and aligning moment levels are small, implying that the slip angle value is also small—within 4 degrees, or so. With relatively small slip angle, the tire is able to deform, thus following the non-straight-ahead direction of rolling, without suffering a significant amount of sliding in its contact with the road. Accordingly, primary lateral properties of interest are not influenced by frictional considerations such as pavement texture, water depth, and vehicle velocity. (Of course, under very low friction conditions such as ice and snow, even "normal" driving maneuvers become abnormal in the sense that the tire may be unable to generate the lateral forces needed to maintain vehicle control.)

Shown in Figure 2.1.4 is a plot of the basic relationship between lateral force and slip angle. We see that lateral force increases fairly steadily with slip angle, in the low range of values, and gradually flattens out as frictional mechanisms begin to limit the grip between the tire and the road. It is useful, in studying the response of vehicles, to focus on the "cornering stiffness" measure, C_{alpha}, which is indicated as the initial slope of this curve. This measure has units of pounds of lateral force per degree of slip angle.

Although data will be presented showing that C_{alpha} is influenced by all of the major tire design variables, the most important thing to recognize relative to vehicle behavior is that C_{alpha} is profoundly dependent upon the vertical load supported by the tire. Indeed, the two aspects of the C_{alpha} characteristic which are to be presented as PMP's are the following:

• The Cornering Coefficient, [C_{alpha} / F_z] (where F_z is at the rated load for the tire)

• The Curvature in the C_{alpha} vs. F_z relationship (This property can be understood by noting that, as load increases, the cornering stiffness, C_{alpha}, increases. The *rate* of this increase, however, tends to decline at higher loads and eventually becomes flat or even mildly negative. The

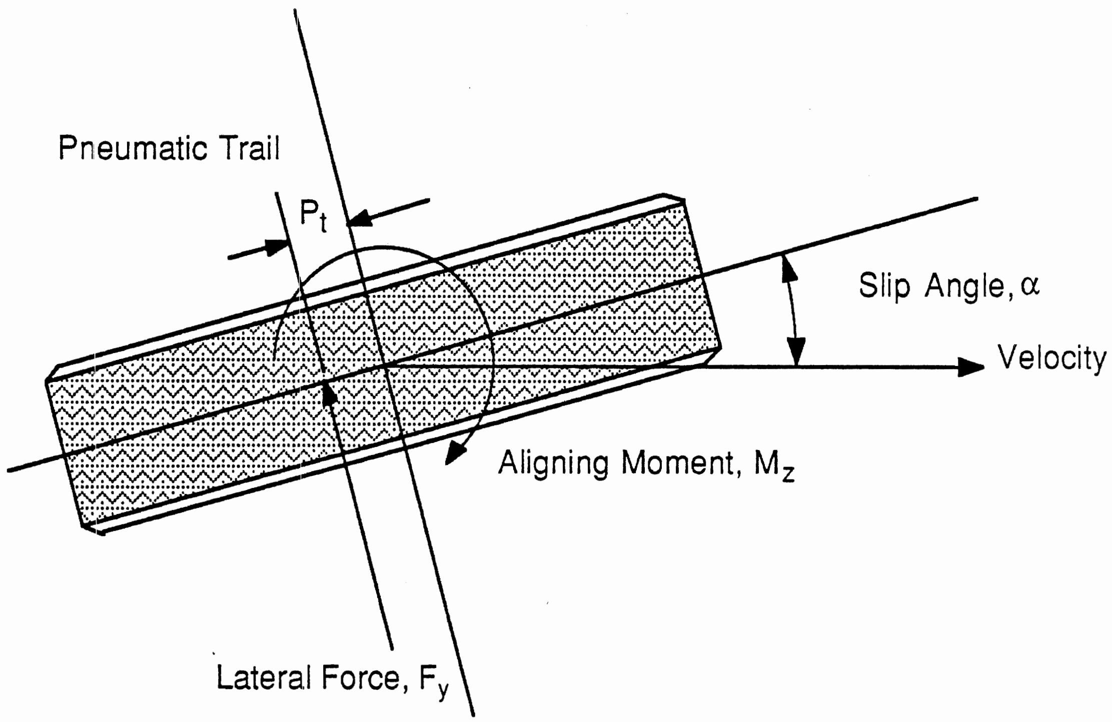


Figure 2.1.3 The tire operating at a slip angle (i.e., "pure cornering")

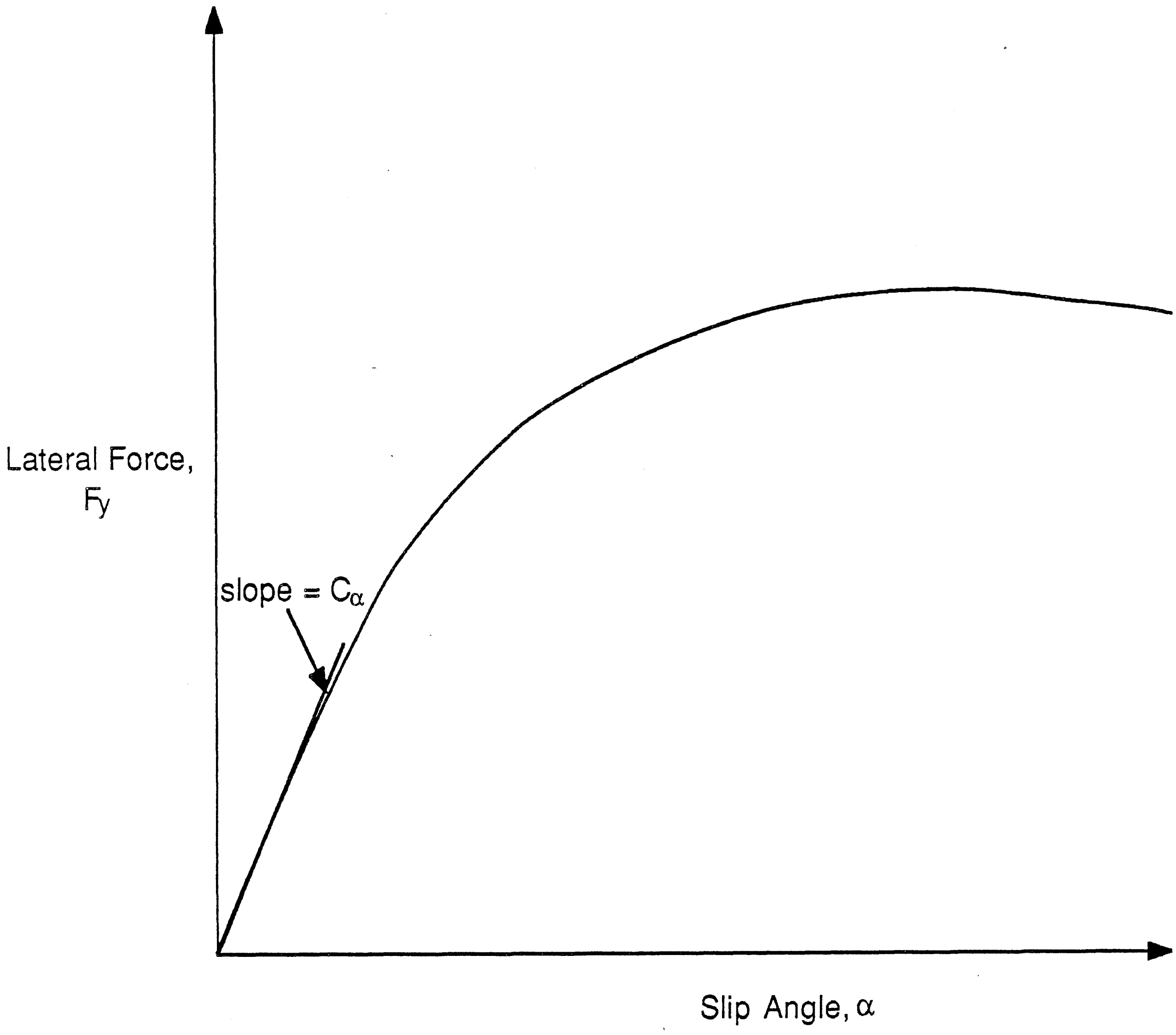


Figure 2.1.4 Lateral force vs. slip angle, illustrating cornering stiffness property, C_α

"fall-off" or curvature in the C_{alpha} vs. F_z relationship is defined by the value of a coefficient, C_2, which is multiplied by $(F_z)^2$ in fitting a quadratic function to the tire data (viz., $C_{alpha} = C_o + C_1F_z + C_2F_z^2$. Larger negative values of C_2 indicate that the (C_{alpha} vs. F_z) relationship is more strongly curved with increasing load.)

Shown in Figure 2.1.5 is a plot of the aligning moment response to slip angle. We see that, unlike the lateral force response, aligning moment rises quickly to a peak value and falls back toward zero as slip angle increases. The peaking in the aligning moment behavior is classic to all pneumatic tires and derives from the process in which an increasing portion of the tread rubber contacting the pavement begins to slide. At high slip angles, when essentially the whole contact area is sliding, there is no mechanism for the generation of a moment about the vertical axis of the tire, and the pneumatic trail dimension, P_t, becomes zero. In the vicinity of zero slip angle, however, it is convenient to define the pneumatic trail dimension as the reference value with which to compare the moment-generation behavior of various tires. This value of P_t is equal to the slope of the aligning moment curve, C_{Mz} (termed the "aligning stiffness"), divided by the cornering stiffness, C_{alpha}.

Since the aligning moment response is also profoundly influenced by the prevailing vertical load, it is useful to simply quantify the reference P_t property at the rated load condition. While such a measure provides a convenient indicator of an important tire property applying to all normal maneuvering of vehicles, the property is clearly of little value for addressing the aligning moments developed during severe maneuvers or, say, while traveling on ice and snow.

One mechanism of lateral force generation which is not of significance to the typical heavy–duty truck is that deriving from camber, or lateral inclination of the wheel. With passenger cars or other vehicles having independently suspended wheels, significant camber angles are produced and the tire, rolling at an inclined attitude, does develop a substantial lateral force as a result. With heavy-duty trucks and trailers, however, all wheels are mounted on solid axles which do not produce camber angles of significance except when the vehicle is rolling over. Thus, the lateral force response to camber angle is commonly neglected in the measurement of truck tire characteristics. It is recognized, however, that camber misalignment can be a significant cause of tire wear.

2.1.1.2 Longitudinal properties. Assuming that the tire is rolling straight ahead, the application of a brake torque on the wheel causes the wheel to slow down relative to its free-rolling speed. This "slowing" process produced so-called "longitudinal slip," causing the tire to experience deformations in the tread contact area. Longitudinal slip is basically expressed as a

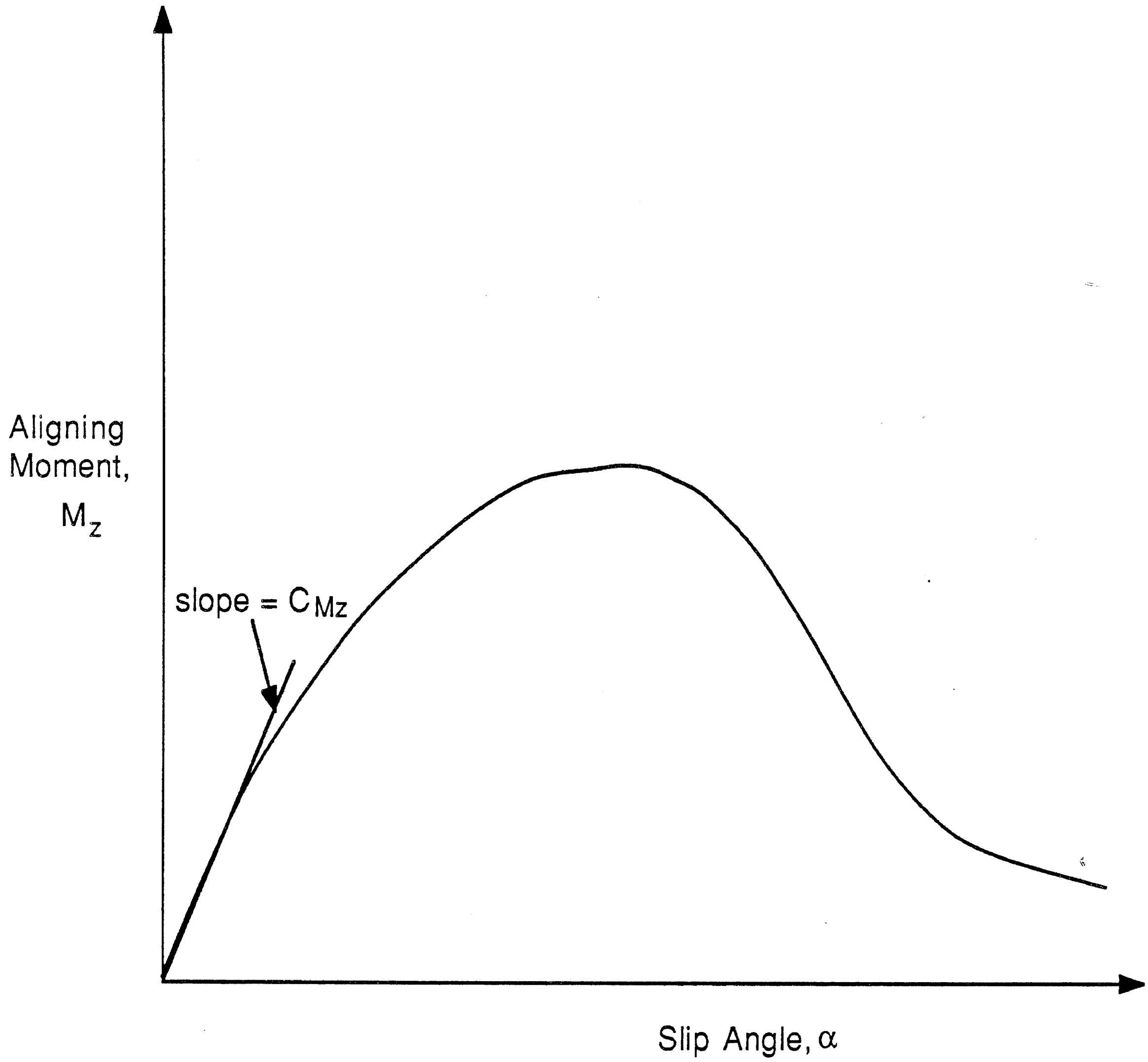


Figure 2.1.5 Aligning moment vs. slip angle, illustrating aligning stiffness, C_{Mz}

percentage indicating how close the wheel velocity is to the lock up condition. Longitudinal slip is zero percent, for example, when the tire is freely rolling and reaches 100 percent at lockup.

The deformations of the tread and carcass result in the development of a longitudinal force between the tire and the road. As with lateral force development, longitudinal forces are zero in the nonslip state and rise to limit values determined by frictional factors. Shown in Figure 2.1.6 is a characteristic plot of the longitudinal force, F_x, produced as a result of longitudinal slip, s. The figure illustrates two friction-limited features of the curve, namely, the "peak" and "slide" values of longitudinal force, which are useful for summarizing longitudinal force behavior. Since the friction forces are directly dependent upon the prevailing vertical load, the PMP's of the tire pertaining to longitudinal performance are defined as:

- Peak Longitudinal Traction, $[F_{x\,(peak)} / F_z]$

- Slide Longitudinal Traction, $[F_{x\,(slide)} / F_z]$

Since both of these measures are known to be determined by friction mechanisms, their values are strongly influenced by factors such as pavement texture, water, snow or ice covering, and vehicle speed.

Although the initial, elastic range, slope of the longitudinal force response curve is seen as analogous to the cornering stiffness parameter defined for lateral forces, this slope is not seen as having particular importance to the behavior of trucks.

2.1.1.3 Combined slip interaction. When a vehicle is being operated in a curve, with brakes being applied at the same time, the combined lateral and longitudinal slip conditions which prevail result in respective lateral and longitudinal forces which have a certain interdependence. To put it simply, there is only a fixed total level of frictional force that can be generated, and this fixed value will be "split up" between the two "demands" according to the respective slip levels which prevail. Shown in Figure 2.1.7 is a characteristic plot of the cross-influence of lateral and longitudinal forces. We see that, for differing values of slip angle, the level of lateral force declines sharply as longitudinal force approaches its peak value. Thus, strong braking in a curve raises the potential for losing a major portion of the lateral forces which would otherwise be developed if braking were absent.

Since there is essentially no data in the public domain addressing the combined slip behavior of heavy-truck tires, no PMP has been defined and no results are available in this document. Nevertheless, analysis of truck behavior under combined slip conditions has been

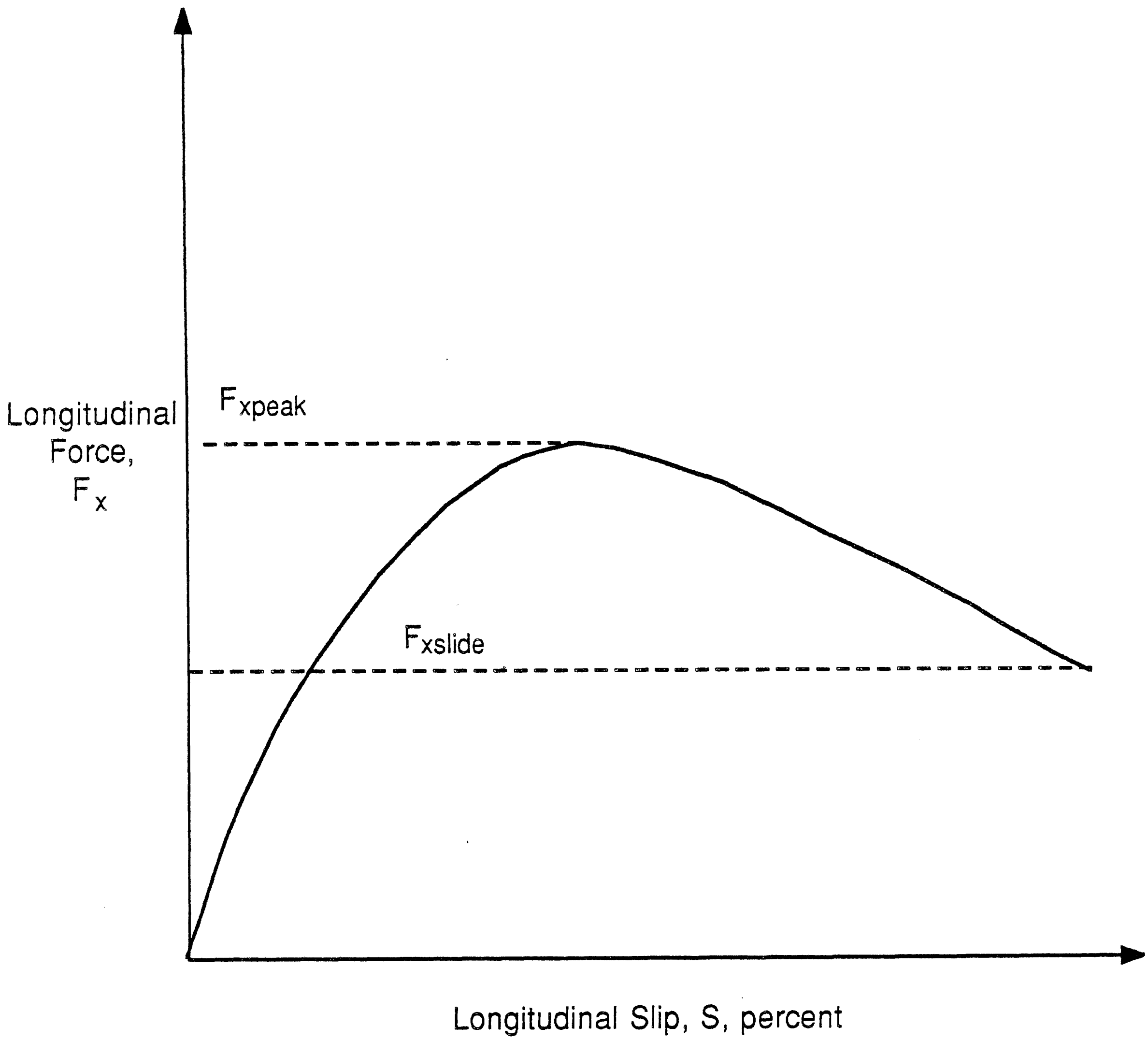


Figure 2.1.6 Longitudinal force vs. longitudinal slip, illustrating peak and slide measures, F_{xpeak} and F_{xslide}

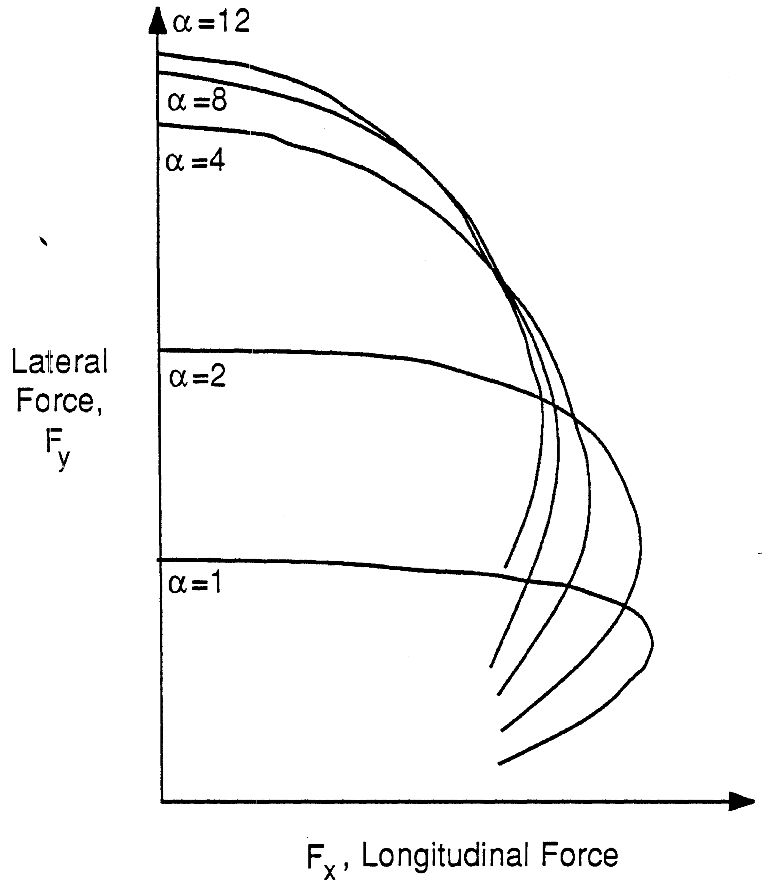


Figure 2.1.7 The cross-influence of lateral & longitudinal tire forces such as accrue during combined braking and cornering (note that this is hypothetical and not truck tire data)

undertaken using available lateral and longitudinal traction measurements, together with a theoretical model [4] of the interaction mechanisms. Such a "semi-empirical" method serves to enable analysis while actual combined slip data remain unavailable.

2.1.1.4 Vertical load support. The vertical stiffness of truck tires generally accounts for a significant portion of the vertical and roll "springing" of the vehicle. Thus, for example, the overall ride rates and roll rates of heavy-duty vehicles incorporate a strong influence from the tire's vertical stiffness. Since springing, in general, turns out to be important in determining the net roll stability of a loaded commercial vehicle, the vertical spring rate of the truck tire has been identified as a PMP in the load support function of the tire.

Shown in Figure 2.1.8 is a plot of the vertical load vs. vertical deflection of a truck tire. The plot illustrates that the initial deflection of the tire, from zero load, involves an initially nonlinear region connected to the nominally constant slope which prevails over most of the operating range of the tire. The vertical spring rate, expressed in pounds per inch of radial deflection, is defined as the slope of the relationship at a load value equal to the tire's load rating.

2.1.2 Importance of Tire Properties to Vehicle Maneuvering Behavior. This section provides a brief overview on the influence of specific tire properties on truck behavior. The Pertinent Mechanical Properties, plus certain other factors defining truck tire behavior will be cited in terms of both the level and the nature of their influences on truck response in braking and steering maneuvers. Shown in Table 2.1.1 is a summary of the levels of these influences for the maneuvering cases cited earlier in the Factbook. The table indicates the level of importance of each property simply by means of *High, Medium,* and *Low* designations. Each tire property on the table is discussed below.

2.1.2.1 Cornering coefficient, (C_{alpha} / F_z). The table shows that the cornering coefficient of the tire is unimportant in straight-line braking (where slip angles are zero), but has high importance in all maneuvers involving transient and steady-state turning at highway speed. Taking the maneuvering cases in which the level of importance is significant, the influences are as follows:

High-Speed Offtracking -- The cornering coefficient parameter is directly instrumental in determining the outboard offtracking of trailer axles in a high-speed turn. In order for the vehicle to achieve a given level of turn severity (described by the lateral acceleration level), the tires must operate at a slip angle in producing the needed level of lateral force. The cornering coefficient determines the magnitude of this slip angle, and thus the extent to which the trailing units "hang

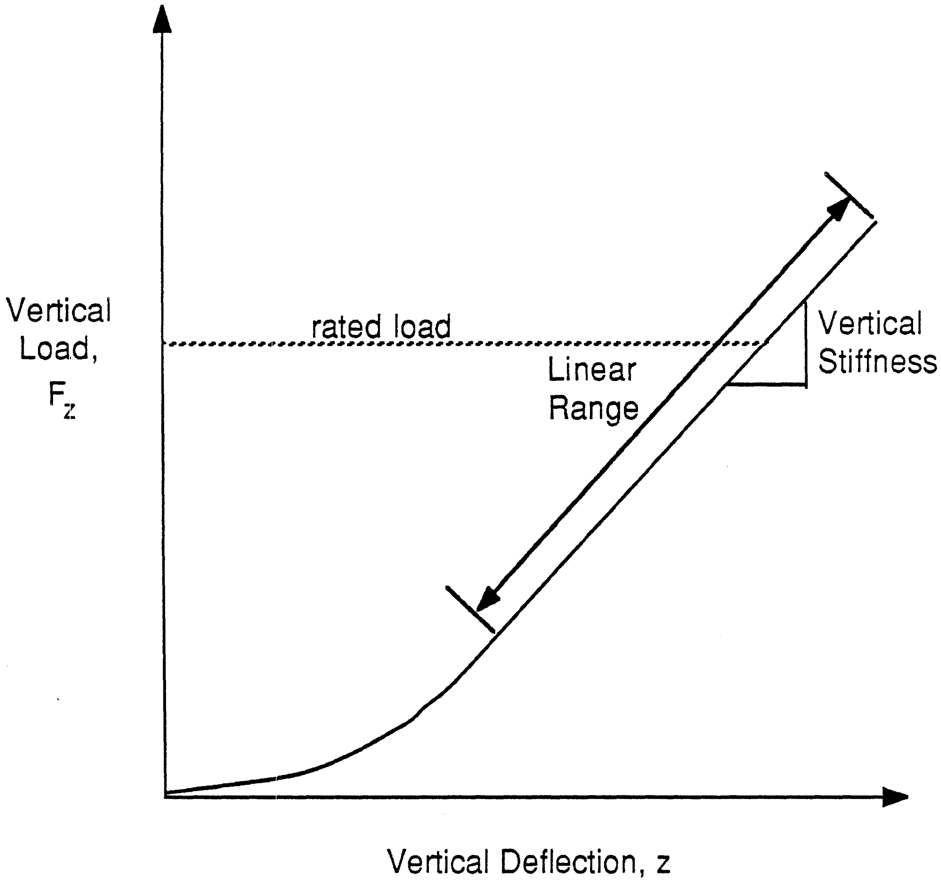


Figure 2.1.8 Vertical load vs. vertical deflection illustrating vertical stiffness at rated load

Table 2.1.1

The Importance of the Pertinent Mechanical Properties of Tires on Vehicle Dynamic Performance

Pertinent Mechanical Property of Tires	Low Speed Tracking	Hi Speed Tracking	Roll Stability	Handling Stability	Response Time	Rearward Amp	Braking Efficiency	Transient Braking & Turning	Downhill Braking	Response to Disturbances
Cornering Coefficient, C_α/F_z	-	Hi	-	Hi	Hi	Hi	-	Hi	-	Hi
Curvature in C_α (C_α vs. Vertical Load)	Low	-	-	Hi	Low	Low	-	-	-	-
Align. Stiff. (Pneumatic trail)	-	-	-	Low	-	-	-	-	-	-
Vertical Stiffness	-	-	Med	-	-	-	-	-	-	-
Peak Friction, m_p	-	-	-	-	-	-	Hi	Hi	-	-
Sliding Friction, m_s	-	-	-	-	-	-	-	Med	-	-
Long./Lat. Interaction	-	-	-	-	-	-	-	Med	-	-

out" in the turn to establish the slip angle. *Increased cornering coefficient results in reduced high–speed offtracking.*

Steady-State Handling Qualities -- Cornering coefficient is a direct determinant of the handling response of trucks or tractors to steer input. The so-called understeer property of the vehicle is, in fact, heavily determined by the difference in the cornering coefficients prevailing at the respective front and rear axles of the unit. Again, cornering coefficient influences this quality because it determines the magnitude of the tire slip angles which accompany a given turn severity. *Increased cornering coefficient on the front axle reduces understeer while increased cornering coefficient on the rear axle increases understeer.*

Transient Turning at High Speed -- The rapidity of a vehicle's response to an abrupt steer input is heavily determined by the cornering coefficient levels at all of the axles. Like in all mechanical systems, a stiffer system responds more quickly. *Increased cornering coefficient causes the vehicle to respond more quickly to an abrupt steer input.*

Obstacle Avoidance (Rearward Amplification of Trailing Units) -- The extent to which the successive trailing units in a vehicle combination tend to amplify the motions initiated by driver steer inputs is directly influenced by the sum of the cornering coefficient values prevailing over all of the axles of the combination. Simply put, higher cornering coefficient levels result in smaller tire slip angles which, in turn, result in smaller lateral motions at the hitch point which provide the excitation input to the successive units. *Increased cornering coefficient causes reduced rearward amplification.*

Braking in a Turn -- The cornering coefficient values determine the magnitude of slip angles at which the tires will operate in a given steady turn. If brakes are then applied, the reduction in tire lateral force ensuing due to longitudinal slip will be less if the initial slip angle value was less. Tire data are not available, however, for confirming the generality of this simplified view of the combined slip process. *The simplified theory indicates that increased cornering coefficient will improve controllability during braking in a turn.*

Response to External Disturbances -- When a lateral force is imposed upon a vehicle, such as due to a side wind, the extent to which the vehicle's motions are disturbed is heavily determined by the cornering coefficient values existing at the various axles. Again, the magnitude of the motion responses is directly determined by the slip angles which the tires must develop in order to produce the lateral forces needed to balance, say, the side wind. *Increases in cornering coefficient cause reduced response to lateral external disturbances.*

2.1.2.2 Curvature in the C_{alpha} vs. F_z relationship. The table shows that the curvature in the relationship between C_{alpha} and F_z is, again, confined to cornering maneuvers since this property of the tire influences only slip angle development. Figure 2.1.9 presents an illustration of the mechanism by which this curvature property influences the effective cornering stiffness level realized on an axle-by-axle basis. Firstly, it must be recognized that, when a truck travels in a curved path, the tires on the outside of the turn become more heavily loaded while the inside tires become more lightly loaded. The figure shows how the curvature in C_{alpha} vs. F_z interacts with these right/left changes in tire load during cornering to reduce the "average cornering stiffness" across both tires on an axle. That is, the increase in cornering stiffness due to increased load on the "outside" tire is much less than the loss in cornering stiffness due to reduced load on the "inside" tire on the same axle. Note that if the C_{alpha} vs. F_z relationship were a straight line, with curvature equal to zero, this so-called "lateral transfer of load" would have no net effect on cornering stiffness levels.

Clearly, then, the influence of the curvature property requires that the vehicle be operating in a maneuver having a lateral transfer of vertical load. The cases in which this occurs are noted as having some level of importance in Table 2.1.1 and are discussed individually, below.

High-Speed Offtracking -- The curvature in the C_{alpha} vs. F_z relationship has a small influence on the high-speed offtracking response since lower effective cornering stiffness levels develop at each axle due to lateral load transfer in a turn. The reduced cornering stiffness level, of course, causes the trailing units of the vehicle to track at higher slip angles, thus subtending paths which tend to fall outboard of the tractor path. *A more negative value of the curvature coefficient causes an increased level of high-speed offtracking.*

Steady-State Handling Qualities -- Because the primary steady-state handling quality, namely, the understeer level, is strongly dependent upon the front-to-rear balance in cornering coefficient levels, a peculiar set of mechanisms combine to render the curvature coefficient highly important in determining steady-state handling behavior of trucks and tractors. It works like this:

1) As noted above, the curvature property becomes important according to the level of lateral load transfer experienced on an axle.

2) Trucks and tractors are virtually always designed with rear axle suspensions which are much stiffer, and thus experience a much higher proportion of lateral load transfer in a given turn than front axle suspensions.

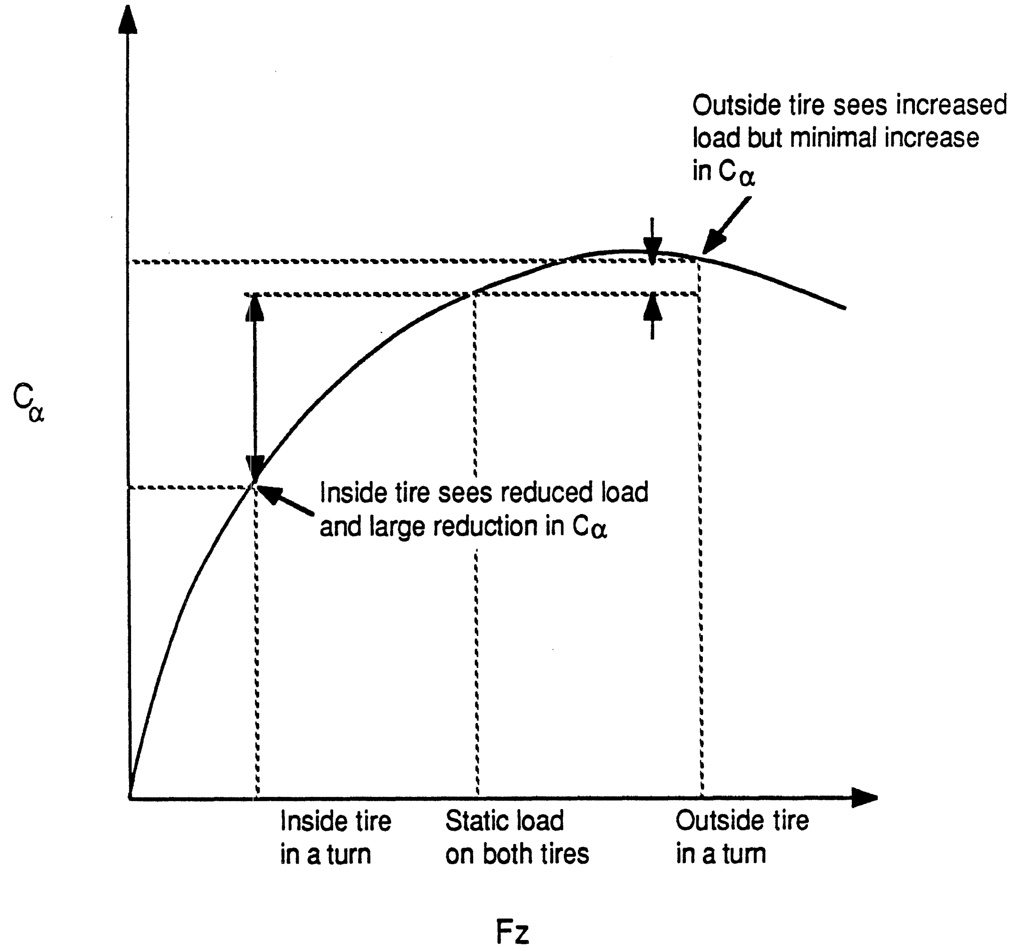


Figure 2.1.9 An illustration of how the curvature in (C_α vs. Fz) causes a net reduction in C_α due to load transfer in a turning maneuver

3) With the rear tires experiencing large amounts of load change while cornering, relative to the front tires, the effective cornering stiffness level prevailing at each rear axle suffers a greater net loss than does the front.

4) As a result, the front-to-rear balance in cornering coefficient values makes a decided shift in the direction which reduces understeer and tends to bring about yaw instability.

5) The magnitude of this shift is determined by the total changes in cornering coefficient which have accrued, respectively, at the front vs. rear axle as a result of load transfer.

A more negative value for the curvature coefficient on front-mounted tires causes a small increase in understeer level. A more negative value for the curvature coefficient on rear-mounted tires causes a large reduction in understeer level.

Transient Turning at High Speed -- To the degree that the response times of trucks and tractors change, methodically, with the understeer level, the influences cited in the above section on steady turning apply here. That is, greater levels of understeer are associated with reduced response times, or quicker yaw response. Thus, one aspect of the influence of curvature coefficient on transient turning response can be stated as: *A more negative value for the curvature coefficient on front-mounted tires causes a small reduction in yaw response time. Conversely, a more negative value for the curvature coefficient on rear-mounted tires causes a substantial increase in yaw response time.*

By the simpler mechanisms described earlier, the fact that the curvature characteristic represents a means for reducing the net level of cornering coefficient on any axle indicates that *a more negative value for the curvature coefficient on any axle of a vehicle combination tends to make for a more sluggish response.* The net effect of this mechanism on the yaw response time of a truck or tractor depends upon the balance of properties at the front and rear axles. Further, since the curvature mechanism depends upon the achievement of lateral load transfer, there is an issue involving the phase relationship between the load transfer transient and the yaw transient in which tire cornering stiffness is important.

Obstacle Evasion (Rearward Amplification of Trailing Units) -- The importance of the curvature coefficient in the rearward amplification response is relatively small and derives simply from the effect of reduced cornering coefficient following load transfer. The overall influence of the curvature property on all of the tires in a combination vehicle is more-or-less determined by the net sum of the cornering coefficients which are achieved. Here again, however, the phasing of the load transfer transient at each axle with the yaw response of the involved vehicle unit will heavily determine the importance in individual cases. *More negative values of curvature coefficient will cause small increases in rearward amplification.*

2.1.2.3 Pneumatic trail (P_t). The pneumatic trail of the tire determines the magnitude of the steering moment which is applied to the tire during cornering. Although such aligning moments are generated at all tire positions on the vehicle, the only significance of this property arises on the steering axle of the truck or tractor. As indicated in Table 2.1.1, the pneumatic trail dimension is seen as having a low, but significant, influence on the yaw stability characteristic. This influence derives from the fact that the steering system of heavy-duty trucks and tractors is compliant, or flexible, to a certain degree and thus permits the steer angle of the front wheels to deflect in response to aligning moment. This deflection response plays a moderately significant role in determining the understeer level of the vehicle. *Increased pneumatic trail at tires installed on the steering axle causes an increase in the understeer level.*

2.1.2.4 Vertical stiffness. The vertical spring rate of the tire is of importance as an element of what might be called the "total suspension system" on the vehicle. The only response category in which the tire's vertical stiffness is seen to significantly influence performance is in connection with the rollover threshold. As shown in Table 2.1.1, the vertical stiffness parameter has a medium level of influence on rollover threshold. This influence stems from the fact that any softness in the total suspension system permits the body and payload on the vehicle to roll toward the outside in a turn, and thus to suffer a destabilizing lateral translation of the center of gravity. Accordingly, *an increase in tire vertical stiffness tends to cause an increase in vehicle roll stability.* The magnitude of this influence is largely determined by the matching of suspension stiffnesses to the loads carried on the respective axles.

2.1.2.5 Peak longitudinal traction coefficient, (F_{xp}/F_z). The peak longitudinal traction level of the tire determines the maximum level of normalized braking force which can be reached in a limit stopping condition, without wheel locking. Thus, of course, this parameter is paramount in determining the level of deceleration that can be achieved under a given set of conditions. The prevailing "conditions" of importance are represented by a given pavement, vehicle velocity, and surface contamination state (e.g., water, snow, ice, etc.). A subject tire is superior in traction performance if it produces high values of (F_{xp}/F_z) relative to the values achieved by some *reference* tire under the same conditions. Whatever traction level the truck tire produces, the "braking efficiency" of the overall vehicle will then be determined by the adequacy of the system which proportions brake torques among all the axles.

Moreover, *increased values of (Fxp/Fz) do not influence braking efficiency, per se, but certainly do enable higher levels of deceleration during emergency braking.*

2.1.2.6 Slide longitudinal traction coefficient, (F_{xs}/F_z). The slide value of traction applies to the locked-wheel condition and simply indicates the frictional coupling obtained in that mode of operation. Since wheel lockup is generally associated with loss-of-control because of the virtual absence of lateral force potential, this measure is not used directly in any figures of merit of overall vehicle performance. When braking in a turn, however, the lateral transfer of load from inside to outside tires assures that the inside tires will lock up (generally without serious implications for loss-of-control). In such cases, the prevailing level of slide traction (on the locked wheels) will contribute to determining the overall deceleration level achieved. *Increased slide traction values serve to increase deceleration capability during locked-wheel braking and will increase the apparent "efficiency" of the braking process in a turn.*

2.1.3 Presentation of Characteristic Values. In this section, available data representing typical values for the tire parameters discussed above will be presented. The data are presented in the form of the pertinent mechanical properties which embody the most important tire properties governing vehicle response.

2.1.3.1 Cornering coefficient, (C_{alpha}/F_z). Shown in Figure 2.1.10 is a display of values of cornering coefficient evaluated at the rated load of each of the sample of tires. The figure reveals the following:

• The known range of the cornering coefficient parameter, for new tires in common service, covers values from 0.08 to 0.16.

• Bias-ply tires having lug, or "traction-style," tread designs fall in the lowest portion of this range, with typical values in the vicinity of 0.085.

• Bias-ply tires having tread designs of the "highway-rib" type fall in the intermediate range, with typical values in the vicinity of 0.10.

• Radial-ply tires of differing tread design types cover the upper end of the range, typical values in the vicinity of 0.13. Radial tires manufactured by domestic U.S. companies occupy the lower portion of the radial tire range, with typical values in the vicinity of 0.115.

• When tread wear is accrued, the cornering coefficient always rises (because the height of the "cantilever spring" constituting the tread reduces, thus increasing the tire's total cornering stiffness). The cornering coefficient of an example radial-ply (rib-tread) tire is seen to rise by

Sample of Cornering Coefficient Values Measured at Rated Load

$$\left(\frac{C\alpha}{F_z}\right)_{F_z=\text{Rated Load}}, \ \text{deg}^{-1}$$

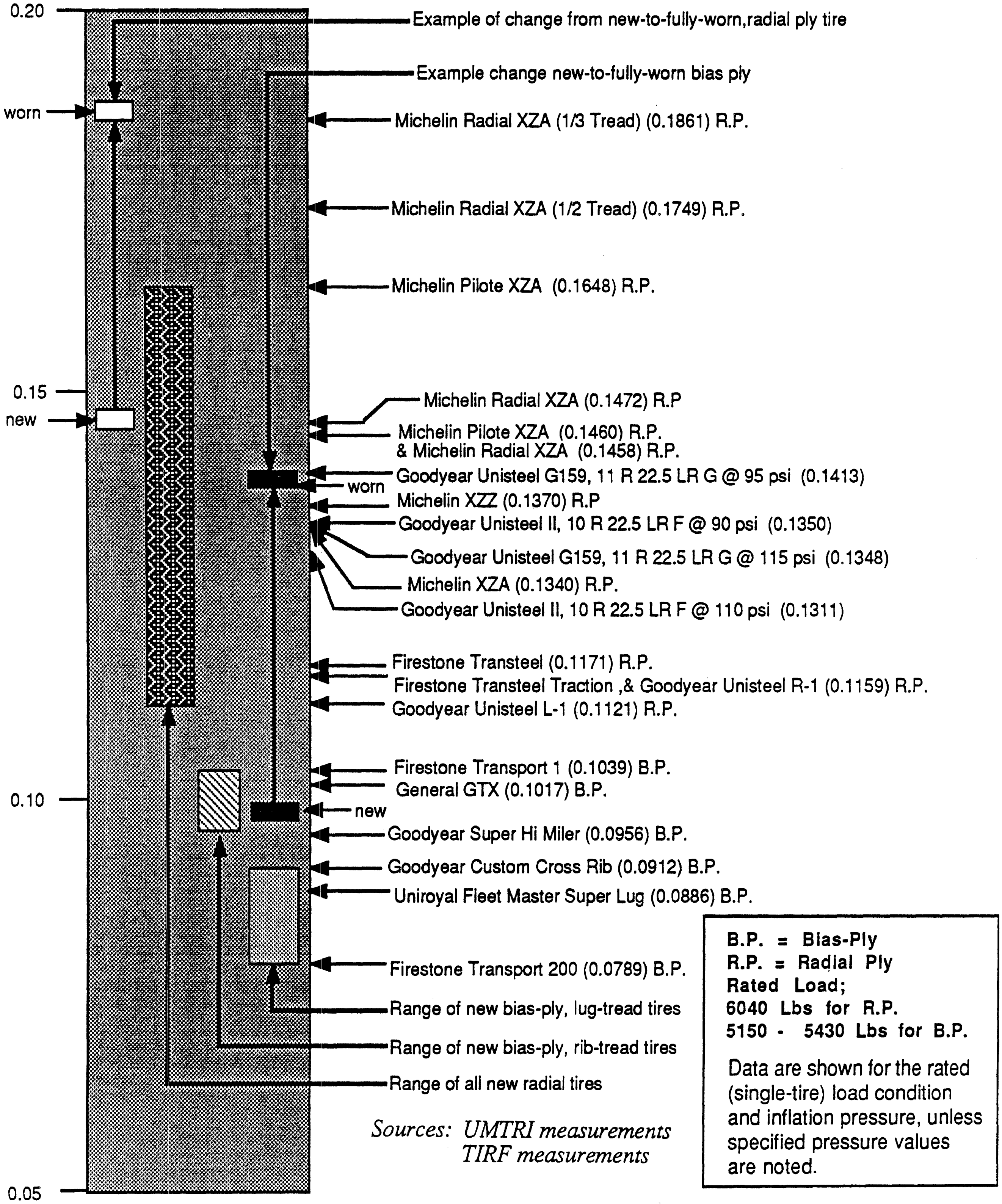


Figure 2.1.10 Cornering coefficient

approximately 0.04 when the tread depth goes from "as-new" to 1/3 of its as-new value ("fully worn").

• Similarly, the cornering coefficient of an example bias-ply (rib-tread) tire is seen to rise by approximately 0.045 when the tread depth goes from "as-new" to "fully worn." (Reflecting further on the fact that treadwear is simply influencing the "tread spring" portion of the overall cornering stiffness "spring," it may be reasonable to generalize that the nominal increase in cornering coefficient with treadwear should be roughly the same with all tires having similar as-new tread depths, regardless of carcass type. It also follows that since as-new tread depths are greater with lug-type treads, the increases in cornering coefficient accompanying the treadwear of lug tires will be correspondingly greater than those shown for example rib treads.)

It is also known that the cornering coefficient of truck tires is not predictably influenced by inflation pressure. In contrast to the predictable rise in cornering coefficient of car tires with increased inflation pressure, the cornering coefficient of truck tires has been seen to vary markedly in both the plus and minus direction with increased inflation pressure—presumably as a consequence of nuances in carcass design.

2.1.3.2 Curvature in the (C_{alpha} vs. F_z) relationship. Figure 2.1.11 displays the range of available data representing the "curvature coefficient" defined earlier. The figure reveals the following:

• The range of values for curvature coefficient seen with new truck tires is from -5.7 to –17.4.

• Methodical differences are seen in distinctions between different types of tire construction. Bias-ply, lug-tread tires occupy the lower end of the range of curvature coefficient values. A typical value for bias-ply, lug-tread tires would be -6.5.

• Bias-ply, rib-tread tires occupy intermediate values in the range of curvature coefficients. A typical value for bias-ply, rib-tread tires would be -10.5.

• Radial-ply tires of all tread designs occupy the upper region of curvature coefficient values, but also include samples which overlap the data for bias-ply rib tires.

• An example radial-ply, rib-tread tire shows a very large increase in the value of the curvature coefficient as a consequence of reduced tread depth. Comparing the curvature coefficient values obtained in the as-new and the 1/3-tread-depth state, an increase of -7.5, or approximately 50 percent, is observed.

Sample of Curvature Coefficient Values Measured at Rated Load

$C_2 \times 10^6$, $(\text{lb-deg})^{-1}$

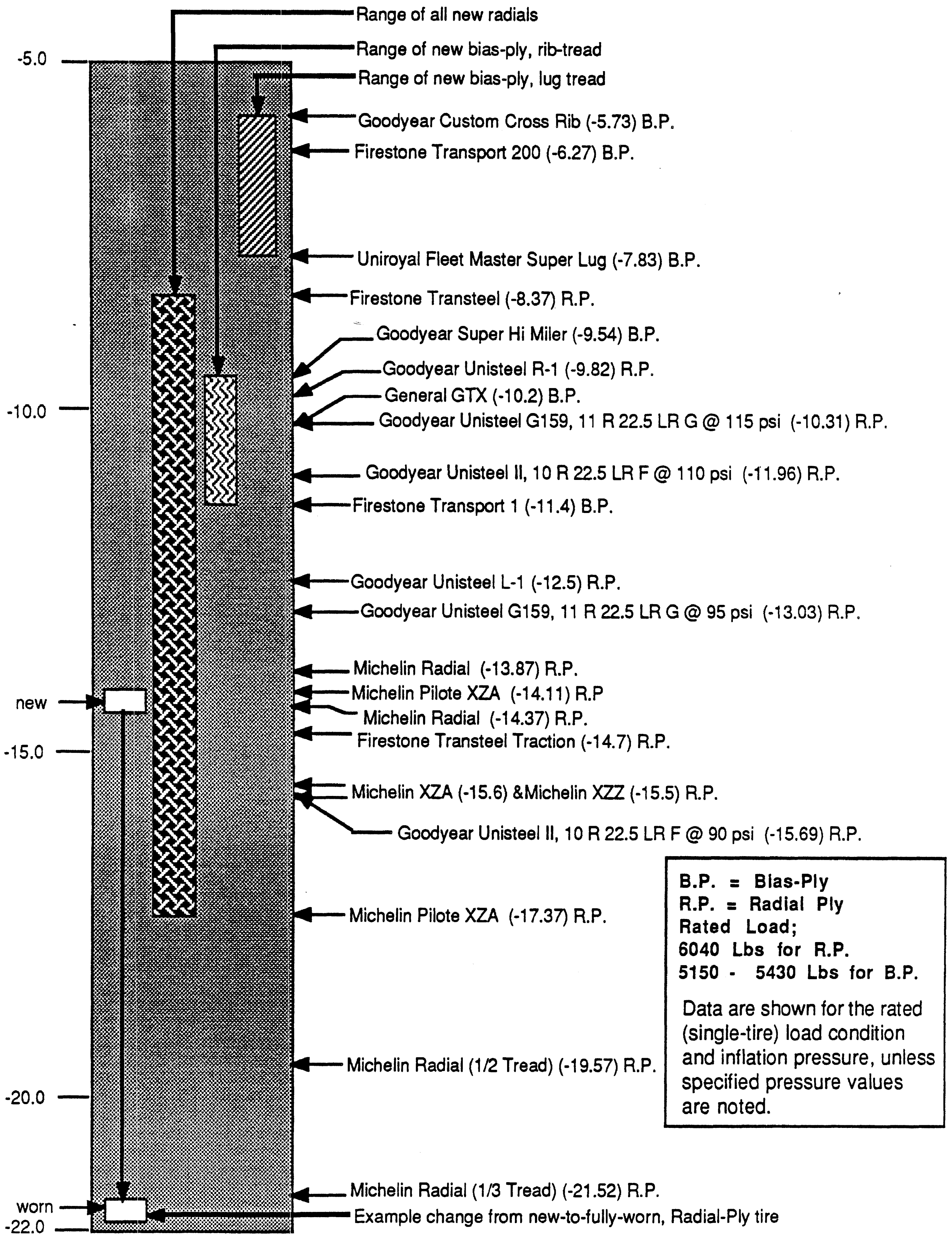


Figure 2.1.11 Curvature coefficient *Sources: UMTRI measurements TIRF measurements*

The mechanisms determining the influence of obvious tire design features on the curvature coefficient have not been identified. Thus, in contrast to the general means for relating changes in tread depth to the cornering coefficent, (C_{alpha}/F_z), as mentioned in the preceding section, the complexity of the curvature property currently precludes ready generalizations. Nevertheless, one correlation is quite obvious. Simply put, tires having a large value of cornering coefficient will certainly exhibit a relatively large (negative) value of curvature coefficient.

2.1.3.3 Pneumatic trail, (P_t). The pneumatic trail value, measured at rated load for a sample of tires, is shown in Figure 2.1.12. The data show the following:

• The pneumatic trail dimension for tires in common highway service covers a range of values from 1.8 to 2.8 inches.

• The tires in this sample indicate generally higher values of pneumatic trail with bias-ply tires than with radials. A typical value for new bias tires is 2.3 inches while a typical new radial would be approximately 2.1 inches.

• Pneumatic trail is significantly affected by treadwear. With both bias and radial tires, pneumatic trail is seen to increase on the order of 10% from the as-new to fully-worn tread depth condition.

2.1.3.4 Longitudinal traction coefficients, ($F_x/F_z)_{peak}$ and $(F_x/F_z)_{slide}$. Longitudinal traction coefficients have been compiled for summary presentation using values for 40 mph, only. Shown in Figure 2.1.13 are peak and slide traction coefficients measured with a sample of truck tires on a wet concrete pavement. The pavement was aggressively textured, such as exists on relatively new surfaces meeting the requirements of the Federal Interstate Highway System. Nevertheless, since the friction potential of pavements varies tremendously over the range of physical sites and weather situations, the absolute values shown in the figure have no general significance. The data show the following:

• There is a substantial range of traction coefficient values exhibited for this set of tires which were uniformly tested under the same pavement, water depth, and velocity conditions. The peak values range from 0.57 to 0.83 and the slide values range from 0.38 to 0.58.

• The data illustrate large "falloff" from the peak to the slide values with individual tires. The ratio of the peak value to the slide value ranges from approximately 1.3 to 1.6.

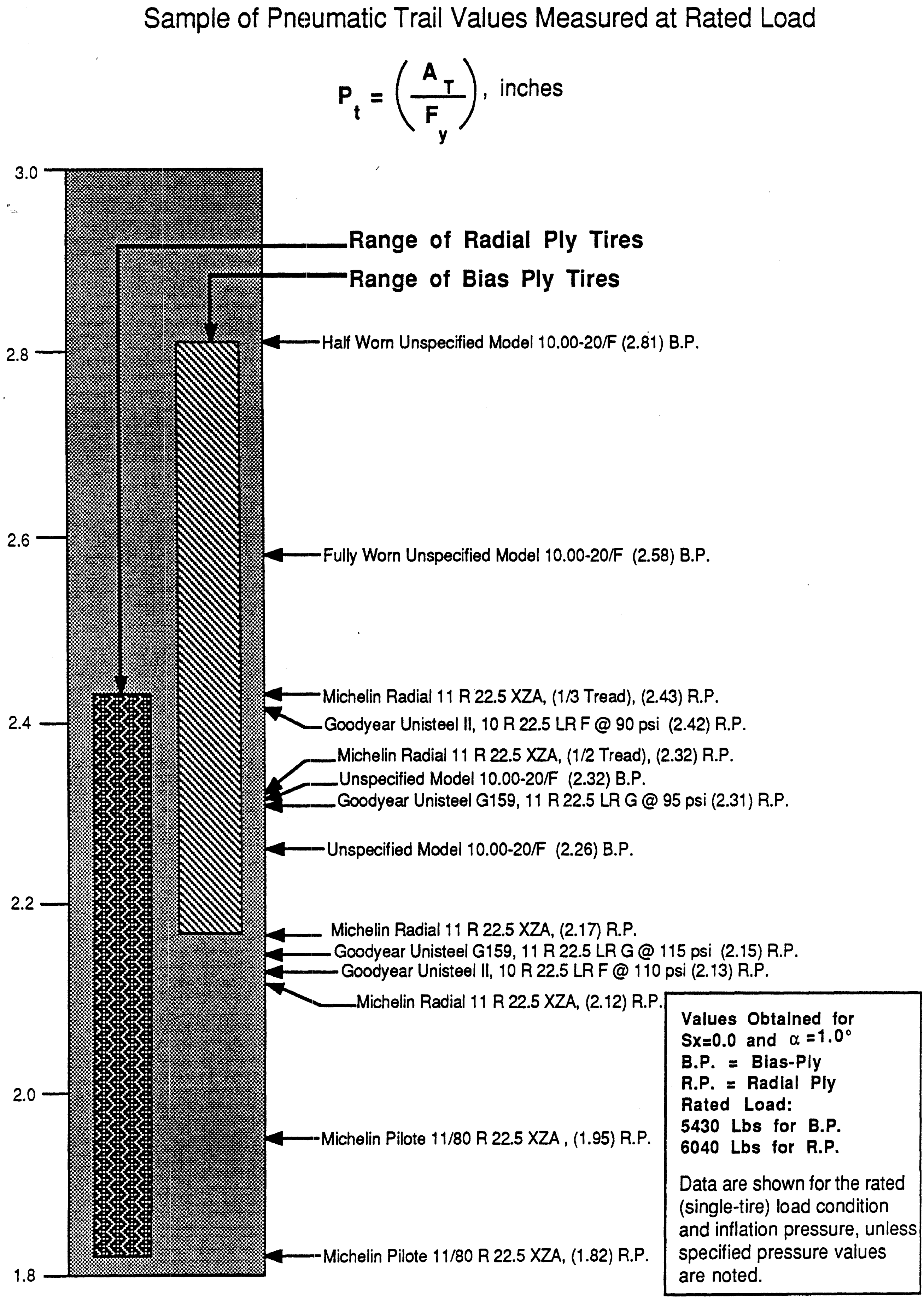


Figure 2.1.12 Pneumatic trail, P_t

Source: UMTRI measurements

Sample of μ_p and μ_s at Rated Load and 40 MPH.
(Wet Surface)

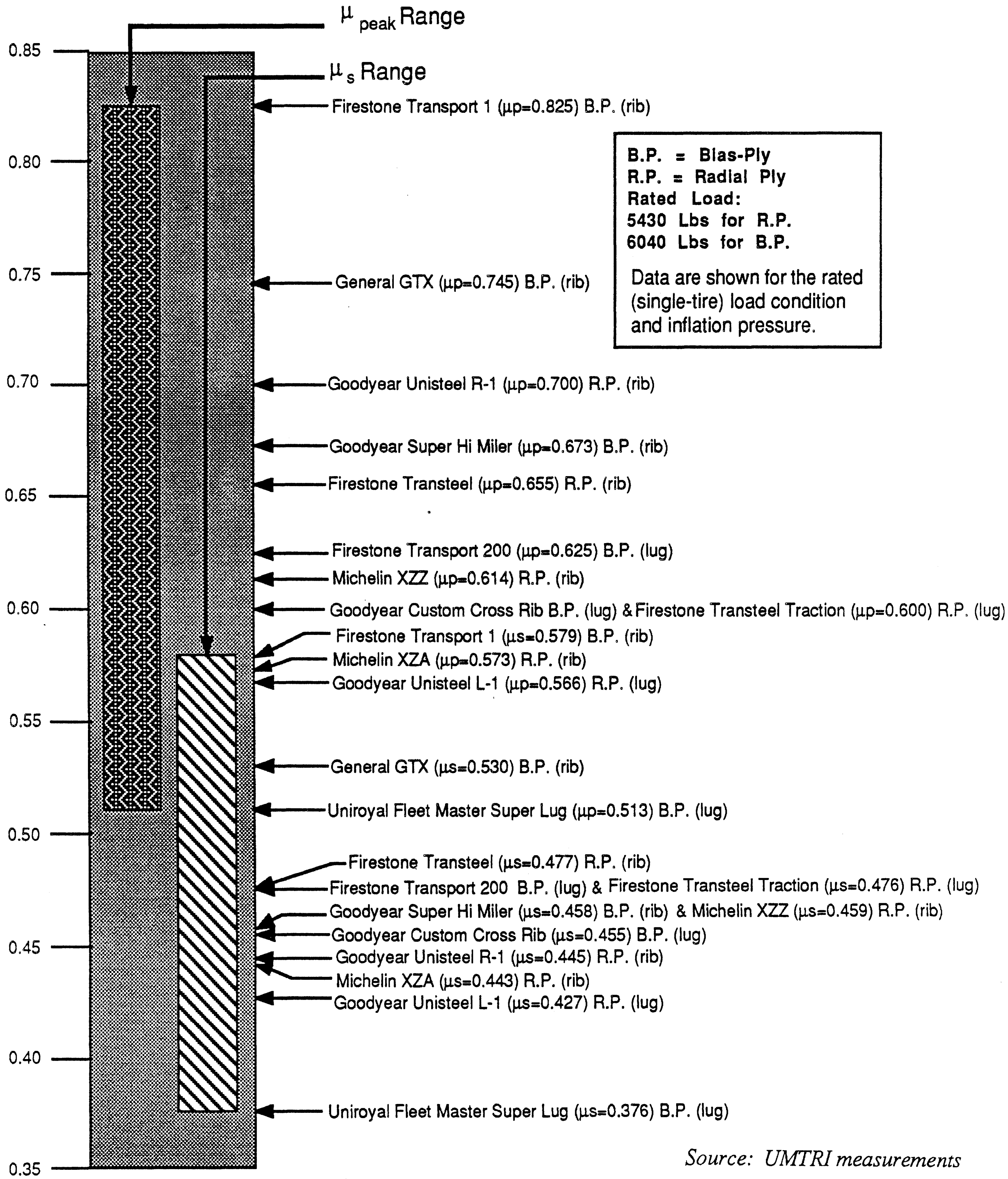


Figure 2.1.13 Peak and slide traction coefficient values

• Little in the way of clear distinctions exist in terms of the traction performance levels of these sample bias-ply vs. radial-ply tires. Nevertheless, data show that the higher performance levels are achieved by bias-ply, rib-tread tires.

• A substantial distinction exists between the traction performance levels of bias-ply tires having rib- vs. lug-type tread designs. Rib-type bias tires are generally seen to occupy the upper portions of the ranges of both peak and slide traction values.

Shown in Figure 2.1.14 is the corresponding data set for the same group of tires measured on the same surface in the dry condition. The data show the following:

• The range of dry peak and slide traction values is considerably narrower than that seen on the wet pavement. Peak values on dry pavement range from 0.72 to 0.85 while slide values range from 0.51 to 0.60.

• The ratio of the peak value to the slide value for individual tires is fairly uniformly near 1.40.

• The peak values of the rib-type, bias-ply tires again lie in the upper portion of the overall range of data. No significant distinctions exist, however, in the placement of differing tire types in the slide traction data.

Although these data do not incorporate tread depth variations, other results for tires on various surfaces show major losses in traction levels with declining tread depth. For example, at 31 mph, peak traction values on wet pavement declined by 20 to 40 percent and slide values by 25 to 50 percent when tread depth was reduced from the as-new to the fully-worn condition [4]. At 62 mph, peak traction values declined by 35 to 60 percent and slide values by 40 to 70 percent over the range of tread depths [4].

Additionally, research has shown that complete hydroplaning of very lightly loaded truck tires (which happens on rear axles of unloaded vehicles) can occur when (a) tread depth is low, (b) pavement surface texture is relatively smooth, and (c) vehicle speed is above 60 mph, or so. "Complete hydroplaning" implies that longitudinal and lateral traction capability is essentially zero. The phenomenon develops to an exaggerated degree with truck tires because of the very short but wide contact patch geometry which derives under very light load conditions. Insofar as tire load can reach a remarkably low fraction of rated load at the dual-tire installations of empty trucks and combination vehicles, the truck tire is seen as unusual among tires in motor vehicle service for its exposure to this traction-loss phenomenon.

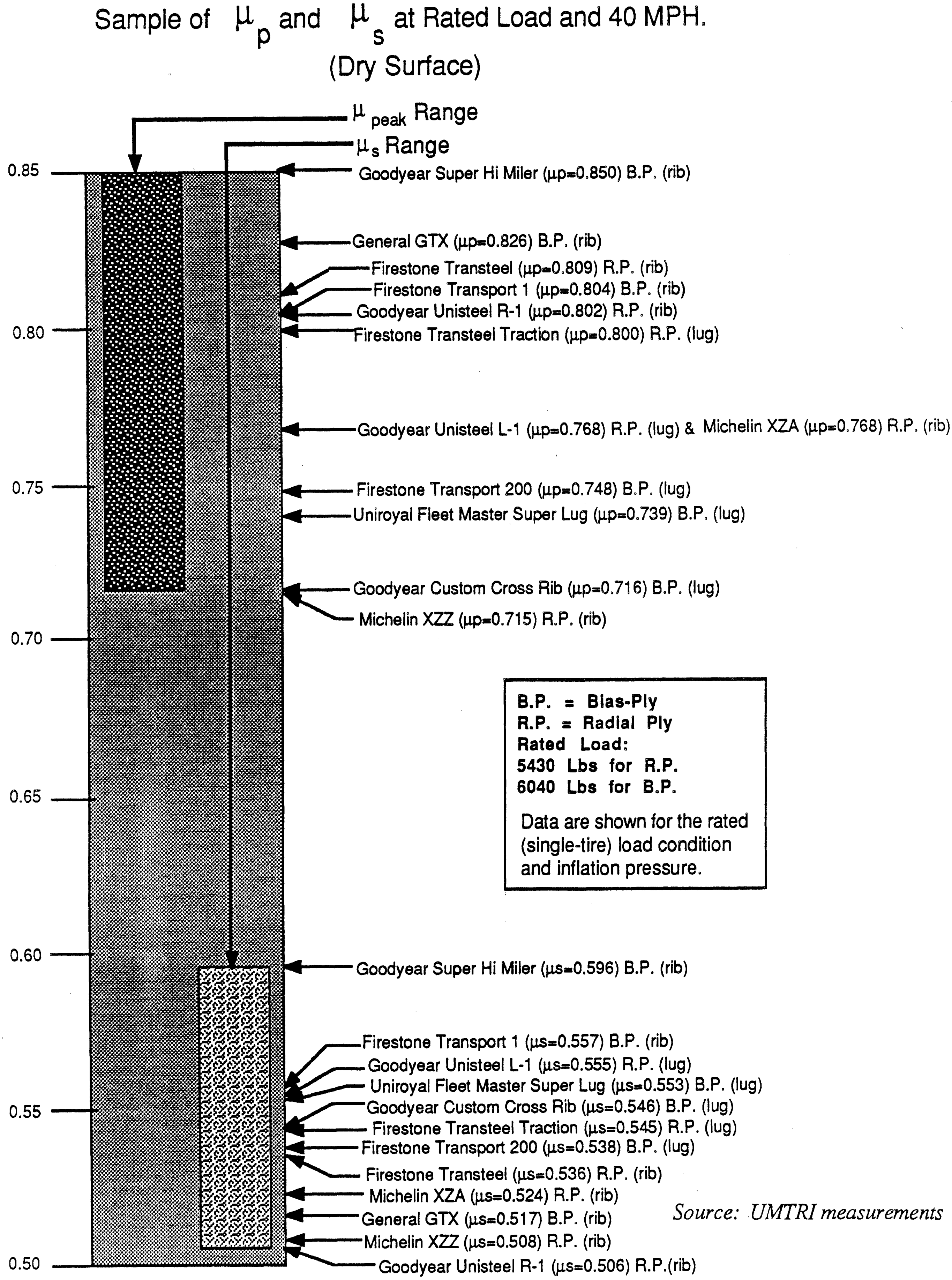


Figure 2.1.14 Peak and slide traction coefficient values

2.1.3.5 Vertical stiffness. The vertical spring rate of a sample of tires is shown in Figure 2.1.15. The data represent stiffness values in the vicinity of the rated load for tires which are rolling at a relatively slow speed.

• Values of vertical stiffness measured on tires in common highway service range from 4,400 to 5,800 lb/inch.

• A sample of bias-ply tires is seen to cover the entire range of measured values. A typical vertical stiffness value for new bias-ply tires is 5,000 lb/in.

• Available data on radial-ply tires occupy the lower end of the range of reported values, with a typical number for new radials of 4,600 lb/in.

• A modest increase in vertical stiffness, on the order of 2 percent, is seen to accompany treadwear with one radial sample.

Inflation pressure obviously has a strong effect upon the vertical stiffness of any tire. The vertical stiffness of a given tire will derive from a more-or-less constant value associated with the inherent stiffness of the carcass and tread structures plus that due to inflation. Thus, although the vertical stiffness is not directly proportional to inflation pressure, it is rather nearly so in the vicinity of the inflation pressure recommended for rated load.

The rolling velocity of the tire is also known to influence the vertical stiffness property to a mild degree.

2.1.4 Methods for Measuring or Estimating Tire Properties. Because the tire's overall force and moment response involves such a complex process of deflection of the tire structure, there is no general means of estimating the pertinent mechanical properties without experimental measurement. Further, the measurement of all tire properties of interest requires that the tire be rolling and that it be mounted on a force- and moment-measuring device. Thus, tire properties are generally obtained only through the use of specialized apparatuses. Both laboratory and mobile devices have been developed for making such measurements.

The laboratory devices are typically devoted to measuring the stiffness characteristics which do not require an authentic friction interface with the tire. Both circular drum-type facilities and flat-surface test machines have been employed in the laboratory. The flat-surface devices are preferred to the degree that the distribution of vertical pressures in the tire contact patch are distorted on curved surfaces and thus tend to distort such measures as cornering stiffness and pneumatic trail. A type of flat-surface device which has been broadly developed in recent years

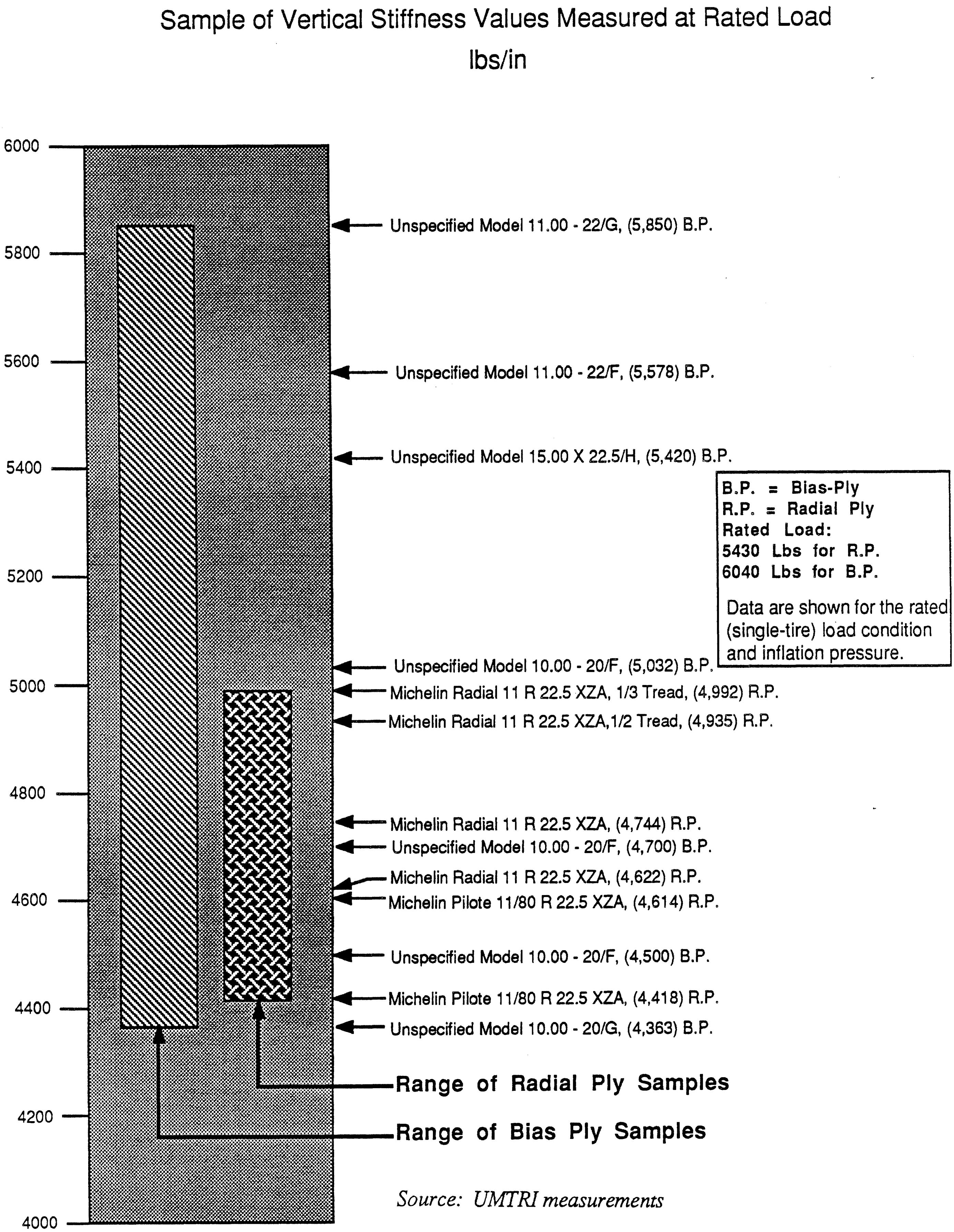


Figure 2.1.15 Vertical stiffness values

employs a steel belt which supports the tire over a flat fluid bearing. The belt can be run at highway speeds and can be operated with a water film to approximate wet-pavement conditions.

Mobile devices are typically devoted to measuring friction-limited properties of the tire under authentic pavement and surface contamination conditions. A heavy test rig is outfitted with a dynamometer for imposing the desired lateral and longitudinal slip conditions and for measuring force and moment responses. Surfaces can be pre-wetted by means of sprinkling systems or can be watered by means of on-board pump units. Together, laboratory and mobile tire test machines can provide the types of raw data from which the pertinent mechanical properties are derived.

2.2 Suspensions

2.2.1 Pertinent Properties of Suspensions. Heavy-vehicle suspensions have a variety of practical performance requirements ranging from the basic ability to carry the load and enhance ride quality, to considerations of cost, weight, maintainability, and service life. In this document, however, interest is limited to suspension properties which influence vehicle dynamic performance, that is, the braking and directional performance properties of the vehicle. Recognizing that these performance areas are dominated by the forces and moments produced by the tire in contact with the ground, then it is clear that the importance of the suspension is embodied in the role which it plays in influencing the various tires of the vehicle.

1) Suspensions play an important role in determining the dynamic loading conditions of the tires .

2) Suspensions play an important role in orienting the tires with respect to both the road and the vehicle.

3) Suspensions also play an important role in influencing the motions of the vehicle body, relative to the axles, which, in turn, contributes to tire loadings and orientations, and to stability.

Although commercial vehicle suspensions come in a tremendous variety of shapes and sizes, with a wide variety of specific springs, linkages, and other hardware elements, every suspension type has several basic mechanical properties which determine how the suspension performs these three fundamental roles. In virtually all cases, the performance of the various hardware elements can be interpreted in terms of the following pertinent mechanical properties:

Composite Vertical Stiffness
Composite Roll Stiffness
Damping

Load Equalization
Interaxle Load Transfer
Roll Center Height
Roll Steer Coefficient
Compliance Steer Coefficients

In evaluating the dynamic performance qualities of any suspension, it is, therefore, important not to become overly involved in the specific details of all the various hardware elements of the suspension, but rather to be concerned about the values of these pertinent mechanical properties which result from the designers specific part selections and designs. The general relationship between the specific suspension parts and these pertinent mechanical properties is shown in Figures 2.2.1 through 2.2.3.

2.2.1.1 Composite vertical suspension stiffness. The most fundamental property of virtually all suspensions is vertical stiffness (or, conversely, vertical compliance) provided by the spring elements. When the suspension deflects vertically, all the springs deflect in unison and their individual stiffnesses sum to determine the composite vertical stiffness of the suspension:

$$K_V = F_Z/Z = \Sigma K_S$$

This equation shows that vertical stiffness (or spring rate, K_V) is defined as the vertical force (F_Z) required per unit of vertical deflection (Z) and is composed of the sum of the stiffnesses of all of the springs of the suspension (ΣK_S). (Additional vertical compliance is provided by tire deflection. See the section on tires.)

Most commercial vehicle suspensions use steel leaf springs. The next most common spring is the air spring. Other suspensions may use steel torsion bars or rubber elements to provide the spring action. The very wide range of loads carried by the suspension (from ladened to empty conditions) puts difficult demands on the suspension spring. The spring must be quite stiff to support the full load without undue deflections. This high stiffness may make the ride of the empty vehicle quite rough. Air springs can provide better ride over the full range of loads since their spring rate changes in response to the load being carried.

The leaf spring, as used in commercial vehicle suspensions, displays a complex force/displacement relationship which includes friction as well as stiffness qualities. Figure 2.2.4 shows the typical form of the force/deflection relationship. Local spring stiffness depends on the value of load and the length of displacement, as does the level of Coulomb friction. Some springs show an overall stiffening with increasing load. Springs mounted with "slippers" usually display

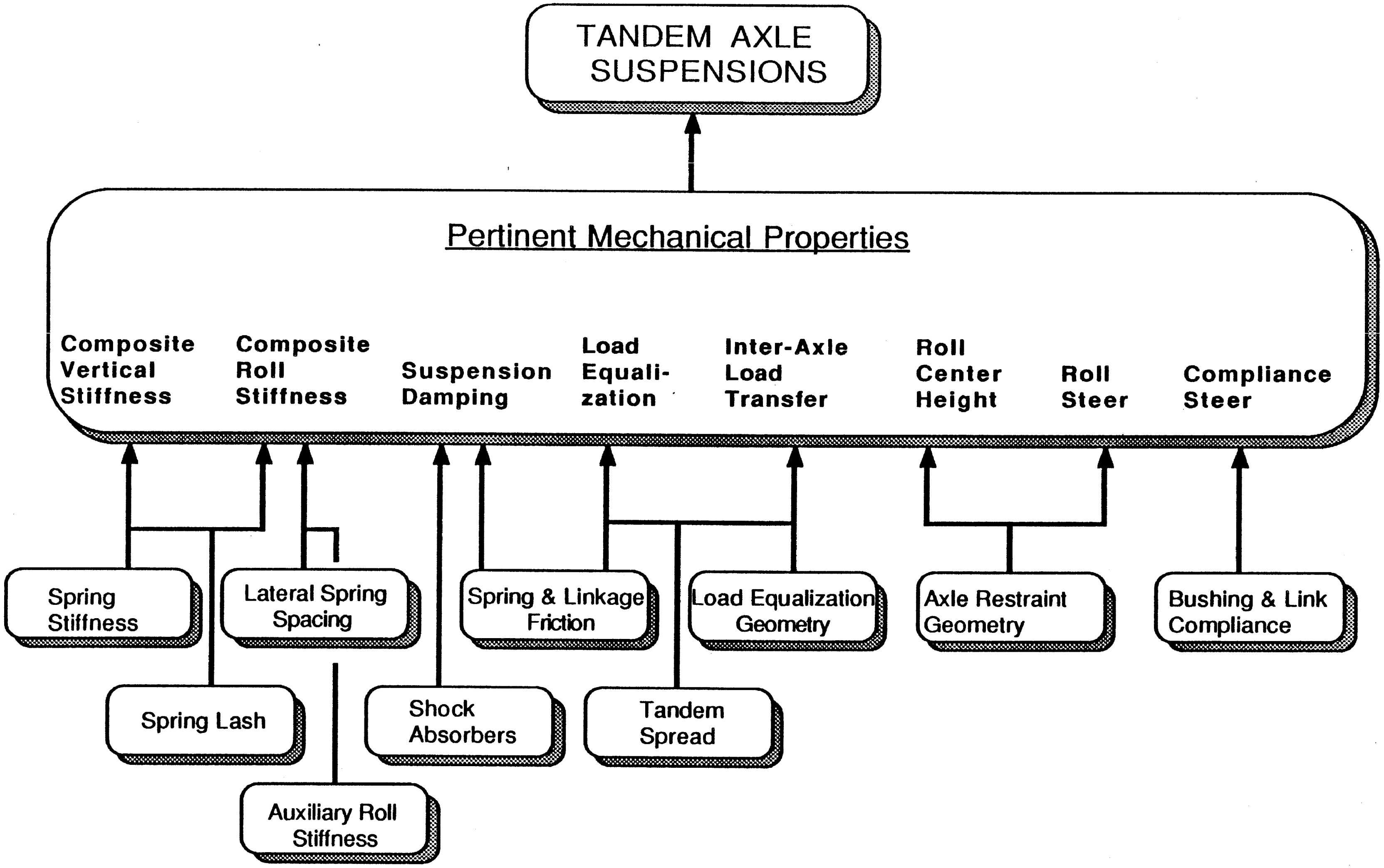
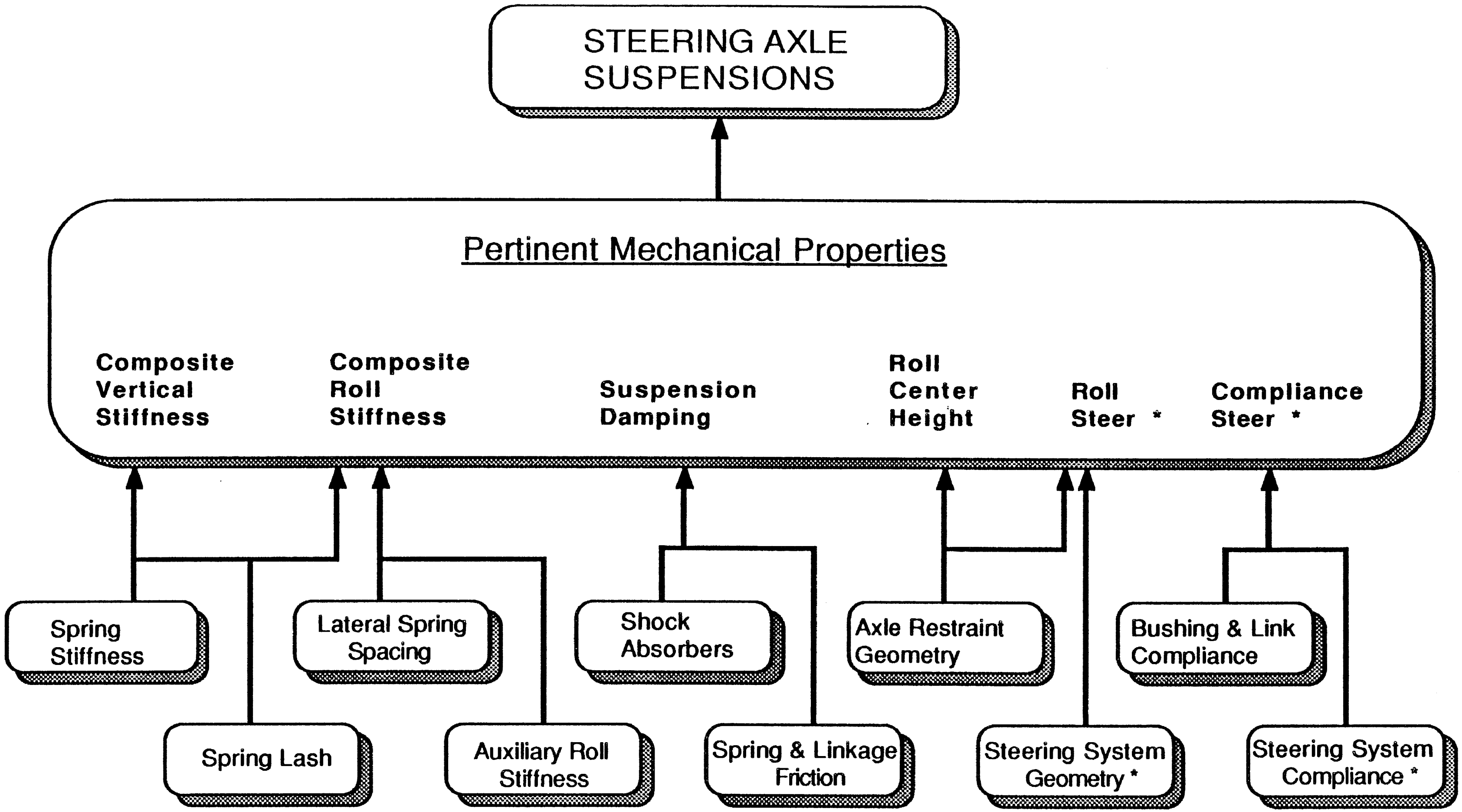


Figure 2.2.1 Pertinent mechanical properties of tandem axle suspensions



* SEE "Steering System"

Figure 2.2.2 Pertinent mechanical properties of steering axle suspensions

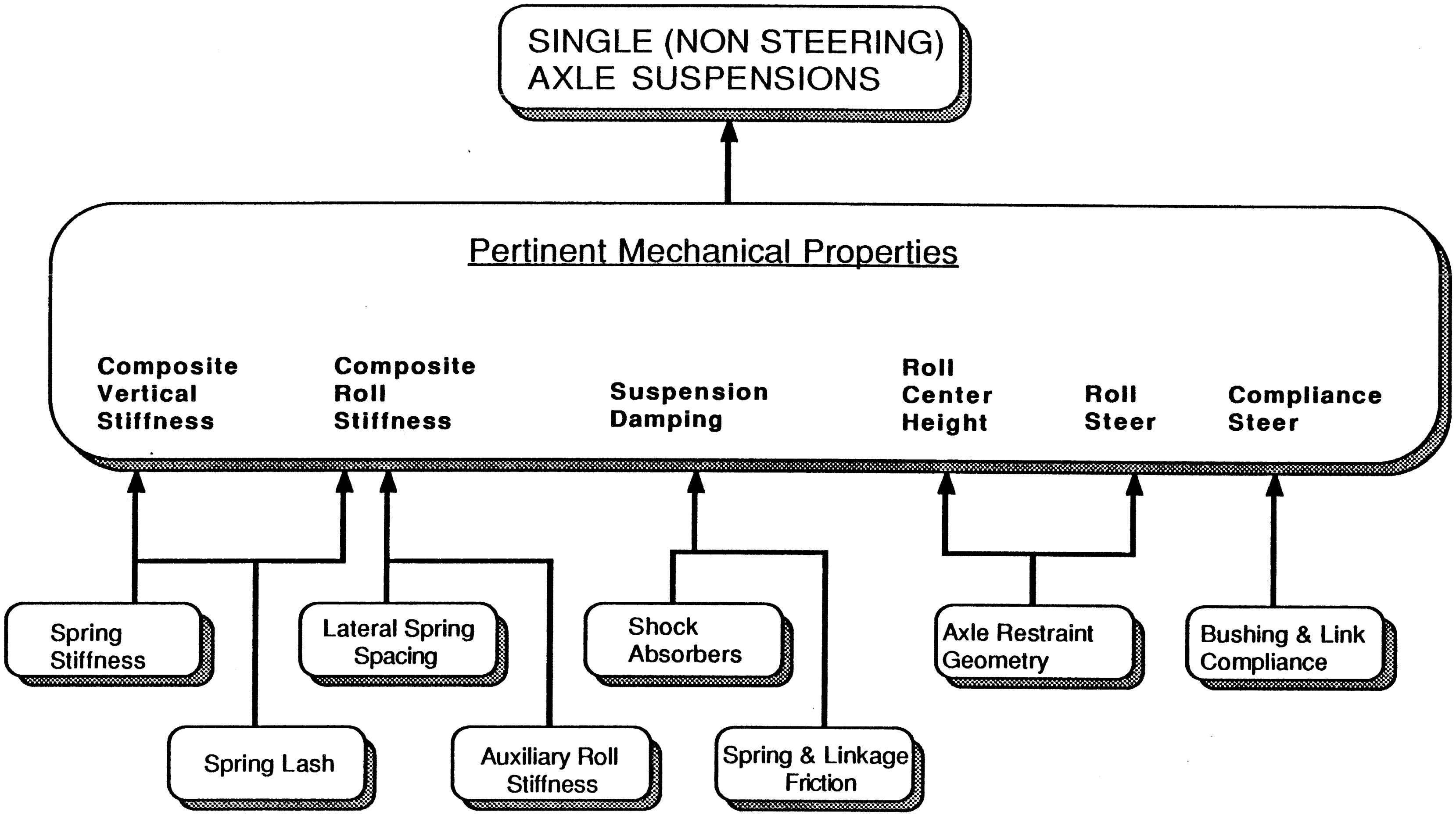
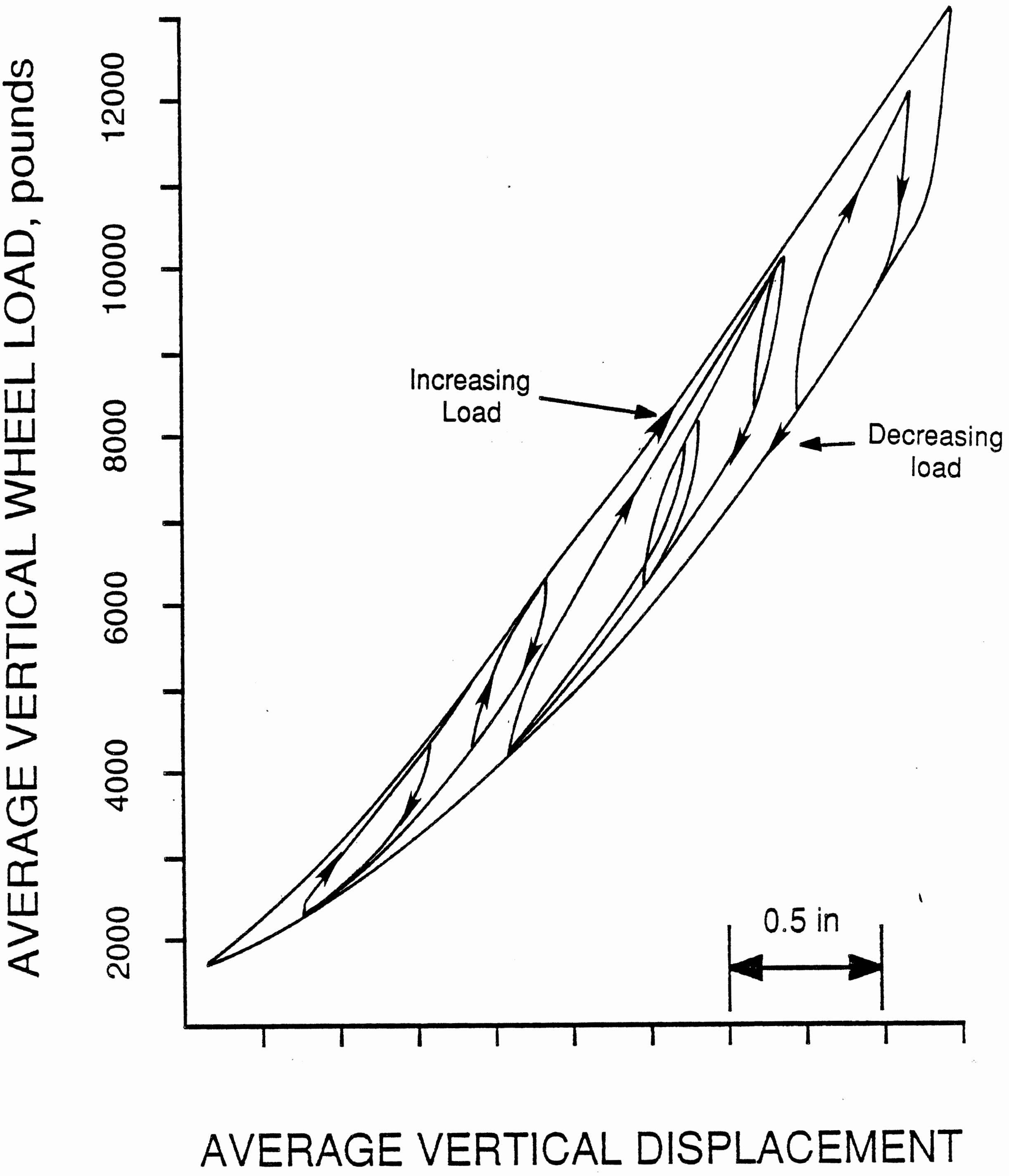


Figure 2.2.3 Pertinent mechanical properties of single (non steering) axle suspensions



Source: UMTRI measurements

Figure 2.2.4 Average vertical displacement vs. average vertical wheel load lash when passing from tension to compression. Springs on the "light side" of the vehicle can operate in the lash area during extreme maneuvers which approach the rollover limit.

Figure 2.2.5 shows typical air suspension spring behavior. The stiffness is strongly dependent on vertical load (actually, on the initial air pressure in the spring), so that behavior is shown separately for different nominal loads. Air springs are sufficiently soft so that the compression and tension bump stops may come into play in severe turning or braking maneuvers. (Steering-axle leaf-spring suspensions may also be soft enough to make bump stop limits important in heavy braking. Non-steering leaf-spring suspensions are generally so stiff that bump stops are not a concern.)

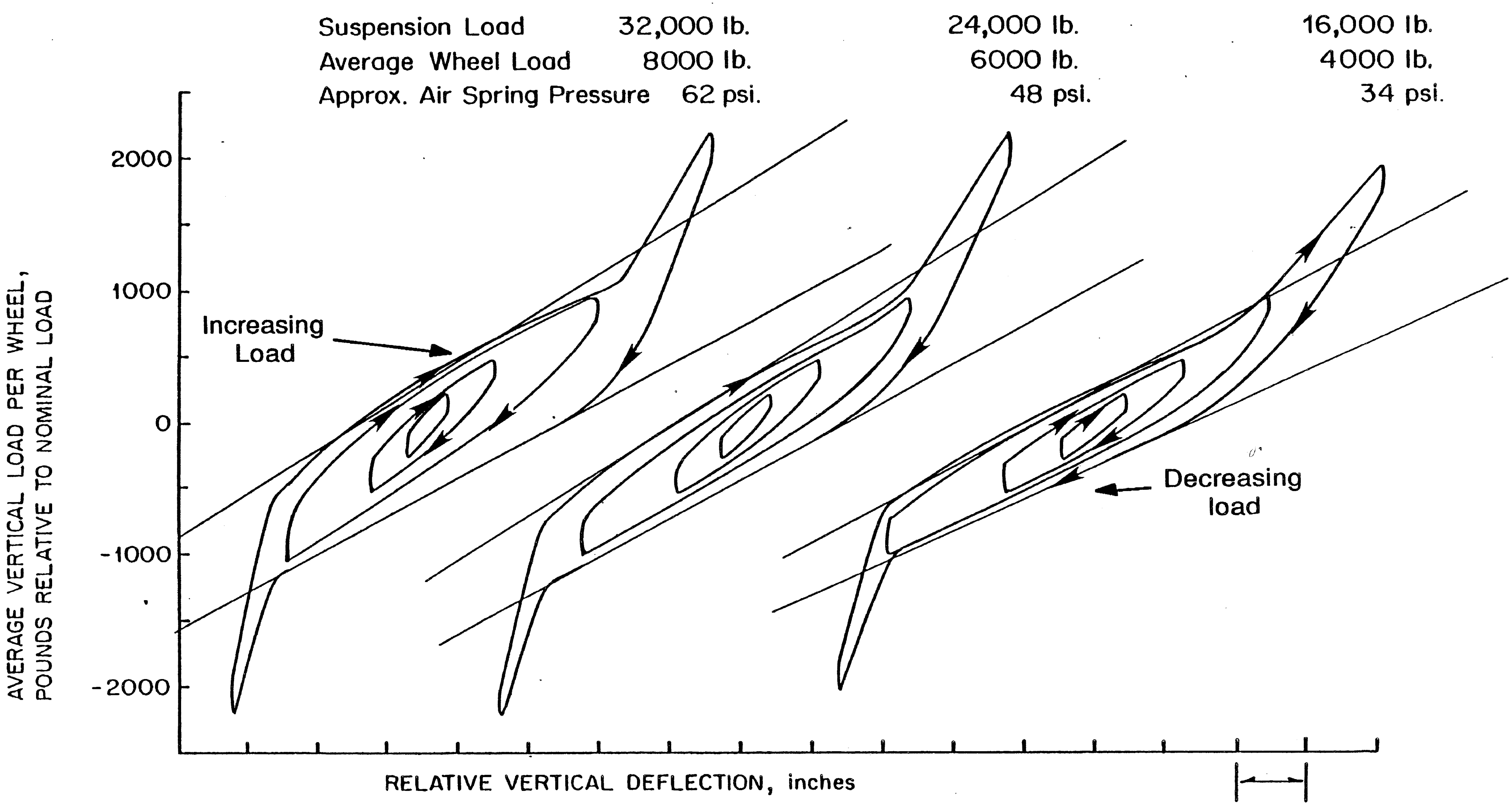
2.2.1.2 Composite roll stiffness. When the vehicle rolls, springs on either side of the vehicle deflect in opposite directions, and as shown in Figure 2.2.6, the spring forces produce a restoring roll moment. The relationship between suspension roll angle and restoring moment is known as the roll stiffness of the suspension. Composite roll stiffness is a function of the individual spring rates and the lateral spring spacing plus any auxiliary roll stiffnesses:

$$K_r = M_x/\phi_s = 2 \times K_s (T_s/2)^2 + K_{aux}$$

That is, roll stiffness (K_r) is roll moment (M_x) per degree of suspension roll (ϕ_s) and derives from the spring stiffness (K_s) times the square of one half of the lateral spring spacing (T_s), plus any auxiliary roll stiffness (K_{aux}).

Auxiliary roll stiffness comes from mechanisms which provide roll stiffness without being involved in vertical stiffness. Auxiliary roll stiffness is commonly provided on cars by using an "anti-sway bar." Some European truck suspensions, as well as some U.S. air suspensions use anti-sway bars. Air suspensions usually have some auxiliary roll stiffness device. Often the trailing arm is rigidly clamped to the axle so that the whole assembly acts as an anti-sway bar. Other air suspensions have a cross member between the trailing arms to provide auxiliary roll stiffness. Even steel spring suspensions usually have a small amount of auxiliary stiffness provided by the fact that the springs must be twisted along their length in order for the suspension to roll.

2.2.1.3 Suspension damping. Suspension damping derives from two major sources, viscous friction from the shock absorber action, and Coulomb friction from the leaf spring and linkage actions. Figure 2.2.4 showed the Coulomb friction property of leaf springs. Typically, interleaf friction in leaf-spring suspensions is so large that additional damping of shock absorbers is not required. Since Coulomb friction damping provides poor ride quality, special effort is often



Source: UMTRI measurements

Figure 2.2.5 Relative vertical deflection vs. average vertical load per wheel

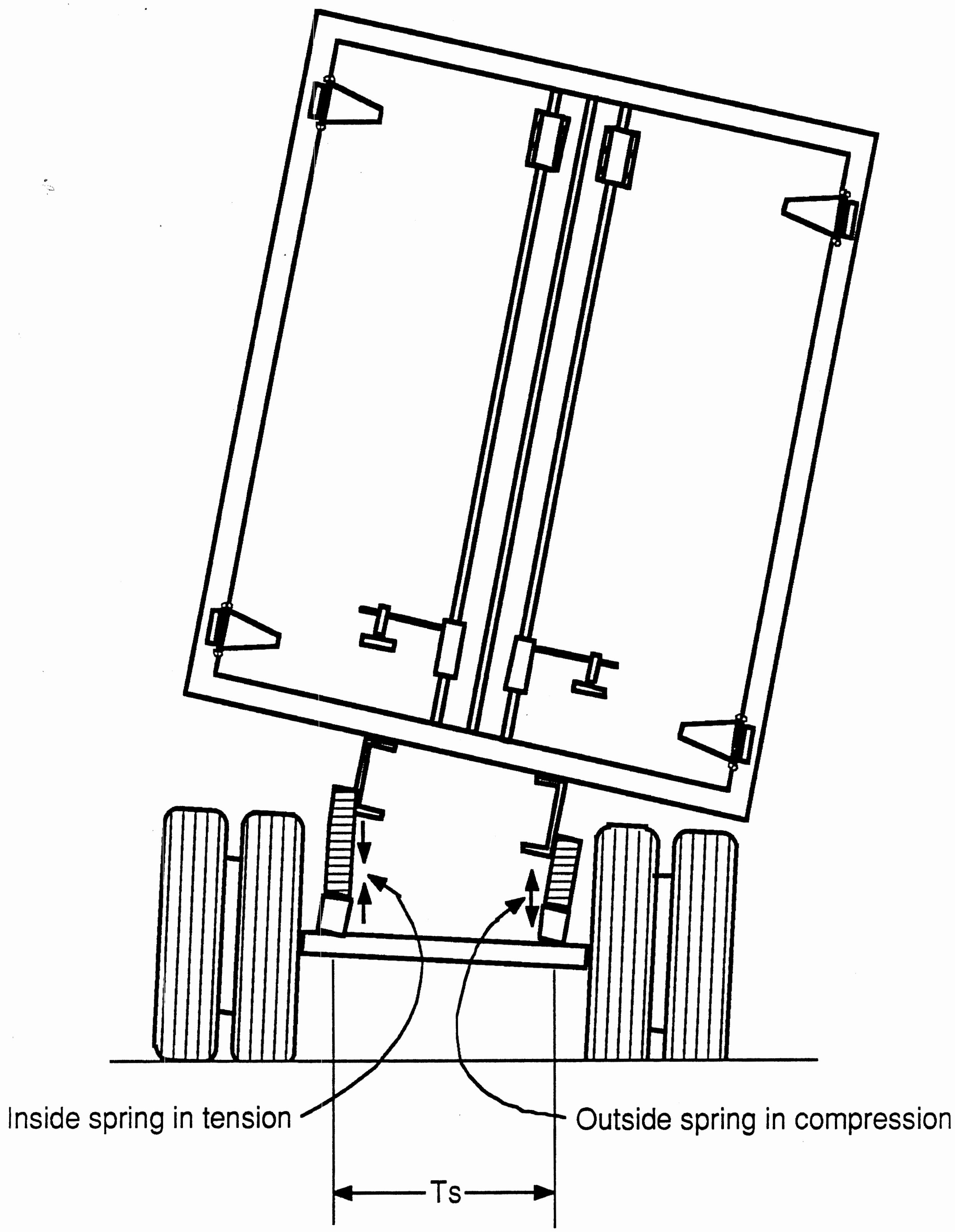


Figure 2.2.6 Spring displacement in roll

made to reduce front spring friction, such as using tapered springs with anti-friction, interleaf inserts. As a result, shock absorbers are often added to front suspensions. Air springs provide little friction so that air suspensions usually have shock absorbers.

2.2.1.4 Load equalization. In order to carry very large loads, commercial vehicles are often equipped with multi-axle suspensions. And to avoid excessive loading of frame and/or suspension members when traversing uneven road surfaces, these axles are often inter-connected with a mechanism intended to maintain equal loading between the axles. The two-axle "tandem suspension" is particularly common among non-steering suspensions.

The most common tandem suspension types are the "four-spring" and the "walking beam." Figure 2.2.7 shows the "load-leveler" mechanism typical of the four-spring suspension. Figure 2.2.8 shows a typical walking-beam suspension. Of course, four-spring suspensions always use leaf springs. Walking beams may use leaf springs, rubber blocks, or sometimes, no spring at all. In a "two-spring" suspension, leaf springs, fastened to the axles at each end and pivoted at the center, much like a walking beam, provide both the spring and load balance functions. Parallel plumbing of the air springs on adjacent air-suspended axles is another way in which load equalization is acheived in a tandem suspension.

The load equalization quality of tandem suspensions is influenced by the geometry of the equalization mechanism as well as by Coulomb friction present in the linkages and/or springs. In most cases, the mechanisms are symmetric, or nearly so, such that very good equalization is expected. In some suspensions (especially the four-spring type), friction may cause the mechanism to "hang-up" such that fairly high imbalances may be measured statically. On the other hand, the absence of friction in the load equalization mechanism, as in the walking-beam suspension, means the system may be poorly damped dynamically when the vehicle is traveling at speed. This low damping may result in suspension oscillations known as tandem axle tramp, chatter, or hop which can produce very high, dynamic axle loads.

2.2.1.5 Interaxle load transfer. The same mechanisms intended to provide load equalization between axles of a suspension during normal travel may serve to produce unequal axle loads during periods of braking and/or acceleration. Many tandem suspensions produce interaxle load transfer between the axles of a tandem suspension as a result of the application of braking or driving torques. Among the common tandem suspensions, the four-spring type is most susceptible. During braking, many four-spring suspensions will transfer significant loading from the lead axle to the trailing axle. Walking-beam suspensions generally transfer less load due to braking, and load transfer is in the opposite direction. Other things being equal, interaxle load

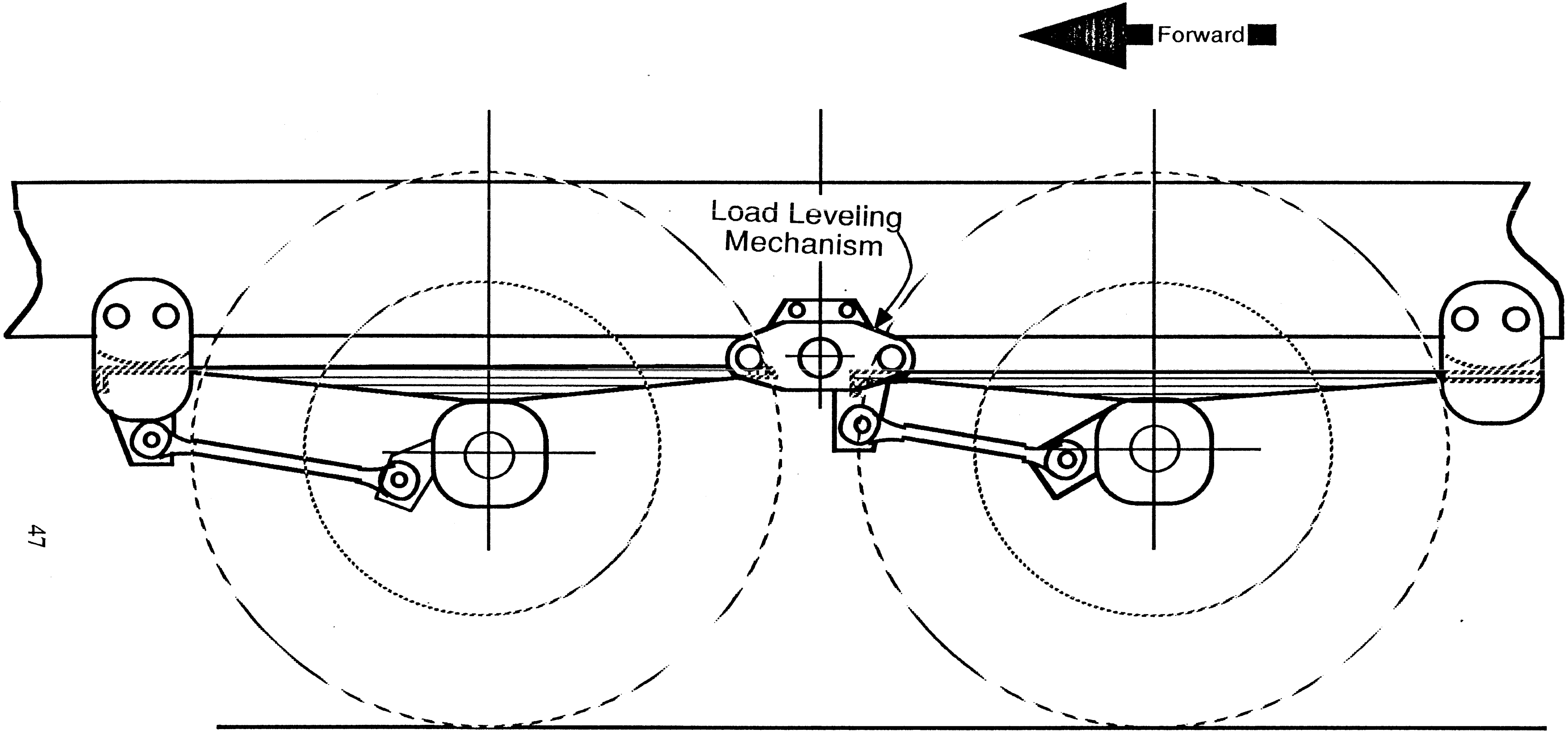


Figure 2.2.7 Four-spring suspension

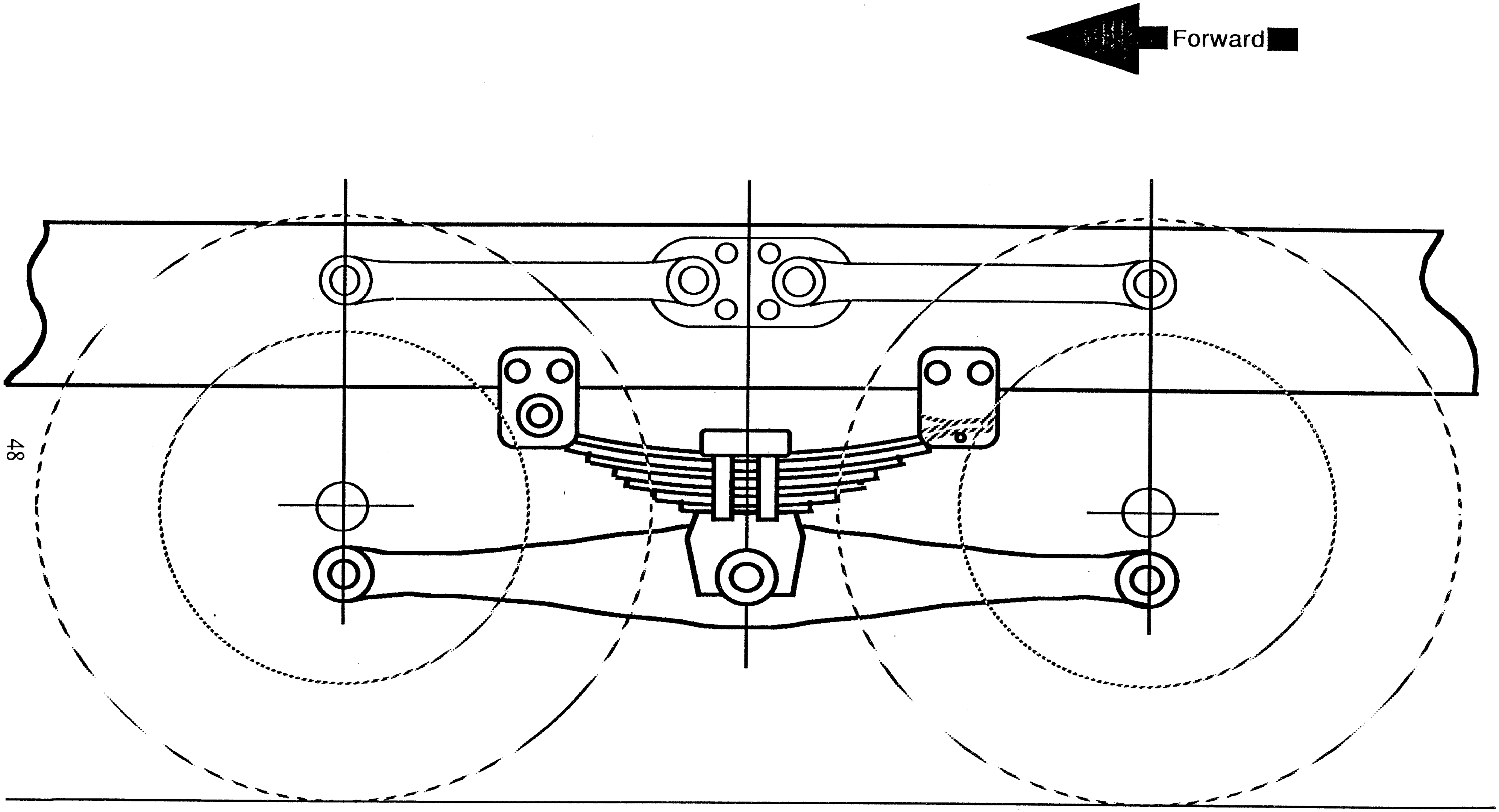


Figure 2.2.8 Walking beam suspension

transfer will generally be less for larger tandem spreads, but not necessarily. Tandem air suspensions generally produce significant interaxle load transfer only if the suspension linkages of the two axles are different.

2.2.1.6 Roll center height. When a vehicle body rolls on its suspensions during a turning maneuver, the relative roll motion of any axle with respect to the body can be pictured to occur about some specific point, as shown in Figure 2.2.9. That is, during rolling motions, there is some point fixed in the axle, which appears to also stay fixed, except for rotation, in the body. This point is called the axle, or suspension, roll center, and its location depends on the details of the suspension parts which locate the axle (laterally). In fact, the roll center is generally located on the vehicle centerline at a height above the ground where lateral forces are passed from the suspension to the chassis. Indeed, the importance of the roll center concept lies simply in the fact that the roll center locates the line of action of lateral suspension forces. In most four-spring and single-axle leaf-spring suspensions, it is the leaf spring itself which locates the axle laterally. In walking-beam and air suspensions, special lateral links may be added which provide the primary lateral restraint.

2.2.1.7 Roll steer coefficient. When a vehicle rolls on its suspension during turning maneuvers, the wheels of the vehicle steer slightly as a result of the rolling motion. This is true even for wheels of the so-called non-steering axles. As seen in Figure 2.2.10, when a spring deflects and the axle moves "up and down" relative to the body, the axle motion is not generally pure vertical motion. Actually, because of the layout of links or other parts that restrain the axle in the fore/aft direction, the axle moves in an arc about a center which is, in concept, very much like the roll center. Motion on the arc means that the axle moves slightly in the fore/aft direction as it moves up and down. When the vehicle rolls, one end of the axle moves up while the other moves down, and as a result, one end moves slightly forward as the other moves slightly aft. That is to say, as the vehicle rolls, the axle steers slightly. The relationship between the amount of axle steer which occurs per degree of suspension roll is known as the roll steer coefficient. (Steering axles also display roll steer properties, but these will be considered under the steering system discussion.) Although the steer angles which occur as a result of roll steer are small, they are important to vehicle handling behavior.

2.2.1.8 Compliance steer coefficients. The wheels of a vehicle also steer slightly as a result of deflections within the suspensions. Braking forces, side forces, and tire aligning moments generated at the tire/road interface all produce substantial forces which must be carried by suspension linkages and other components. As a result of the application of these forces, rubber bushings, and even steel and brass members, can deflect sufficiently to produce small, but

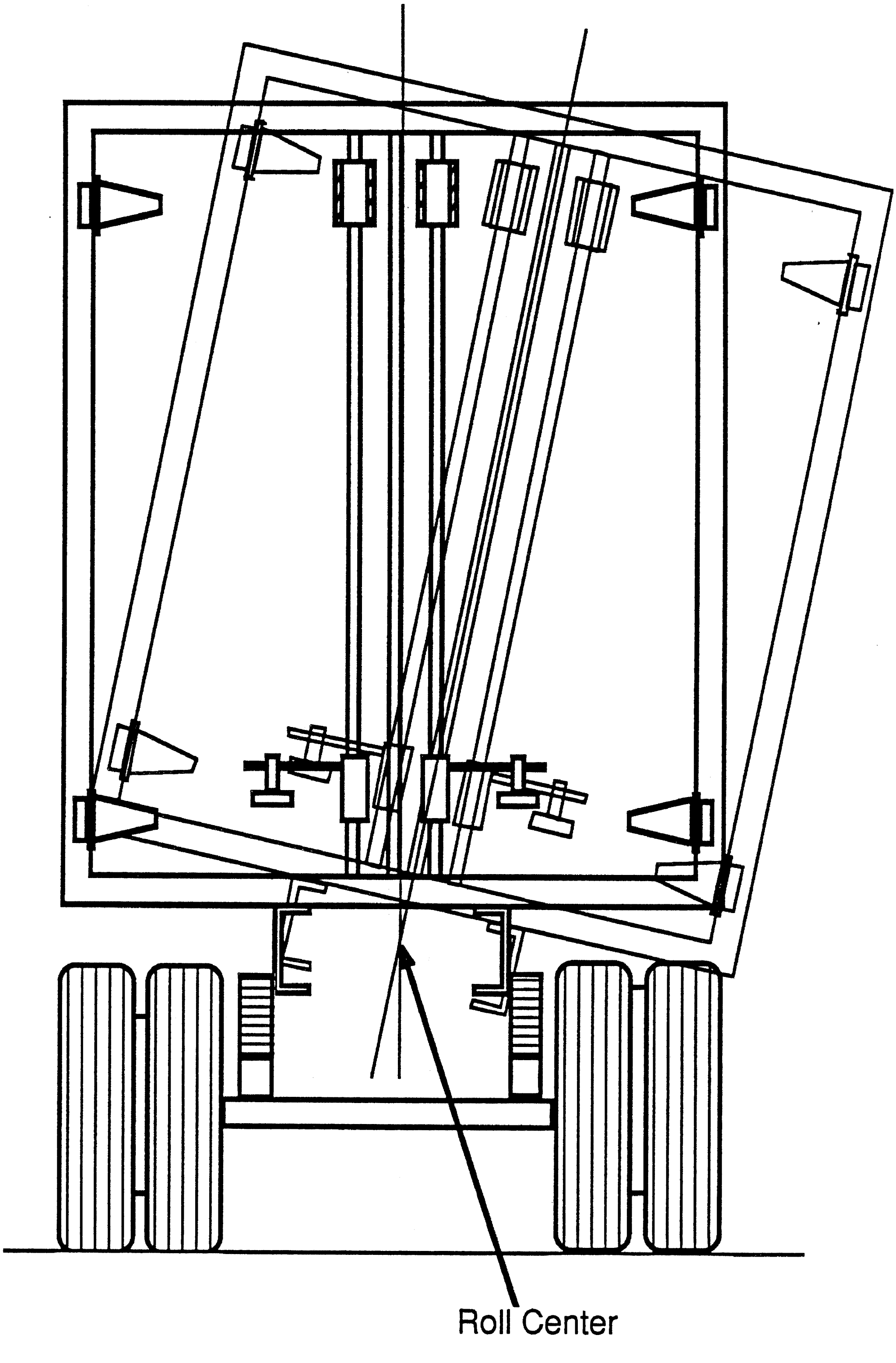


Figure 2.2.9 Illustration of roll center

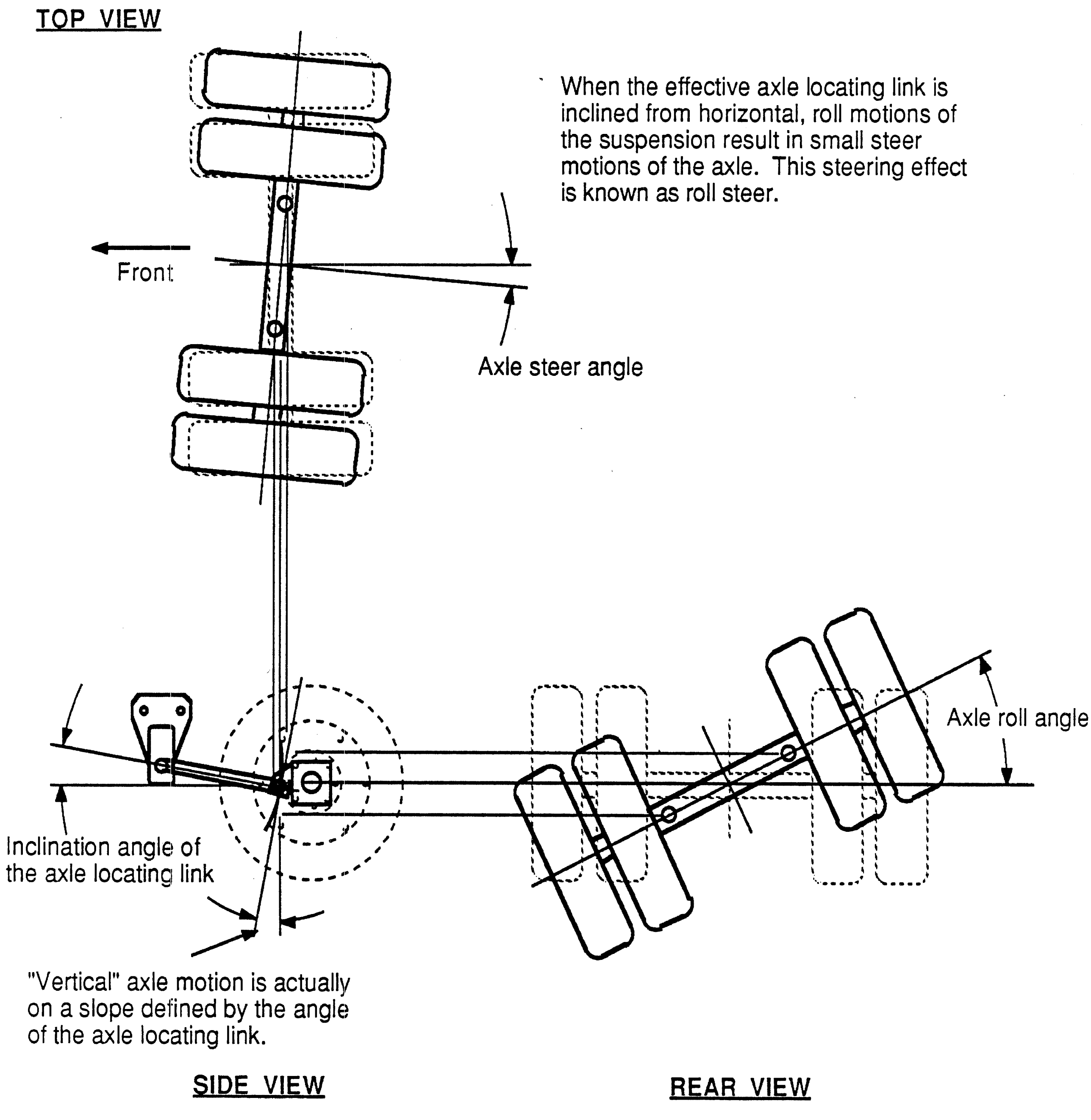


Figure 2.2.10 Illustration of roll steer

important, steer angle deflections of the axles. (Again, the same is true of steering axle suspensions as well as fixed axle suspensions. Steering axle matters will be discussed in the steering system section.) The steering reactions to brake force, side force, and aligning moments are known as compliance steer and the amount of steer per unit of force or moment is known as the compliance steer coefficient with respect to that force or moment. Non-steering axles may steer in response to lateral forces and aligning moments. Steering axles typically steer in response to these, plus brake force.

2.2.2 The Importance of the Pertinent Mechanical Properties of Suspensions to Vehicle Performance. As pointed out at the beginning of Section 2.2.1, suspension performance is important primarily through its effect on tire loading and orientation. Specifically:

1) Suspensions are important in determining the dynamic loading conditions of the tires.

2) Suspensions are important in orienting the tires with respect to both the road and the vehicle.

3) Suspensions also are important in influencing the motions of the vehicle body, relative to the axles, which, in turn, contributes to tire loadings and orientations, and to stability.

Tire loading and orientation, and body motions, in turn, have very basic influences on vehicle dynamic performance. As described in the first section of this Factbook, vehicle dynamic behavior can be described or evaluated through several basic performance measures. This section will indicate the importance of the pertinent mechanical properties of suspensions with respect to each of the vehicle dynamics measures and will briefly explain the mechanisms by which this importance arises.

Table 2.2.1 presents a summary of the level of importance which each pertinent mechanical property of suspensions has with respect to each of the performance measures. The following paragraphs provide some background understanding to the relationships shown in the table.

2.2.2.1 Vertical stiffness. Vertical stiffness is obviously an important suspension property. Vertical stiffness is the single most important property influencing ride quality, and in that sense, is "what a suspension is all about." Nevertheless, as Table 2.2.1 shows, vertical suspension stiffness does not have a major influence on any of the important vehicle maneuvering qualities. Indeed, vertical stiffness has only a moderately important influence on transient braking. This influence comes about through the suspension's deflection as load is transferred from rear to front axles and the vehicle pitches forward during braking. Vertical stiffness influences the amount

Table 2.2.1

The Importance of the Pertinent Mechanical Properties of Suspensions on Vehicle Dynamic Performance

Pertinent Mechanical Property of Suspensions	Low Speed Tracking	Hi Speed Tracking	Roll Stability	Yaw Stability	Response Time	Rearward Amp	Braking Efficiency	Transient Braking	Downhill Braking	Response to Disturbances
Vertical Stiffness	-	-	-	-	-	-	-	Med	-	-
Roll Stiffness	-	Med	Hi	Hi	Hi	Hi	-	-	-	Med
Roll Center Height	-	Med	Hi	Hi	Hi	Hi	-	-	-	Med
Damping	-	-	-	-	Med	Med	-	Low	-	Low
Roll Steer	-	Low	-	Low	Low	Low	-	-	-	Low
Compliance Steer	-	Low	-	Low	Low	Low	-	-	-	Low
Interaxle Load Transfer-	-	-	-	-	-	-	Hi	-	-	-

of pitch and the response time of the pitching motion. Similar pitching motions may affect anti-lock braking performance. Stiffer suspensions will decrease the amount of pitching and quicken the response time. These matters are of rather minor importance, overall, so that stiffening suspensions to the degradation of ride quality for these reasons is not warranted.

2.2.2.2 Roll stiffness and roll center height. Roll stiffness and roll center height are, arguably, the most important suspension properties, since they have a major influence on all directional performance properties of the vehicle other than tracking. They are considered together here, since they always act together in determining the roll response of the vehicle in turning maneuvers.

Figure 2.2.11 shows the rear view of a commercial vehicle in a turn. Rather than showing a specific, "real" suspension, the figure describes "all" suspensions by illustrating the suspension in terms of its pertinent mechanical properties, roll center, and roll stiffness. (In this representation, the ability of the suspension to deflect vertically is ignored since it is of no importance to the performance matters being considered. Also, this representation does not distinguish among all the suspensions along the length of the vehicle. Rather, it "lumps" the properties of all the suspensions. Later, the influence of the distribution of suspension properties will be considered.)

In a turn, the vehicle experiences a centrifugal force through its center of gravity (c.g.) which is proportional to the lateral acceleration (a_y) and the weight of the vehicle (W). This force has a "moment arm" with respect to the tire springs equal to the height of the c.g. above the ground (h_{cg}), and a moment arm with respect to the suspension roll stiffness equal to the height of the c.g. above the roll center (h_{cg}-h_{rc}). The moments applied to the suspension springs and the tires cause the axle and body to roll outboard. These rolling actions cause an outboard shift of the c.g. with respect to the suspension (S_{sus}) and with respect to the tires (S_t), which increase the rolling moments by giving the weight of the vehicle a moment arm about the suspension and tire roll stiffnesses.

The amount of body roll on the suspension depends on roll stiffness and roll center height. If the suspension has very high roll stiffness, the body will not roll much on the suspension due to this high stiffness, or if the roll center is very high (near the c.g.) the body will not roll much since the moment arm of the centrifugal force will be small. In equation form:

$$f_s = [a_y W (h_{cg} - h_{rc})] / [K_f - W (h_{cg} - h_{rc})]$$

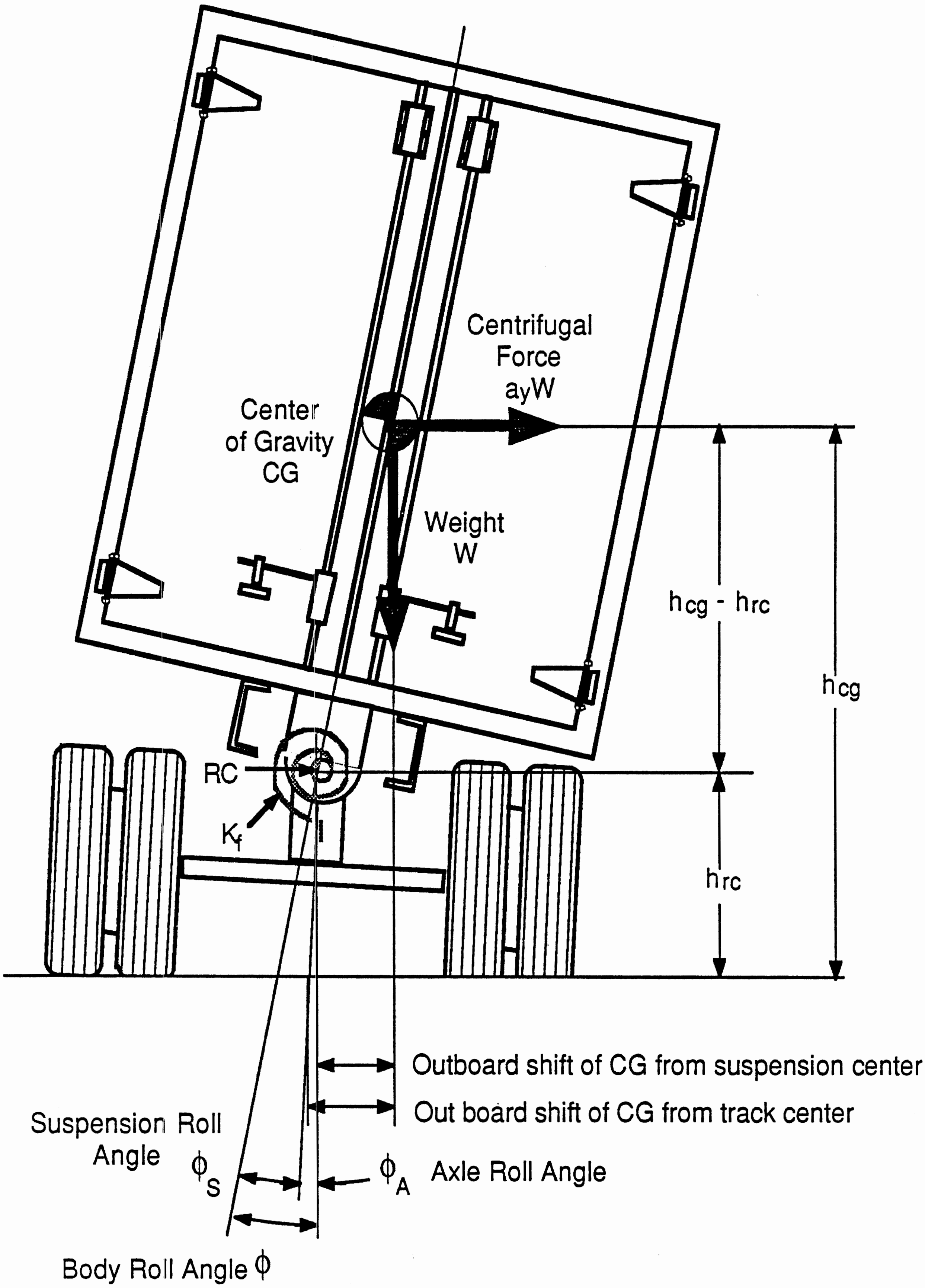


Figure 2.2.11 Illustration of a commercial vehicle in a turn

The whole rolling behavior of the vehicle causes vertical load to be transferred from tires toward the inside of the turn to tires toward the outside of the turn. More precisely, the roll moment of the centrifugal force and the total outboard shift of the c.g. is reacted by a moment produced by this outboard shift of load. The limit of the ability of the vehicle to react to roll moment is defined by the condition when the inboard tire load drops to zero, that is, when the inboard tires lift off the ground. Since a very significant portion of the roll moment can be due to the outboard shift of the c.g., the suspension roll stiffness and roll center height are important in determining the rollover stability limit of the vehicle. *High roll stiffness and high roll center heights serve to improve roll stability.*

As mentioned above, Figure 2.2.11 "lumped" the properties of all suspensions into one in order to simplify the discussion. The total roll stiffness is most important, but the distribution of roll stiffness among the suspensions of the vehicle also has an influence on roll stability. Although the explanation is more complicated than appropriate, here, as regards roll stability, the best situation is for roll stiffness to be distributed among the suspensions in proportion to the load carried by the suspensions. In practice, this is very hard to acheive since steering-axle springs are usually relatively soft to provide good ride, and auxiliary roll stiffening is not commonly added in the U.S. Also, tractor rear suspensions are usually somewhat softer than trailer suspensions, even though they carry about the same load in general. In "correcting" such situations, advantage is gained only by stiffening the relatively soft suspension, not by softening stiffer suspensions. In this regard, a suspension can be considered effectively "stiffer" if its roll center is higher.

The distribution of suspension roll stiffness also has a major influence on the distribution of tire vertical loads during cornering. The previous discussion, based on Figure 2.2.11, explained the general manner in which vertical load is transferred across an axle from inside tires to outside tires during cornering. The distribution of roll stiffness among the suspensions of the vehicle determines how this side-to-side load transfer is distributed among axles. The tires of the relatively stiff suspensions will experience higher load transfer while tires on suspensions with low roll stiffness will experience little side-to-side load transfer. The tire section of this Factbook explains that the aggregate of the pertinent mechanical properties of the tires on one axle generally degrades as the axle experiences side-to-side load transfer. Virtually all turning behavior of the vehicle (at speed) is very strongly influenced by the balance of these tire properties among the various axles of the vehicle. Therefore, the distribution of roll stiffness and roll center heights among suspensions, through their influence on the balance of tire properties, is highly influential with respect to yaw stability, handling response times, and rearward amplification. These suspension properties also have a moderate influence on tracking and the response to external disturbances through the same tire influence mechanism.

Finally, roll stiffness and roll center height have an indirect influence on all turning behavior at speed, including tracking, through the influence on the amount of suspension roll, which, in turn, determines the amount of roll steer.

2.2.2.3 Damping. Damping is the property of any mechanical system which tends to slow down or stop motion. Accordingly, suspension damping is involved only in determining transient vehicle performance measures, not steady-state measures. Damping helps determine how rapidly steady state is achieved, for example, how rapidly a roll oscillation will die out. Accordingly, the level of damping has a moderate influence on the response time and rearward amplification measures of Table 2.2.1.

2.2.2.4 Roll steer and compliance steer. Roll steer and compliance steer behaviors of suspensions generate small steer angles at the tires of "non-steering" suspensions in response to body roll motions and to tire forces and moments, respectively. (See the steering system discussion for steering suspension influences.) Depending on the direction of steer which results, roll and compliance steers have virtually the same effect as would increasing or decreasing the cornering stiffness of the tires on the axle involved. In concept, for the same reasons that tire cornering stiffness is important, these steer influences are very important to determining the turning performance of commercial vehicles. In practice, most commercial vehicle, non-steering suspensions show moderate values of roll steer and rather small compliance steer coefficients. Therefore, these factors do not generally have a strong effect. (An exception to this rule would be trailing-arm suspensions which do not have special links to control lateral axle motions, e.g., some air suspensions. Such suspensions may have large levels of compliance steer.)

2.2.2.5 Load equalization and interaxle load transfer. Load equalization plays a major roll in determining the static load distributions on the tires of a tandem suspension. Interaxle load transfer is important in determining the dynamic load distribution on the same tires during braking. Through their influence on tire loading, these properties greatly influence braking efficiency.

Generally, the brakes on both axles on a tandem suspension are identical so that, if axle loads were equal, wheel lockup would occur on both axles at the same brake line pressure and deceleration level. If, because of poor load equalization and/or interaxle load transfer due to braking, the two axles of the suspensions operate at unequal loads, the lightly loaded axle will experience "premature" lockup at a lower pressure and deceleration, while the wheels of the more heavily loaded axle will continue rolling at somewhat higher braking efforts and deceleration levels. Assuming the usual case, that the tandem suspension constitutes the rear suspension of either a truck, tractor, or trailer, lockup of the first axle of the tandem substantially reduces the directional stability of the unit, and lockup of the second tandem axle renders the unit completely unstable. Generally, it is felt that the loss of stability from any lockup is undesirable, such that equal axle loading is desirable. It can be argued, however, that the "delayed" lockup of the second axle due to unequal tandem axle loading can maintain some level of stability to higher braking levels.

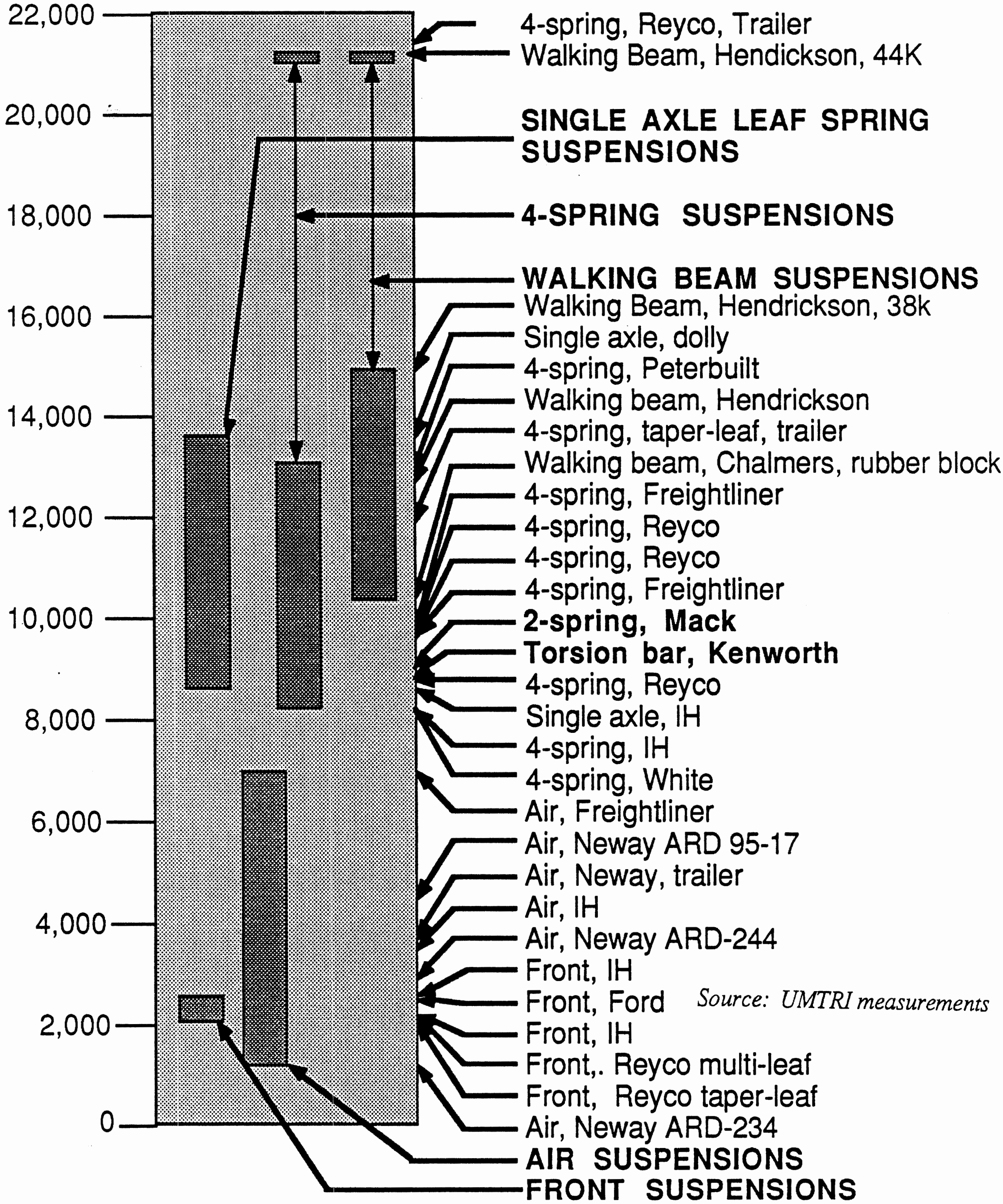
2.2.3 Ranges of Values of the Pertinent Mechanical Properties of Suspensions. This section will review briefly what is known concerning the range of values of the pertinent mechanical properties of suspensions in common use. The values of these properties are not commonly available, so that virtually all of the data presented in this section derive from specialized laboratory measurements made on a selection of heavy vehicle suspensions. For the more important properties, graphical presentations, providing easy comparison of suspension types, will be shown.

2.2.3.1 Composite vertical stiffness. As noted earlier, composite vertical stiffness is not of great importance to the vehicle dynamics properties of concern in this Factbook. Nevertheless, since vertical stiffness is such a basic suspension property, the range of values measured is shown graphically in Figure 2.2.12. Vertical rate is expressed in the figure as the change of vertical axle load required to produce one inch vertical deflection of the suspension. The data are all presented on a per axle basis to enhance comparison of single- and two-axle suspensions. The values shown are the average, large deflection spring rates, generally taken at 16,000-pound axle loads for non-steering suspensions, and 10,000-pound axle loads for steering suspensions. Note that, for steel spring suspensions, front suspensions are consistently softer in an attempt to provide better ride for the driver. Air suspensions are clearly softer than other rear, or trailer suspensions. The generally low rate of air suspensions helps provide a better ride environment. Four-spring and walking-beam suspensions are generally rather stiffly sprung.

2.2.3.2 Composite roll stiffness. A comparison of the composite roll stiffness of a variety of suspensions is shown in the bar graph of Figure 2.2.13. In order to enhance comparison between single- and tandem-axle suspensions, all data are shown in this graph on an average, per axle basis. (Total composite roll stiffness of tandem-axle suspensions would be obtained by doubling the displayed value.) For steering suspensions, data were generally obtained at axle loads of 12,000 pounds. For non-steering suspensions, data were generally obtained at axle loads of 16,000 pounds.

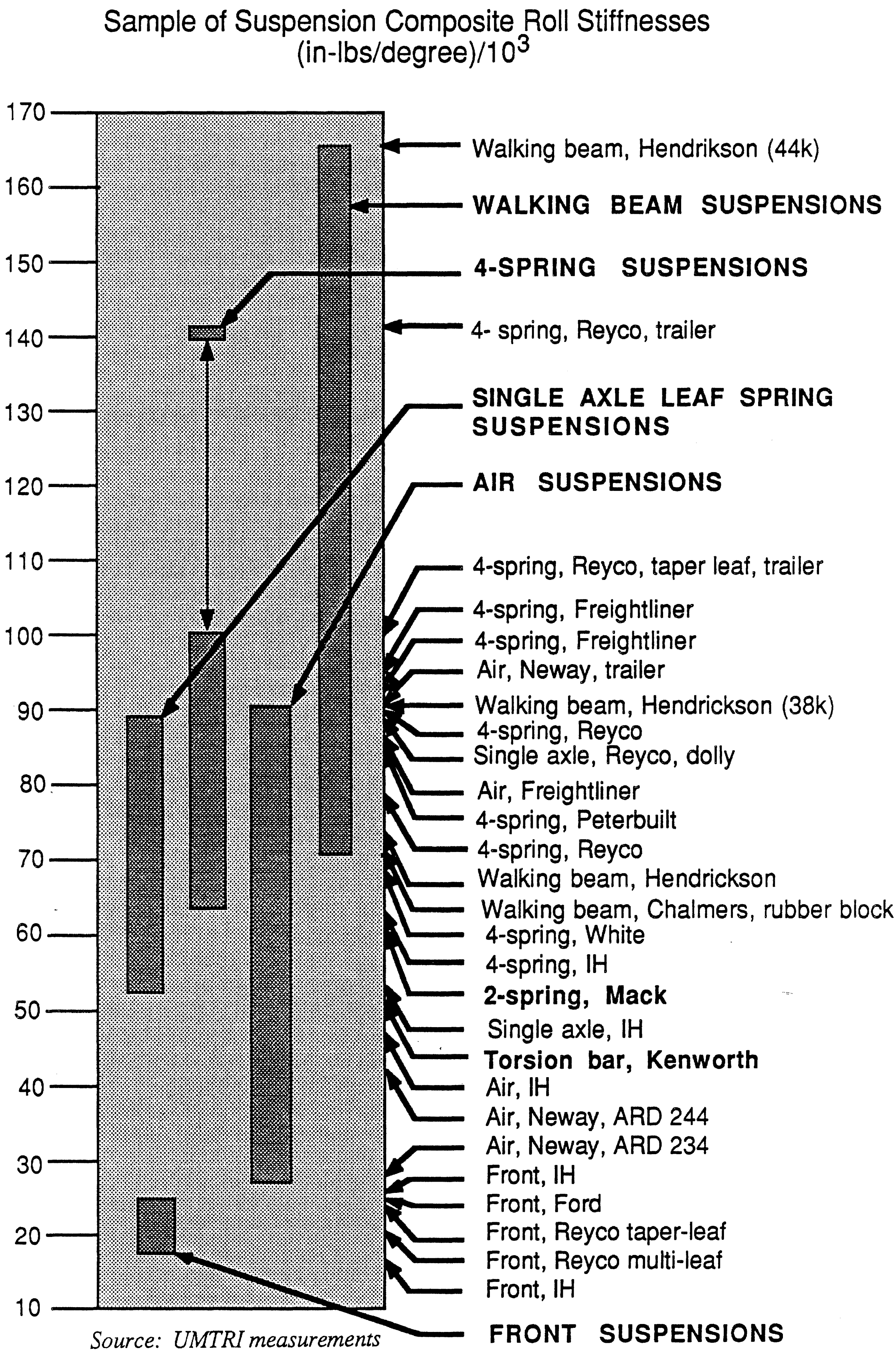
The graph clearly indicates that the measured front suspensions tend to be low in roll stiffness, even for their somewhat lower axle loads. As would be expected from their relatively

Sample of Suspension Composite Vertical Stiffnesses
lbs/in



Note: All values given are on a per axle basis. For tandem suspensions, the value presented is for the average of two axles.

Figure 2.2.12 Suspension composite vertical stiffnesses



Note: All values given are on a per axle basis. For tandem suspensions, the value presented is for the average of the two axles.

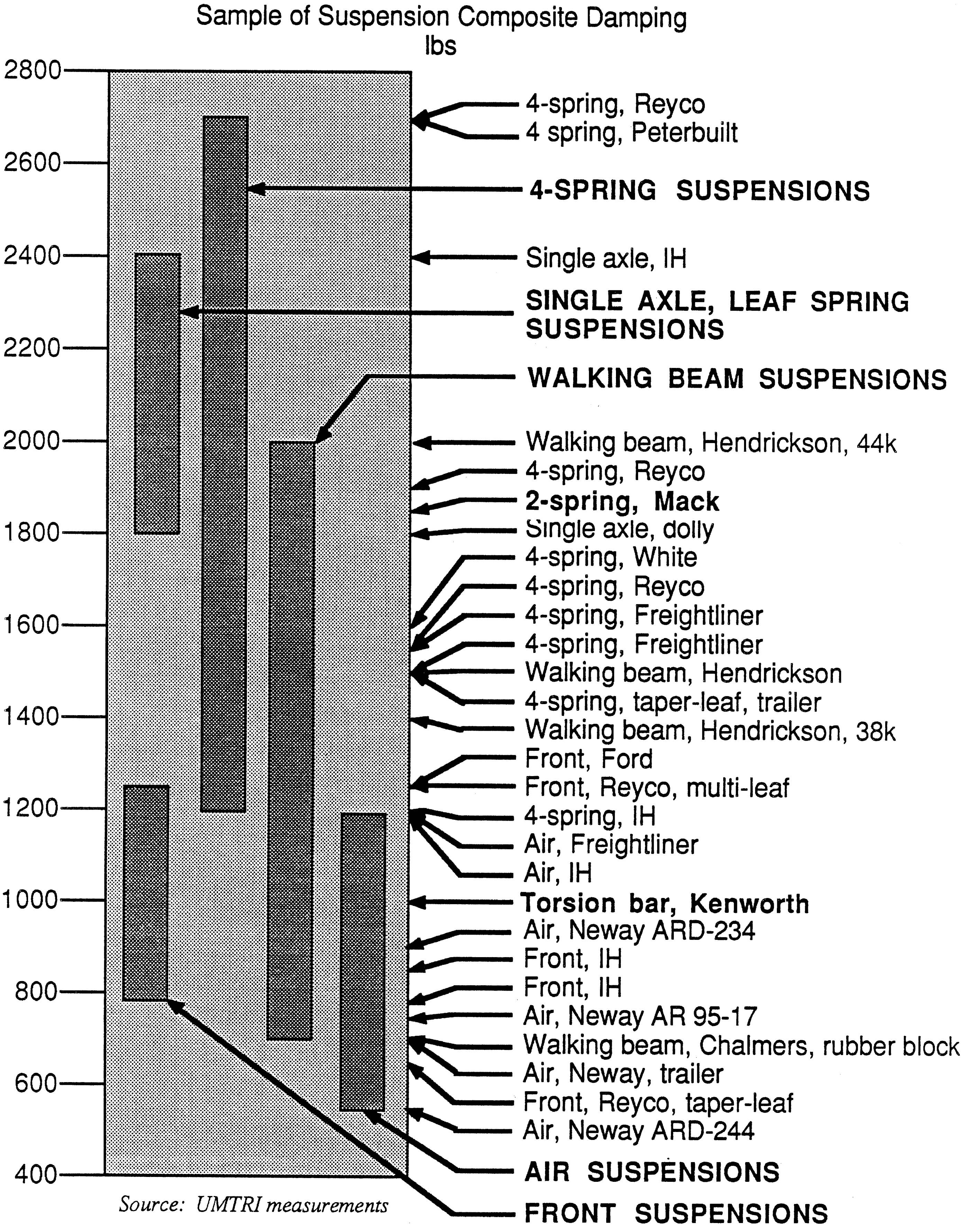
Figure 2.2.13 Suspension composite roll stiffnesses similar construction, single-axle leaf-spring suspensions and four-spring suspensions are generally similar, on a per axle basis (except for one trailer suspension sample). Air suspensions cover a broad range of stiffness. When auxiliary roll stiffness is relatively low, these suspensions are rather soft in roll, but they can be as stiff in roll as steel-spring suspensions when highly effective auxiliary roll stiffness mechanisms are used. The walking-beam suspensions measured show the highest range of roll stiffness, largely due to the somewhat higher load ratings of the measured samples.

2.2.3.3 Damping. Damping provided by suspension shock absorbers can be adjusted over a broad range by the proper choice of internal valving. Shock absorber damping values are chosen by the vehicle manufacturer to "tune" with other suspension properties to provide good ride performance.

Coulomb (or "dry") friction is usually the larger source of damping in steel spring truck suspensions, and derives mostly from inter-leaf friction and slipper friction. Coulomb damping generally is harmful to ride quality, but provides the friction necessary to damp roll and bounce motions of the body in dynamic maneuvers. Figure 2.2.14 shows representative measured data. The values shown are for large vertical deflections. For steering suspensions, data were generally obtained at axle loads of 12,000 pounds. For non-steering suspensions, data were generally obtained at axle loads of 16,000 pounds. Leaf-spring suspensions (single, four-spring, and most walking-beam) generally have higher friction than other types. Springs with fewer leaves provide less inter-leaf friction. Springs may be equipped with strips of low-friction material between the leaves to reduce Coulomb friction. Friction at the slipper may be a major portion of Coulomb friction in the suspension, particularly if the spring is well arched rather than flat.

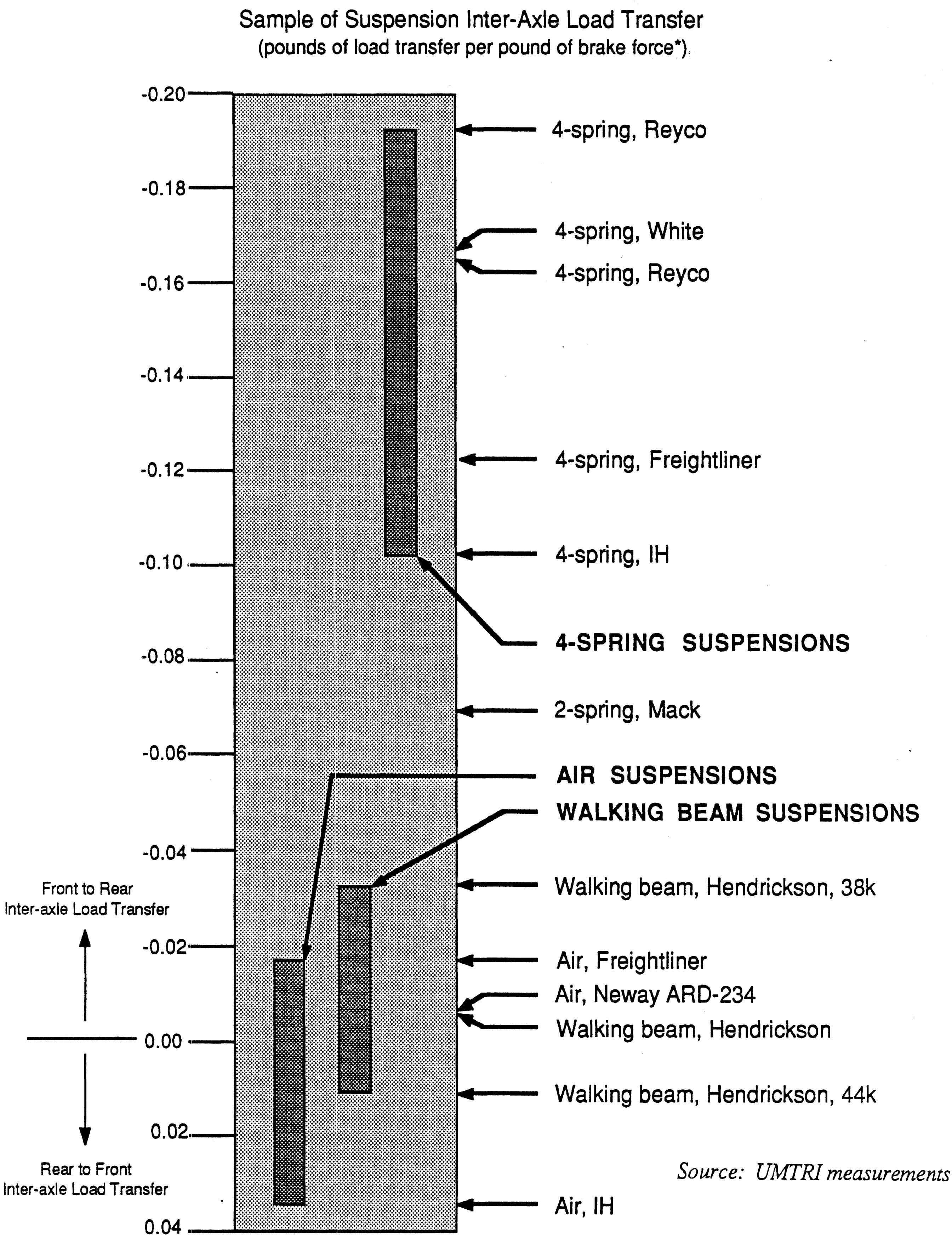
2.2.3.4 Load equalization and inter-axle load transfer in tandem suspensions. Most tandem suspensions are quite effective at equally distributing load. In static measurements of a number of walking-beam, four-spring, and other types, the ratio of leading to trailing axle load in the absence of brake forces range from a high of 1.09 to a low of 0.95. (A value of unity represents perfect load equalization.) Clearly, all the suspensions measured display good load equalization. The data do not, however, include the influence of Coulomb friction on the load equalization mechanisms, which may degrade actual load equalization performance in practice, particularly for four-spring suspensions.

Figure 2.2.15 displays the range of values of interaxle load transfer due to braking which have been measured. Interaxle load transfer is expressed as the ratio of vertical load transferred from the trailing axle to the leading axle, to the total brake force applied by the four wheels of the



Note: All values given are on a per axle basis. For tandem suspensions, the value presented is for the average of two axles.

Figure 2.2.14 Suspension composite coulomb damping



** Axle load transfered from trailing to leading axle / total brake force on suspension.*

Figure 2.2.15 Suspension inter-axle load transfer suspension. A value of zero implies no interaxle load transfer due to braking. A positive value indicates that vertical load is transferred from the trailing to the leading axle during braking, and, of course, load is transferred in the opposite manner if the ratio is negative. Clearly, four-spring suspensions are distinguished by showing more interaxle load tranfer than other tandem suspensions. (It should be noted that special attention to design details can greatly reduce interaxle load transfer in four-spring suspensions.)

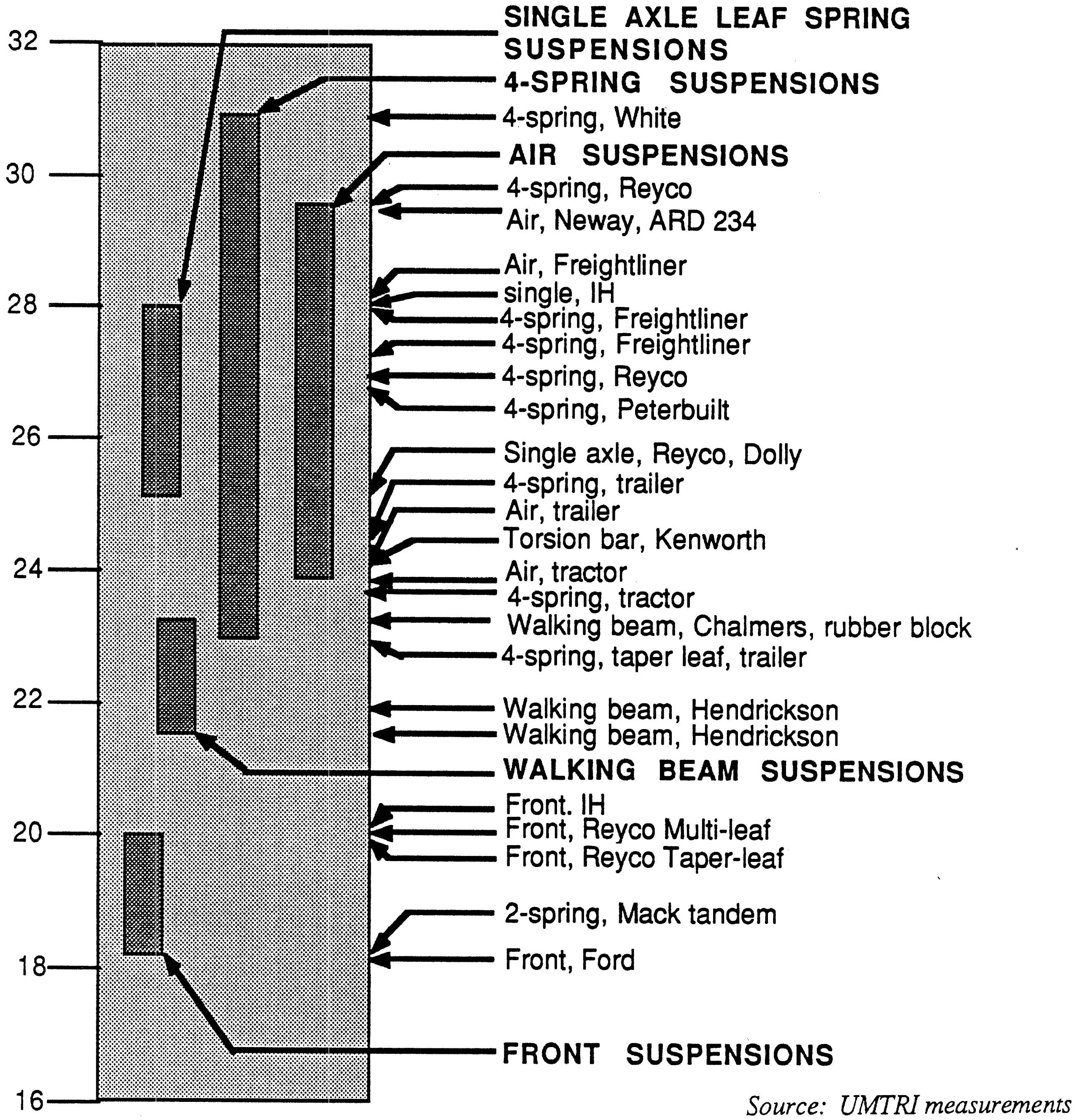
2.2.3.5 Roll center height. Figure 2.2.16 illustrates the range of suspension roll center heights which have been measured. The data show that single-axle, leaf-spring suspensions, four-spring suspensions, and air suspensions all tend to have roll center heights about 27 inches above the ground, with four-springs showing the largest range (probably because more have been measured). Walking-beam suspensions have roll centers several inches lower and the roll centers of front suspensions are still lower. Since a high roll center has a similar influence as does a high level of roll stiffness, then the low roll center of front suspensions compounds the problem of low roll stiffness at the front suspension.

2.2.3.6 Roll steer. Figure 2.2.17 displays measured values of roll steer for non-steering heavy vehicle suspensions. (See Section 2.3 on steering systems for steering axle properties.) Roll steer is expressed in the ratio of degrees of suspension roll deflection per degrees of axle steer angle. Suspension roll deflection is positive when the right side spring is compressed and the left side spring is extended. Steer is positive when the axle (not the vehicle) steers toward the right. For example, in a left-hand turn the body of the vehicle will lean toward the right causing positive suspension roll. If the rear axle roll steer coefficient is positive, then the rear axle will steer toward the right (in a manner which would cause the rear of the vehicle to swing out of the turn.) Thus, positive roll steer coefficients at rear axle positions have an oversteer influence.

Generally, the data in the figure were gathered at axle loads of 16,000 pounds. Single-axle and four-spring suspensions are seen to generally have low, usually positive, values of roll steer. Air suspensions and walking beams generally have higher, positive values of roll steer. The highest values shown would add a couple of degrees per g of oversteer to the vehicle.

2.2.3.7 Compliance steer. Compliance steer influences are generally low for non-steering suspensions. Figure 2.2.18 shows the values of aligning moment compliance steer for a number of suspensions. All the values are low, with the largest amounting to less than 1/2 degree per g of understeer. Lateral force compliance steer is generally even less, the exception being trailing-arm suspensions (for example, most air suspensions) if not equipped with a lateral locating link between the axle and frame.

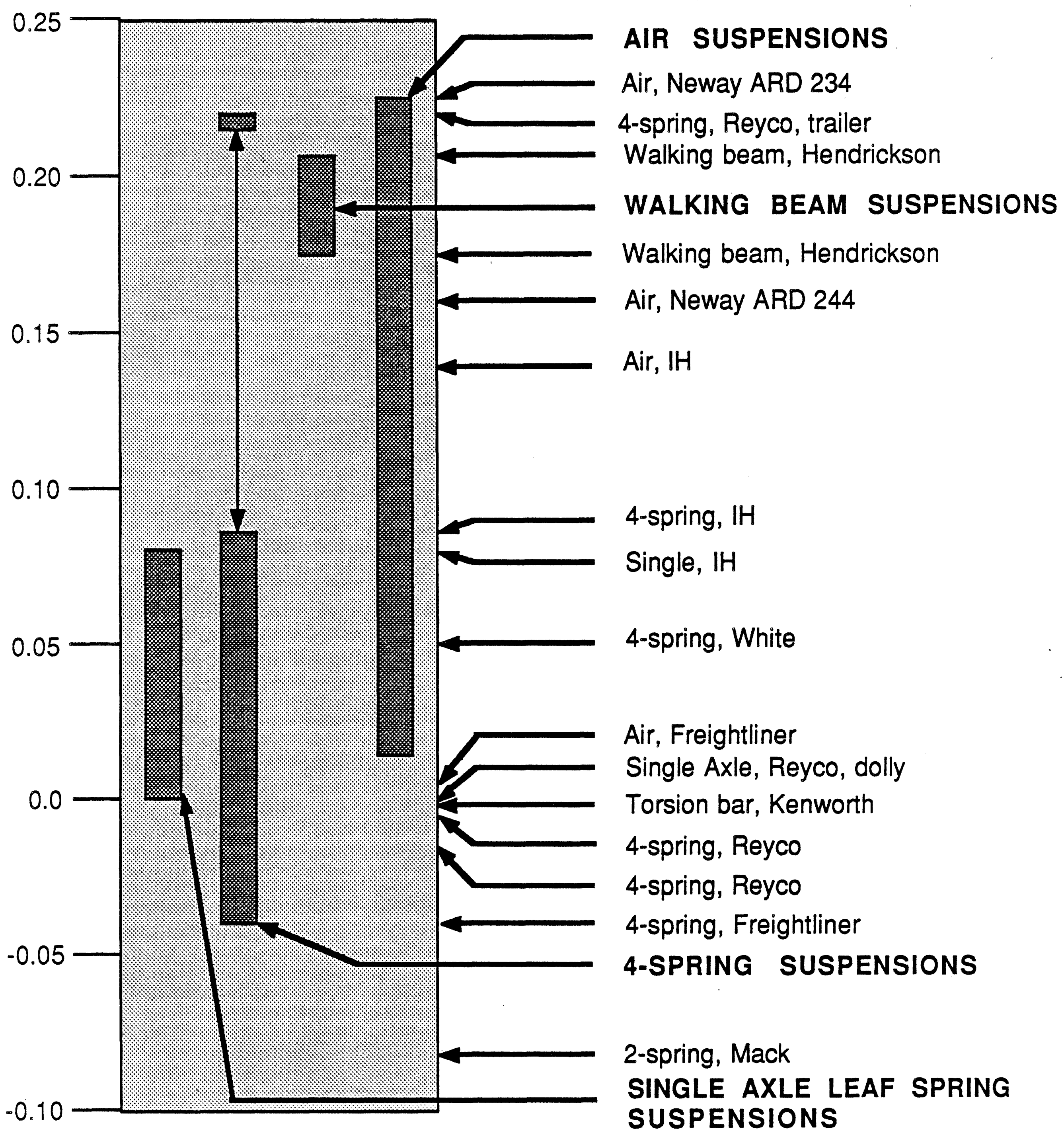
Sample of Suspension Roll Center Heights
(Inches above the ground)



Note: All values given are on a per axle basis. For tandem suspensions, the value presented is for the average of the two axles.

Figure 2.2.16 Suspension roll center heights

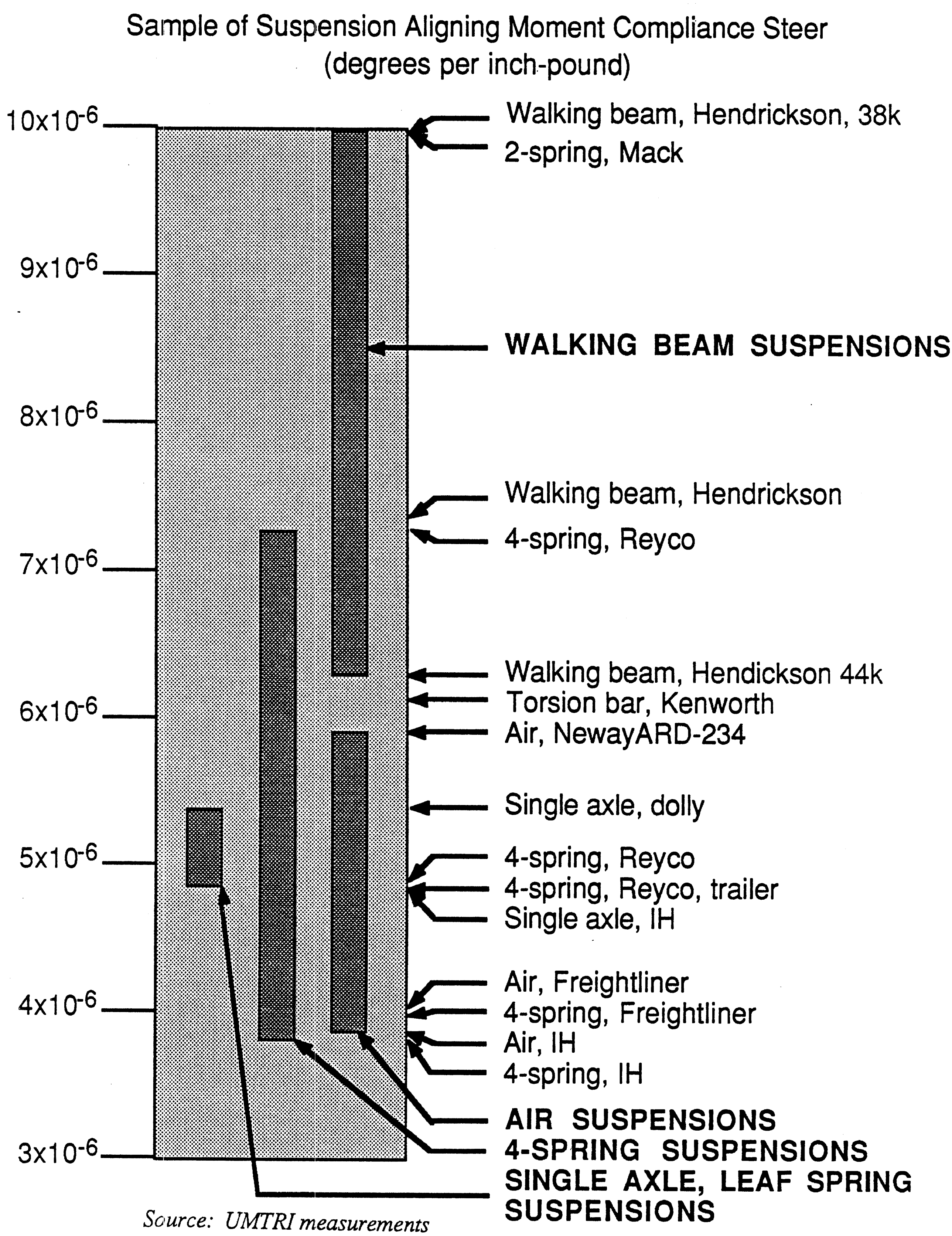
Sample of Suspension Roll Steer
(degrees steer per degrees roll)



Source: UMTRI measurements

Note: All values given are on a per axle basis. For tandem suspensio the value presented is for the average of the two axles.

Figure 2.2.17 Suspension roll steer



Source: UMTRI measurements

Note: For tandem suspensions, the value shown is the average value for the two axles.

Figure 2.2.18 Suspension aligning moment compliance steer

2.3 Steering Systems

2.3.1 The Mechanical Properties of Steering Systems. To most people, a steering system is a set of linkages by which rotation of the steering wheel is transformed into steer angle changes at the front wheels as the means to control the vehicle's path. From a vehicle dynamicist's point of view, however, the system extends down to include the geometry of each of the steered wheels. Most trucks in the medium and heavy classes have an I-beam front axle with a leaf-spring suspension with a steering system configured as shown in Figure 2.3.1 [5]. Steering-wheel rotation is the "command" input by which the driver controls vehicle direction. Hence, the vehicle's control properties are perceived by its response to inputs at this point.

Driver input of an angle at the steering wheel causes a fore/aft movement of the drag link, which steers the left wheel via a steering arm. A tie-rod linkage connects the left and right wheels, thereby steering the right wheel to an appropriate angle. Rotation of the steering wheel is expected to produce a steer rotation angle at the front wheels, causing the vehicle to turn on a radius. A number of mechanisms, described in other sections of this document, alter the radius of turn achieved at a given operating condition, thus giving the vehicle its unique set of response properties. The steering system also includes certain mechanisms which may alter the steer angle produced at the front wheels as a function of its operating state, thereby contributing to the directional response behavior of the vehicle.

The primary mechanical properties of the steering system that are responsible for these mechanisms are presented in Figure 2.3.2.

2.3.1.1 Roll steer. The driver's command input for steer angle at the front wheels is communicated across the deflections of the suspension system by the longitudinally mounted drag link. As the axle moves vertically with respect to the frame, any fore/aft movement at the end of the steering arm that does not center on the ball pivot of the pitman arm will result in a steer angle change at the road wheels, as illustrated in Figure 2.3.3. Error in the linkage motion is commonly called "steering geometry error." Depending on its nature, it may produce steer angle deviations as shown in the upper illustration, which are in the same direction for both jounce and rebound [6]. In this case, the error results in "steering fight" as the vehicle negotiates normal road bumps, and a steer deviation as the front suspension settles down during braking.

Alternatively, the error may produce a steer deviation which changes monotonically with suspension deflection, as shown in the lower illustration. Errors of this type are a source of roll

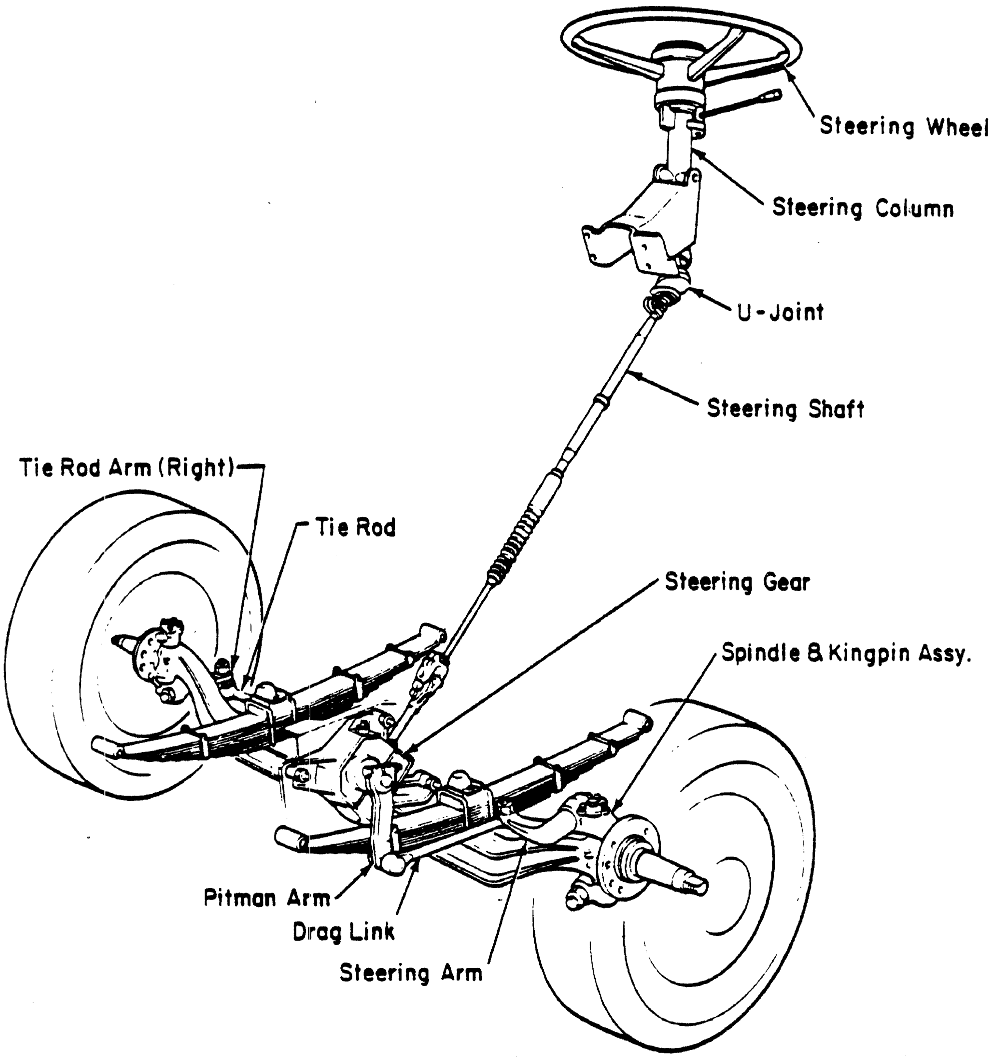


Figure 2.3.1 Typical steering system configuration on medium and heavy trucks

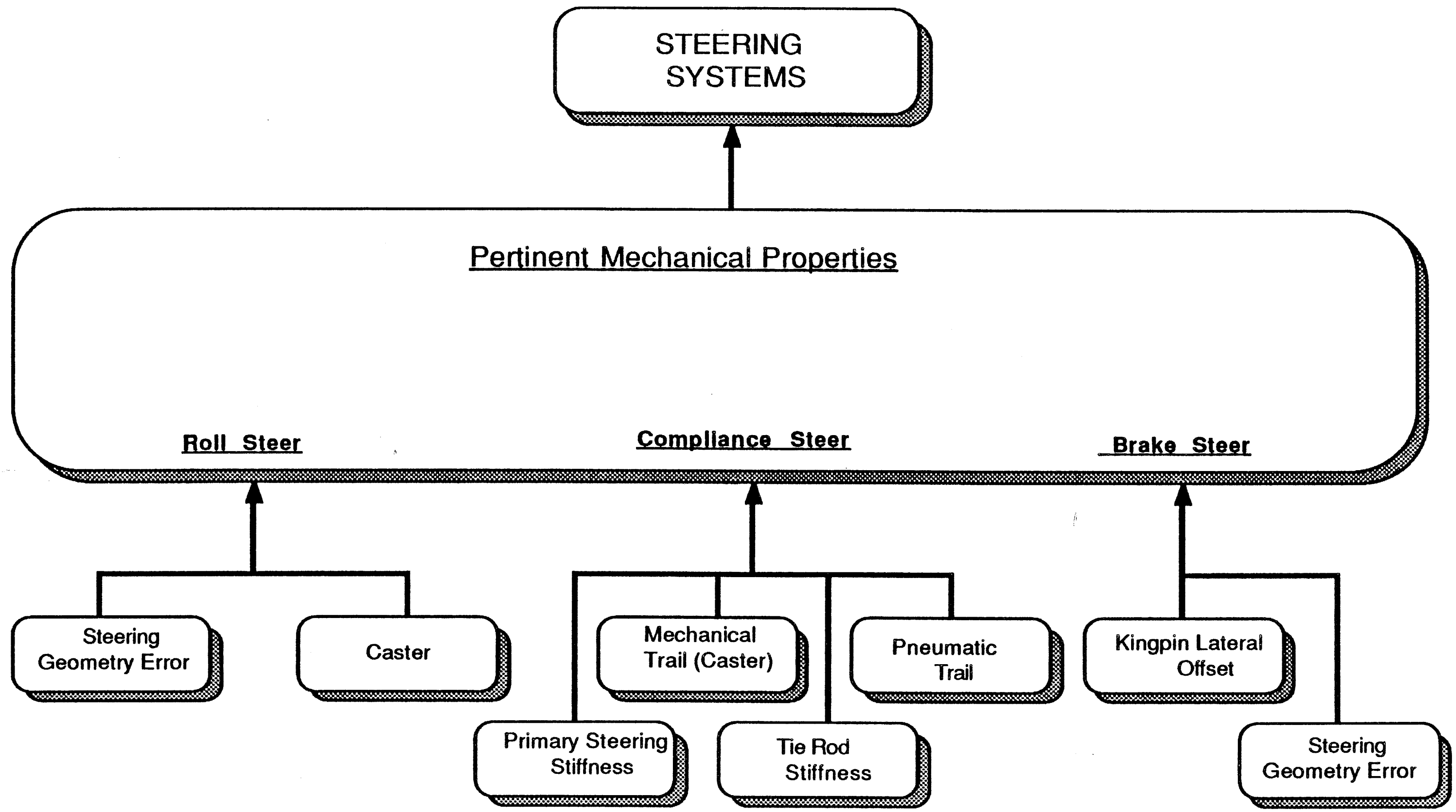
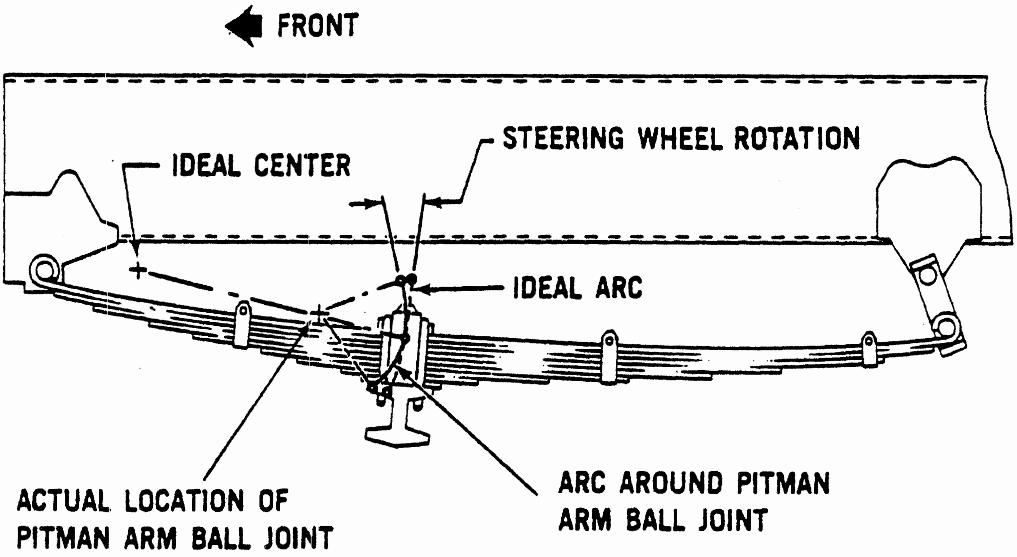
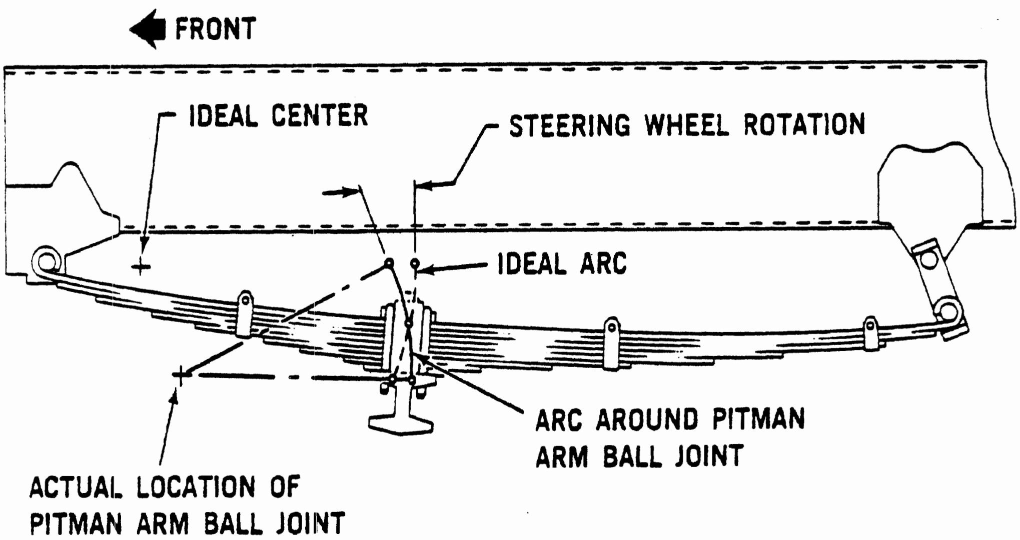


Figure 2.3.2 Pertinent mechanical properties of steering systems



Asymmetric steering geometry error



Symmetric steering geometry error

Figure 2.3.3 Examples of steering geometry error steer on a front axle. That is, as the vehicle negotiates a turn, the roll angle deflects the suspension and causes a steer angle change which impacts on the directional response behavior.

(A secondary mechanism contributing to roll steer arises from the lateral load transfer on an axle caused by roll in cornering. Caster angle on the steer rotation axis generates a moment in the steering system when the left and right wheel loads differ. The moment attempts to steer the wheels, acting against compliance in the linkages to produce a steer angle change at the wheels.)

2.3.1.2 Lateral force compliance steer. During cornering, large lateral forces may be developed at the front wheels. These forces do not necessarily act through the steer rotation axis. Caster on the axle creates an offset (the "mechanical trail"), as illustrated in Figure 2.3.4, with the result that these forces attempt to steer the wheels. The steering system is not perfectly rigid. Every element is compliant to some extent, allowing the lateral forces to cause a deviation in the steer angle of the wheels. The multiple compliances can be lumped into two key elements, as shown in Figure 2.3.5 [7]. The primary steering stiffness, K_{ss}, represents all the compliances between the steering wheel and the left road wheel. This compliance allows the left road wheel to deviate from the "command input" from the driver. The tie rod stiffness, K_{tr}, represents the compliances between the left and right wheels. It allows the right wheel to steer to a slightly different angle than the left. Its influence is taken into account by recognizing that left- and right-wheel steer angles will differ, and a mean steer angle must be used.

The compliance of the steering system allows the steer angles to be affected, as well, by other forces and moments acting on the front tires. The aligning moment is a significant contributor through this mechanism. The aligning moment arises because the lateral force generated in the tire contact patch is offset toward the rear of the contact patch. The offset is known as the "pneumatic trail."

The combined effects of these mechanisms are to allow the steer angles at the front wheels to change in proportion to the lateral force present on those wheels. The change in steer angle reduces the lateral forces in a fashion equivalent to a reduction of cornering stiffness of the tires. The effective cornering stiffness obtained is expressed by Equation (2.3.1):

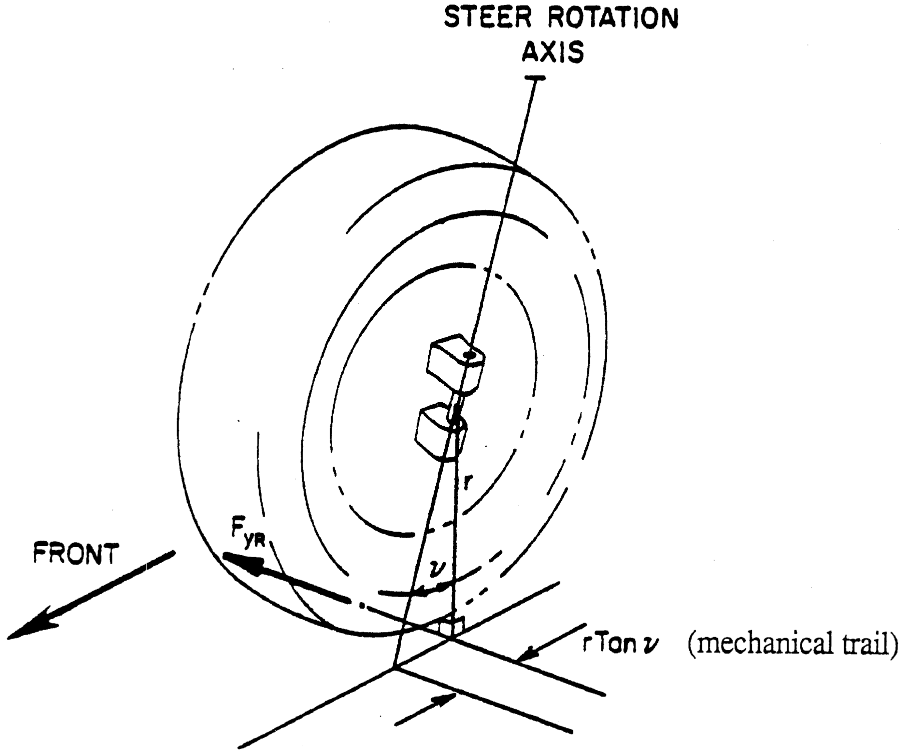


Figure 2.3.4 Kingpin moment produced by lateral force

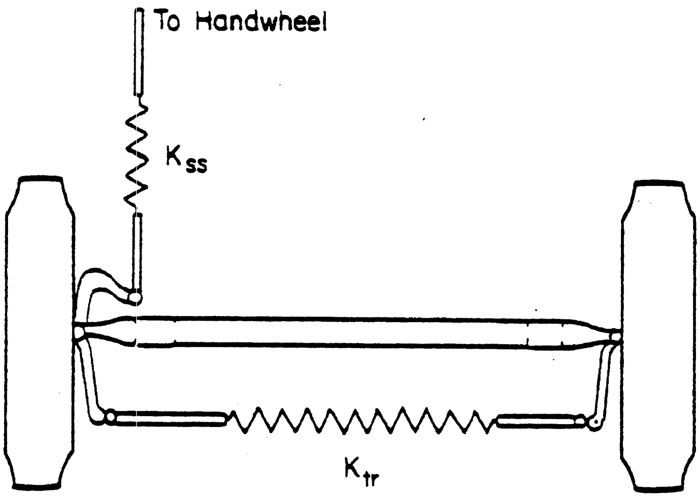


Figure 2.3.5 Steering linkages modeled as stiffnesses

$$C_\alpha^* = C_\alpha \frac{1 + C_{\alpha f}(x_m + x_p)/(2\, K_{tr})}{1 + C_{\alpha f}(x_m + x_p)\,[2/K_{ss} + 1/K_{tr} + C_{\alpha f}(x_m + x_p)/(K_{ss}\, K_{tr})]} \qquad (2.3.1)$$

where

C_{af}^* = effective cornering stiffness value per tire

C_{af} = cornering stiffness of one tire

x_m = mechanical trail

x_p = pneumatic trail

K_{ss} = primary steering stiffness

K_{tr} = tie-rod linkage stiffness

2.3.1.3 Brake steer. Under braking conditions, several mechanisms may act to cause a steer angle on the front wheels, turning the vehicle from its path. The simple forward load transfer in braking will compress the front suspension. Steering geometry error in jounce will produce a steer angle deviation. The pitch of the vehicle frame also changes the effective caster on the front axle. With normal leaf-spring suspensions, the brake torque will cause the axle to wrap up further reducing the caster in the steering system, altering the magnitude of the effective cornering stiffness in the process.

Brake steer will also arise from the brake forces generating torques in the steering system directly, as illustrated in Figure 2.3.6. The brake forces act at the center of tire contact, outside of the steering axis by a distance which is the kingpin lateral offset at the ground. The brake force on the right wheel attempts to steer against the compliance of the tie rod, and in the process imposes a torque on the left wheel. The left-wheel brake force produces a torque in the opposite direction, which is reacted against the primary steering stiffness and the torque coming from the right wheel. As a consequence, each wheel assumes a slightly different steer angle deviation. The angles are given by:

$$\Delta\partial_l = (F_{xr} - F_{xl})\, d/K_{ss} \qquad (2.3.2)$$

$$\Delta\partial_r = \Delta\partial_l + F_{xr}\, d/Ktr = (F_{xr} - F_{xl})\, d/K_{ss} + F_{xr}\, d/K_{tr} \qquad (2.3.3)$$

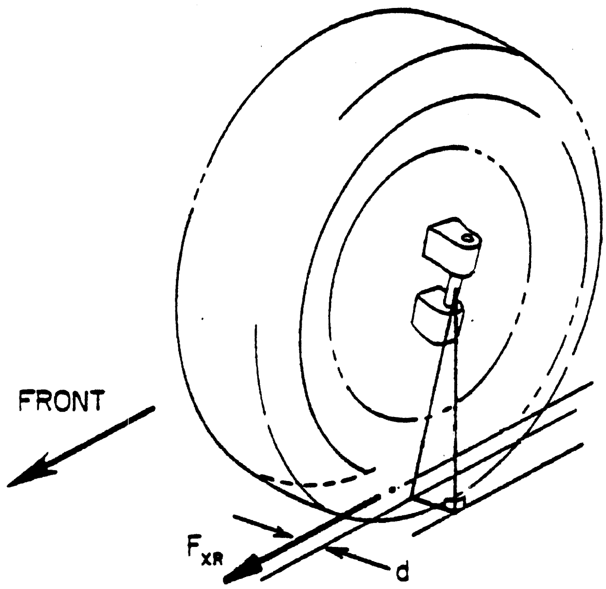


Figure 2.3.6 Kingpin moment produced by tractive forces

where

F_{xr}, F_{xl} = right and left brake forces

$\Delta\partial_r, \Delta\partial_l$ = right and left steer angle deviations

d = kingpin lateral offset at the ground

These mechanisms may be modeled directly in a vehicle computer simulation [8], in which case their effects are taken into account directly in computing vehicle path changes.

Alternately, the brake steer effect can be estimated by determining an average steer angle deviation for both front wheels, $\Delta\partial_{av}$. The average is simply obtained from the equations above, and is given by the expression:

$$\Delta\partial_{av} = (F_{xr} - F_{xl})\, d/K_{ss} + F_{xr}\, d/(2\, K_{tr}) \qquad (2.3.4)$$

With equal forces produced at the left and right wheels (perfect brake balance), nevertheless, a small steer angle results from the second term in the above equation. With positive offset (d positive) the steer is to the right.

2.3.1.4 Gear ratio. The steering gear ratio is an obvious and well-known parameter relating to a steering system, although it has no direct effect on dynamic behavior. For vehicle dynamics work it is defined as the ratio of steering-wheel angle to road-wheel steer angle.

2.3.2 Properties Important to Cornering and Braking. With the exception of steering gear ratio, each of these mechanical properties can affect the dynamic performance of a truck. (Gear ratio affects its maneuverability and steering effort, but is not so directly linked to dynamic performance.) A summary of the influences is presented in Table 2.3.1.

Roll steer exerts its influence on turning behavior in the performance modes of low-speed tracking, high-speed tracking, and yaw stability. The steering geometry errors that cause roll steer also cause steer as the vehicle goes over bumps, which can be very objectionable. Thus there is serious effort in the design to minimize these errors, and the roll steer influences are kept small. Hence, they are indicated as having low influence on these performance modes in the table.

Lateral force compliance steer is a significant mechanism in the high-speed turning of trucks. At low speed the lateral forces are small, hence the influences are low; but at high speed the lateral forces on the front axle give the mechanism importance. *This mechanism has been*

Table 2.3.1

The Importance of the Pertinent Mechanical Properties of
Steering Systems
on Vehicle Dynamic Performance

Pertinent Mechanical Property of Steering Systems	Low Speed Tracking	Hi Speed Tracking	Roll Stability	Yaw Stability	Response Time	Rearward Amp	Braking Efficiency	Transient Braking	Downhill Braking	Response to Disturbances
Roll Steer	-	-	-	Low	-	-	-	-	-	-
Lateral Force Compliance Steer	-	-	-	Hi	-		-	-	-	-
Brake Steer	-	-	-	-	-	-	-	Med	-	Hi
Gear Ratio	-	-	-	-	-	-	-	-	-	-

recognized as one of the most significant understeer sources on a truck, thus it has a high influence on yaw stability.

Brake steer is only relevant to braking situations, and then only in dynamic maneuvers. With moderate brake application in a steady fashion (i.e., downhill braking), the small steer deviations produced are insignificant in comparison to the normal steering corrections applied by the driver. Under high deceleration braking, near the limit of adhesion, the brake forces are higher and the dynamic excursions are greater, thus brake steer falls in the medium range of influence. On short-wheelbase vehicles approaching the lockup limits on the front axles, this mechanism may rank high in its influence on dynamic behavior; while for long-wheelbase vehicles it may more appropriately be classified as low. Disturbances in braking due to cornering, brake imbalance, or side-to-side differentials in surface coefficient may potentially have a high influence on the vehicle's dynamic behavior through this mechanism.

2.3.3 Typical Values of These Mechanical Properties. The roll steer properties of trucks will vary with vehicle design, the suspension system, and the operating condition. Roll steer is characterized by a roll steer coefficient relating the degrees of steer angle per degree of roll. Generally, the roll steer influence is understeer in direction, and ranges in magnitude from 0 to 0.2 (deg steer/deg roll). On a given vehicle the roll steer coefficient will vary within this range as a function of load on the steering axle and as a function of steer angle. Typical measurements and ranges are shown in Figure 2.3.7.

Lateral force compliance steer can be characterized for its influence on truck behavior by the apparent change that it produces in the cornering stiffness of the front wheels. Figure 2.3.8 shows the typical range of values by the ratio of effective cornering stiffness, $C_\alpha{}^*$ to the tire cornering stiffness, C_α. A reference vehicle has been selected for illustrating the range of variation that may occur. The reference vehicle is assumed to have 12,000-lb gross front axle load, the most common maximum for highway tractors. The caster angle (determining the mechanical trail) is variable from vehicle to vehicle, and has a potentially large influence depending on its specific value. Caster normally ranges from 0 to 5 degrees. Figure 2.3.9 shows some typical ranges from Ford vehicles.

The type of steering-axle tire—radial versus bias—has an influence on the effective cornering stiffness ratio, $C_\alpha{}^*/C_\alpha$. The pertinent tire properties are reflected in the values of cornering stiffness and pneumatic trail. The typical values of these parameters for radial- and bias-ply tires are shown in Figures 2.3.10 and 2.3.11. Bias-ply tires have a lower cornering stiffness, but a larger pneumatic trail. The net effect is to have a larger effective cornering stiffness ratio

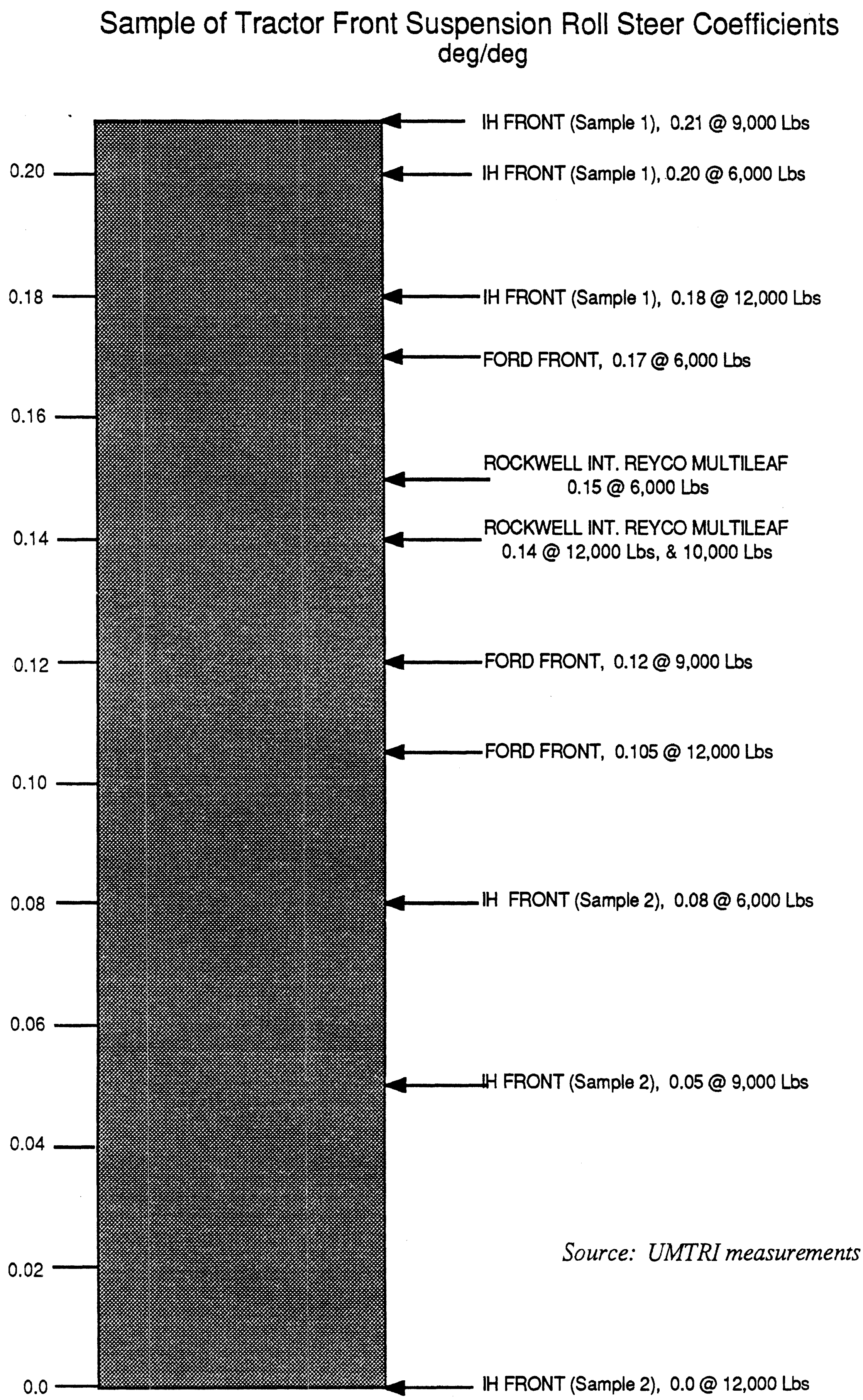


Figure 2.3.7 Tractor front suspension roll steer coefficients

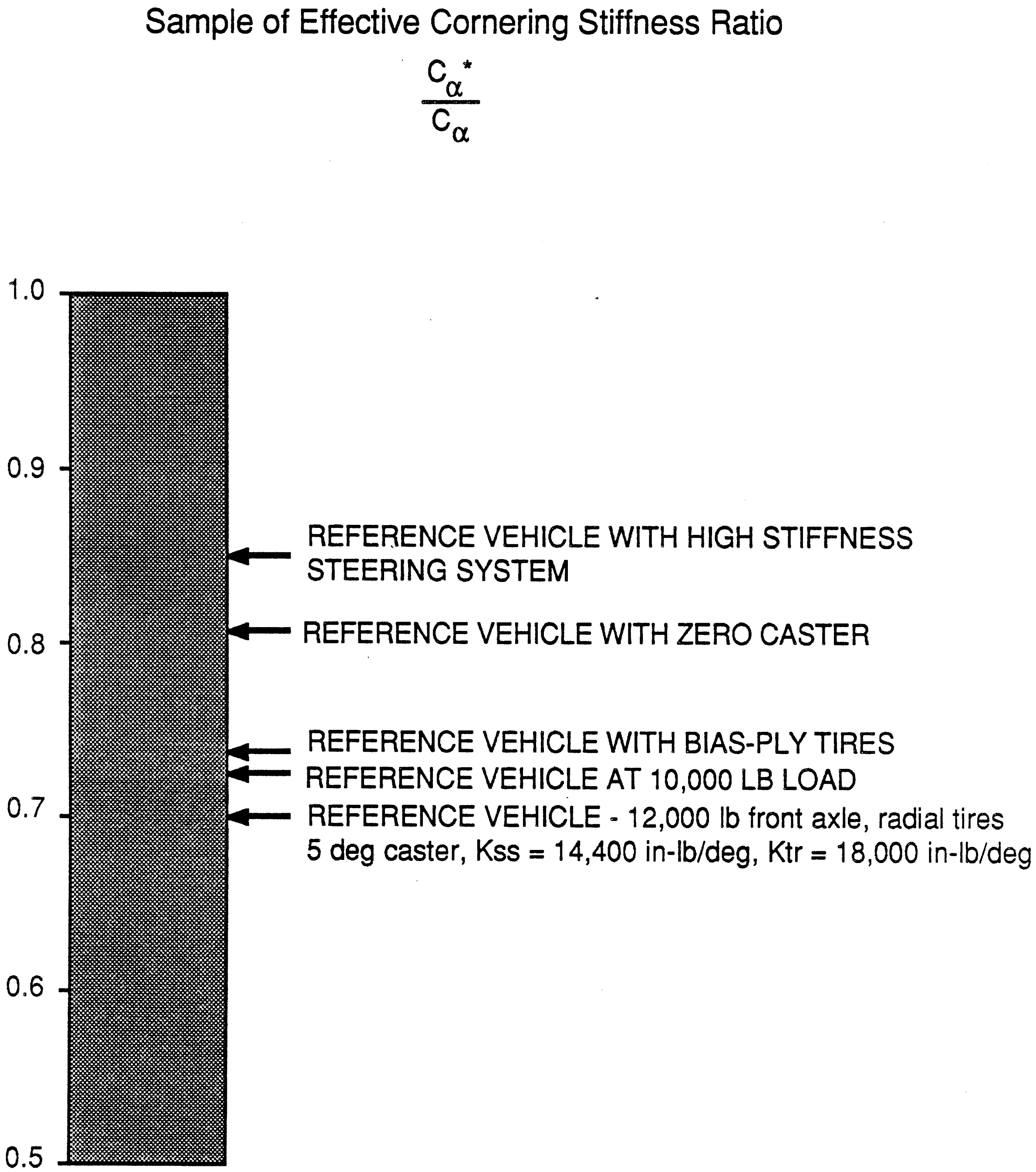


Figure 2.3.8 Effective cornering stiffness ratio (Composite stiffness ratios for the variations indicated)

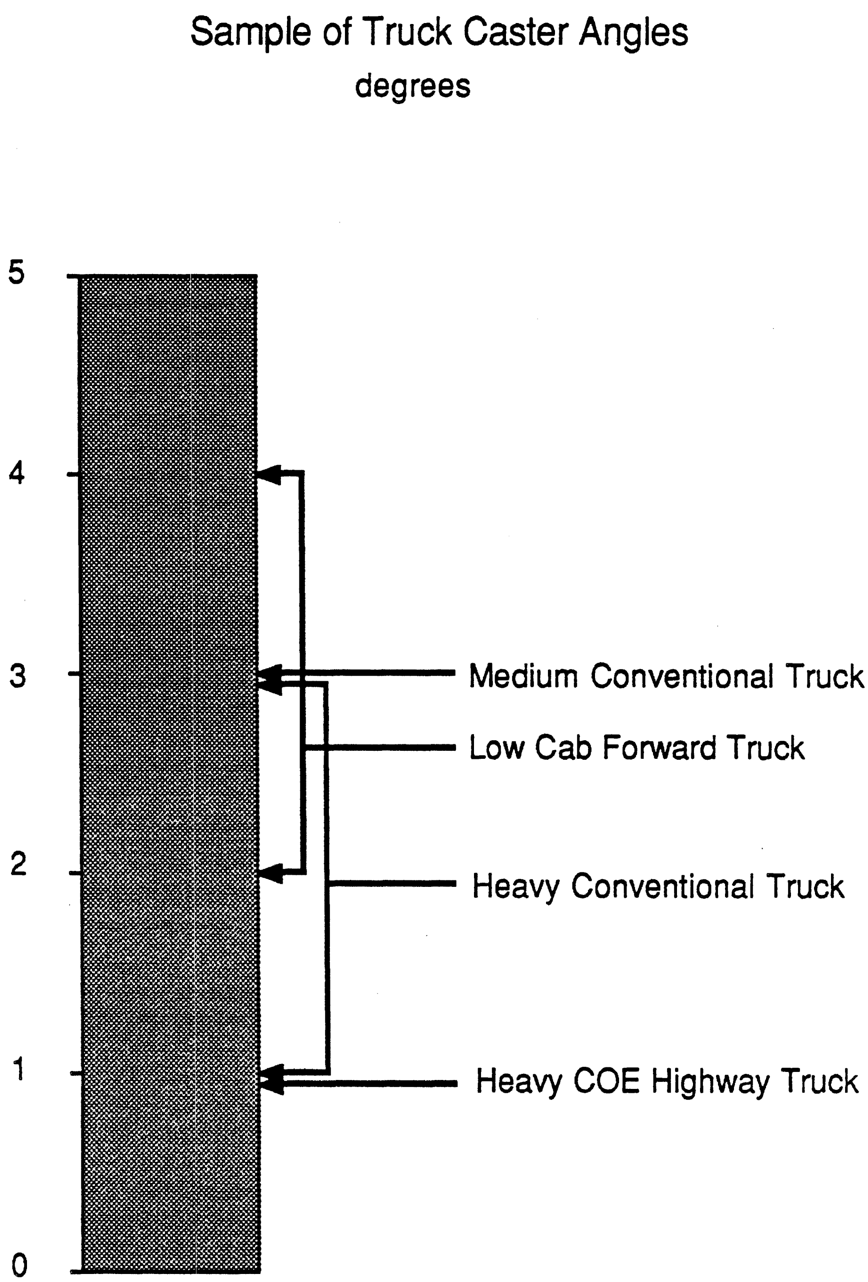


Figure 2.3.9 Typical truck caster angles

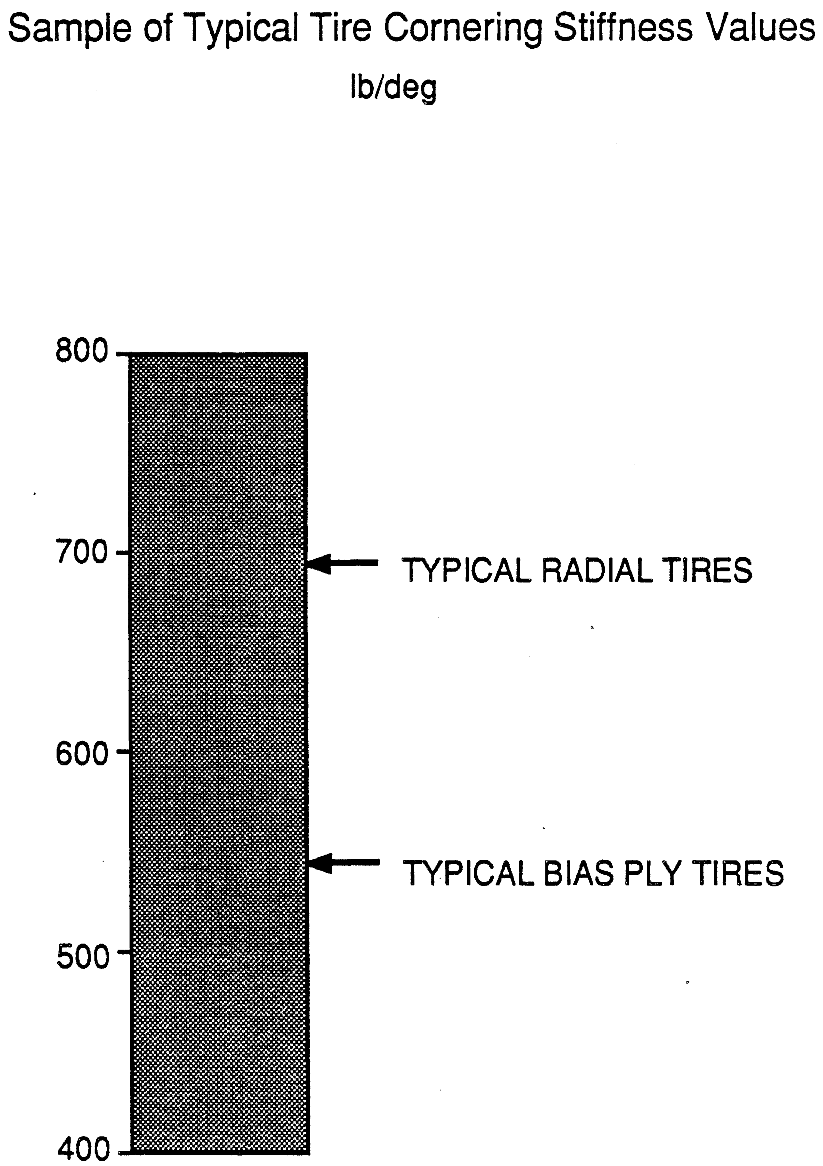


Figure 2.3.10 Typical tire cornering stiffness values
(See also Figure 2.1.10)

Sample of Typical Pneumatic Trail Values
inches

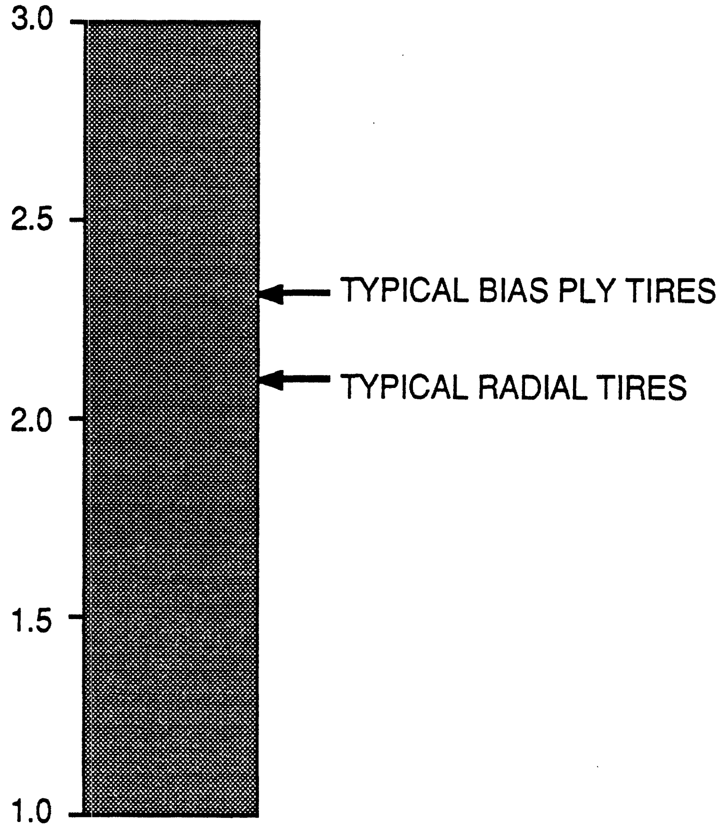


Figure 2.3.11 Typical pneumatic trail values
(See also Figure 2.1.12)

(C_α^*/C_α), as seen in Figure 2.3.8. However, the effective cornering stiffness (C_α^*), which has the direct influence on cornering behavior, is still lower than with radial tires because of the significantly lower cornering stiffness (C_α) of the bias-ply tires. The effect of the steering system compliances is only to reduce somewhat the difference between the two types of tires.

A high steering stiffness increases the effective cornering stiffness significantly, as seen in Figure 2.3.8. Data on steering system stiffness are quite limited. Figure 2.3.12 shows some measured values of primary steering stiffness at zero steer angle obtained at UMTRI. Figure 2.3.13 shows comparable data for tie-rod linkage stiffness. There is not a great difference in the stiffness of manual or power steering systems. Although the data shown were obtained on axles in the 10,800 to 12,000 GAWR range, the stiffnesses are believed to be generally proportional to the GAWR rating of an axle. Thus smaller axles with lighter loads will have steering systems that are proportionately less stiff.

Normally the axle load is not especially significant because front axle loads do not vary over a great range in operation. With a reduction in load on a given axle, the tire cornering stiffness will decrease, changing the effective cornering stiffness as well. An example of a 2,000-lb load reduction is shown in Figure 2.3.8.

Brake steer effects are directly proportional to the magnitude of the brake force, so it is not easy to characterize typical values. Equation 2.3.4 gives a means to estimate steer angle deviation as a function of brake force. The brake force levels vary from zero up to a maximum which is determined by the coefficient of friction of the front tires. An upper limit may be estimated from the product of the coefficient of friction and the dynamic load on the axle (which, in turn, is determined by braking level and other vehicle characteristics). The left-to-right differences are not well known. Generally a 15 percent difference is a threshold at which brake imbalance could contribute to a significant brake steer effect.

Steering stiffness values are given in Figures 2.3.12 and 2.3.13. The kingpin lateral offset at the ground is normally on the order of 2 to 3.5 inches. The special "centerpoint" steering axles are designed to achieve near zero offset, and usually are on the order of 0.5 inches or less. Further, kingpin offset at the ground will be a function of the wheels installed on the vehicle, and the hub and drum combination used.

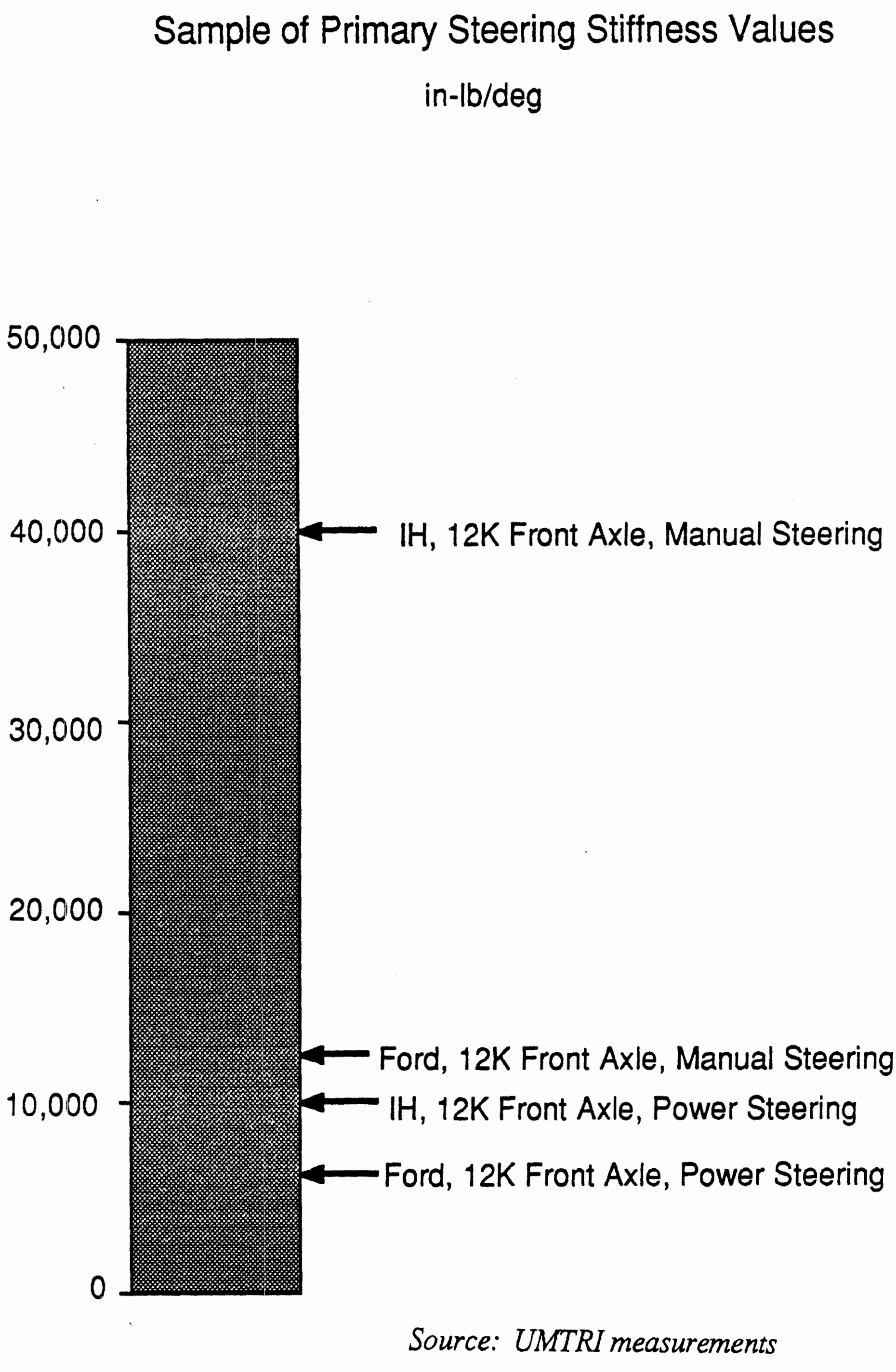


Figure 2.3.12 Primary steering stiffness values,Kss, between the steering wheel and the left road wheel.

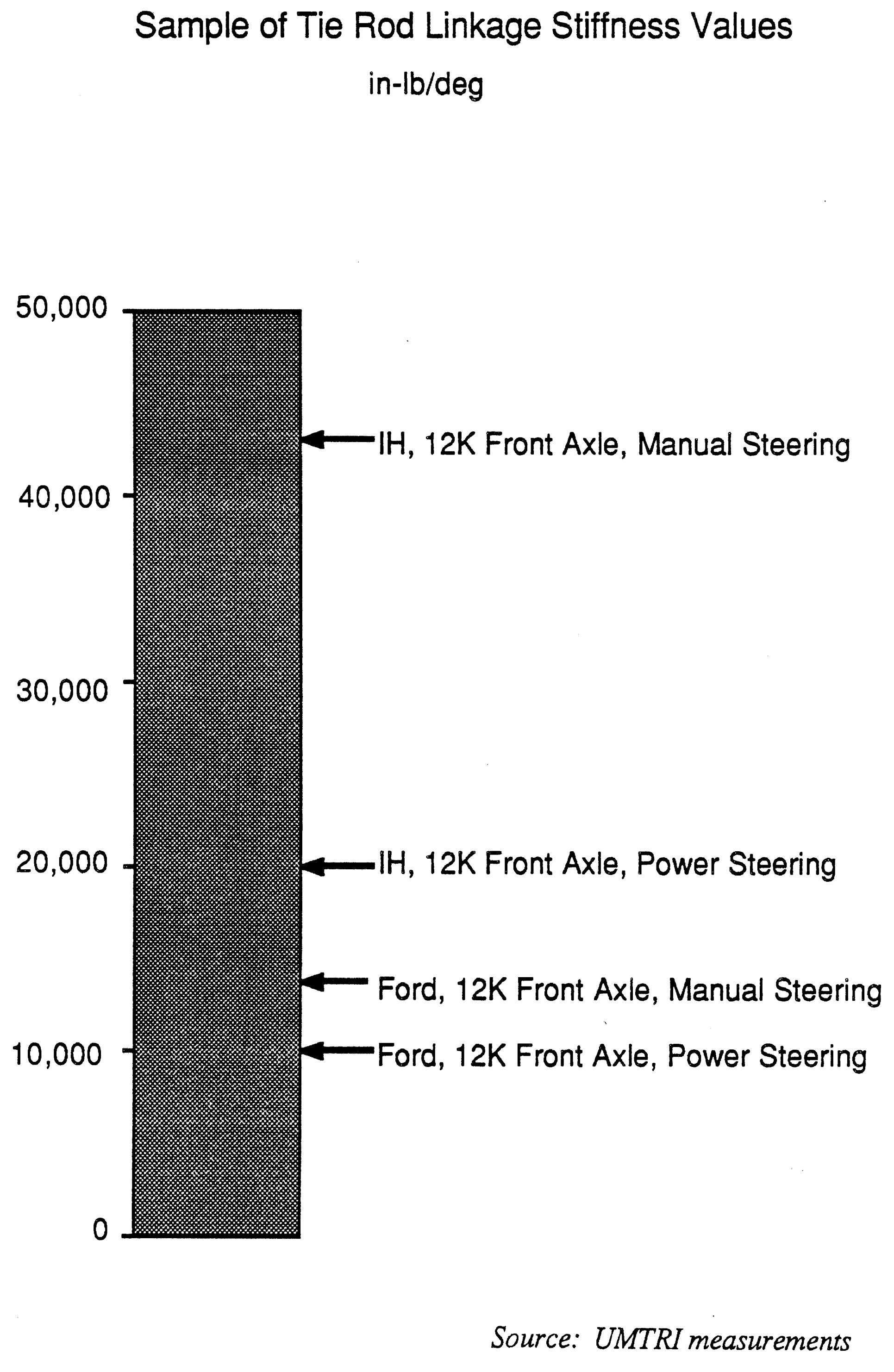


Figure 2.3.13 Tie rod linkage stiffness values, K_{tr}, between left and right roadwheels

2.4 Brakes

2.4.1 The Pertinent Mechanical Properties of Brakes. Clearly the function of a brake is to limit speed. The maneuvers, in which this function is required, are: normal stops; "snubs" (speed corrections); downhill descents (speed maintenance); and emergency stops. In all of these cases most of the energy possessed by the vehicle, be it kinetic or potential energy, is dissipated through its brakes.

From the point of view of energy absorption, heavy-truck brakes are marvelous devices. They can absorb large amounts of energy in short periods of time and they can do this over and over without being damaged to the extent that they will not perform satisfactorily. This capability is obviously necessary if trucks are to be operated in traffic on highways.

With regard to safely stopping a vehicle, the driver controls speed by modulating the air pressure delivered to the brakes. If the vehicle is not decelerating satisfactorily, the driver increases the treadle pressure thereby increasing the brake torque. Treadle pressure is decreased if the vehicle is going to stop short of the intended spot. Drivers can easily control stopping location if they have adequate distance, brake torque, and tire/road friction available.

However, if the driver overbrakes such that the wheels on some axles lock, directional control and stability may be lost with jackknifes, trailer swings, or ploughouts ensuing. These control difficulties depend upon the overall braking system, not just on an individual brake. The balance of the torque capabilities from brake to brake is important in determining the directional stability of the vehicle.

Nevertheless, for the purposes of achieving a good balance of brake torques, it is necessary to know the torque versus pressure relationship for each brake installed on a particular vehicle. This relationship is called the "effectiveness" of the brake. To the extent that the effectiveness function can be approximated by a linear function, brake performance can be characterized by a brake "gain" in units of torque per unit of air pressure. Brake effectiveness or gain is the pertinent mechanical property that is used in evaluating wheel-unlocked stopping performance.

In addition to effectiveness, two other pertinent mechanical properties of brakes are considered to have an important influence on braking and steering performance (see Figure 2.4.1). These properties are related to the timing of changes in brake torque and the management of brake temperature during long or repeated brake usage. The times for brake torques to fall and rise

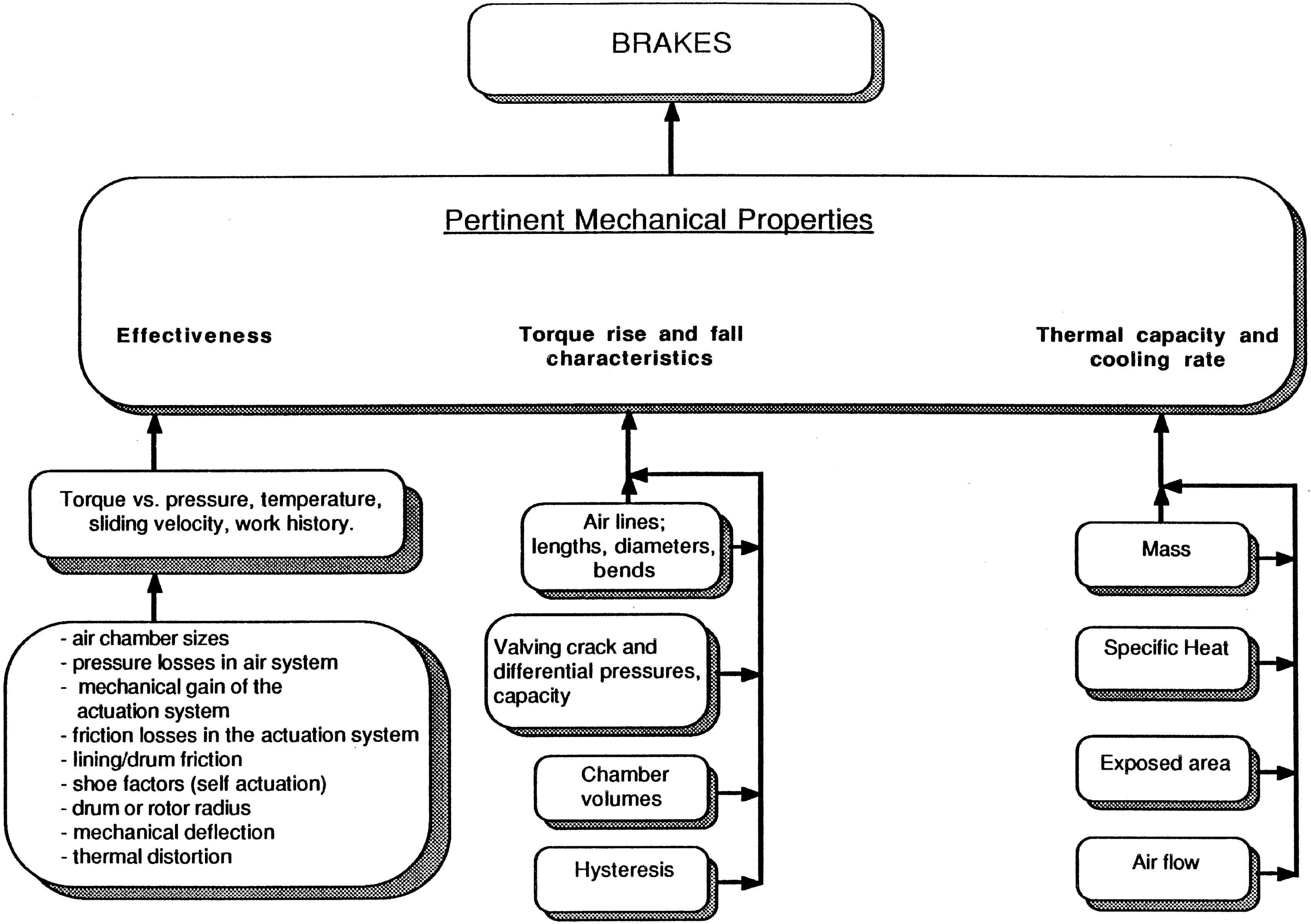


Figure 2.4.1 Pertinent mechanical properties of brakes

determine the rapidity with which the driver can correct for overbraking. The thermal capacity and cooling properties of the brakes determine the control speeds that are suitable for mountain descents on grades of various lengths and slopes.

2.4.1.1 Effectiveness. Instantaneous brake torque depends not only upon air pressure, but also, temperature, sliding velocity between the friction surfaces, and work history (past usage). Figure 2.4.2 illustrates a typical brake-torque time history as obtained from a "constant" pressure stop. In the beginning, torque increases rapidly after the pressure is increased from zero to a constant value. Then the torque decreases as the interface temperature increases in the middle of the stop. The interface temperature falls at the end of the stop when the sliding speed has decreased to the point that the heat flow into the brake is not large enough to maintain the elevated temperature at the interface. Towards the end of the stop, the brake torque again rises due to a complicated interaction of temperature effects and sliding velocity. The average torque during a stop of this nature may be only a rough approximation to the instantaneous torque. Nevertheless, the relationship of average torque to pressure provides a first-order approximation that is used for describing the effectiveness of brakes.

As indicated in Figure 2.4.1, many factors, related to the air system, the actuation mechanism, lining friction, shoe or pad geometry, and the handling of mechanical and thermal stresses in the design of the brake, contribute to the overall torque characteristics of commercial vehicle brakes. Each of these factors influences the nature of the general effectiveness function in which torque depends upon pressure, temperature, sliding velocity, and work history. The influences of these factors depend upon the conditioning of the brake through its "history" of work. By working the brake, the linings become "bedded" into their mating surface as the lining wears. Also, the frictional and mechanical properties of the lining material can be altered by operation of the brake at high temperatures.

In trying to develop predictable and consistent brakes (sometimes referred to as "brakes with good definition") with low cost and weight, manufacturers have employed various designs, for example, s-cam and wedge drum brakes, and disc brakes. The net result has been brakes that serve well as devices for absorbing energy, but brakes whose torque characteristics have only been defined approximately under very restricted and carefully controlled operating conditions.

2.4.1.2 Torque rise and fall characteristics. The dynamic response of an air-actuated brake depends upon the quickness with which air signals can be delivered, the rate that brake chambers can be filled or emptied, and delays caused by hysteresis (friction in the brake mechanism). Since

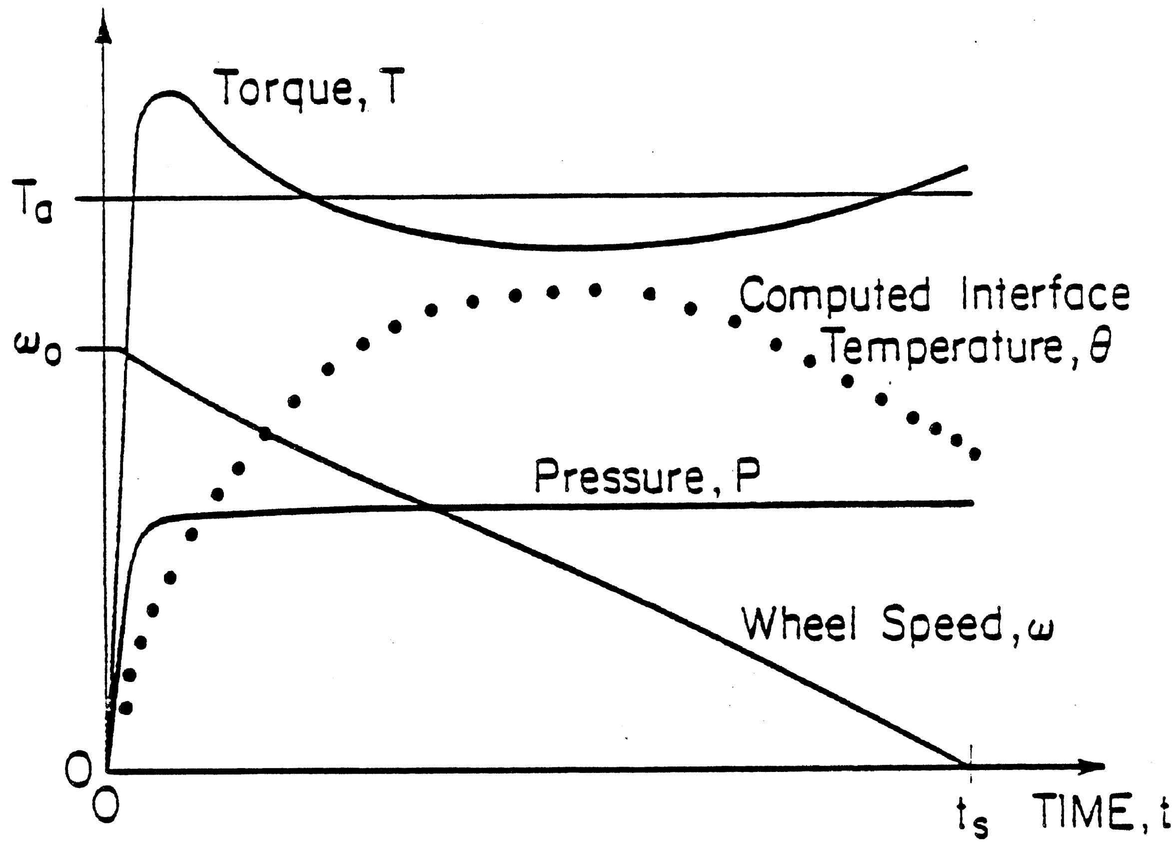


Figure 2.4.2 Data from a spin-down dynamometer test

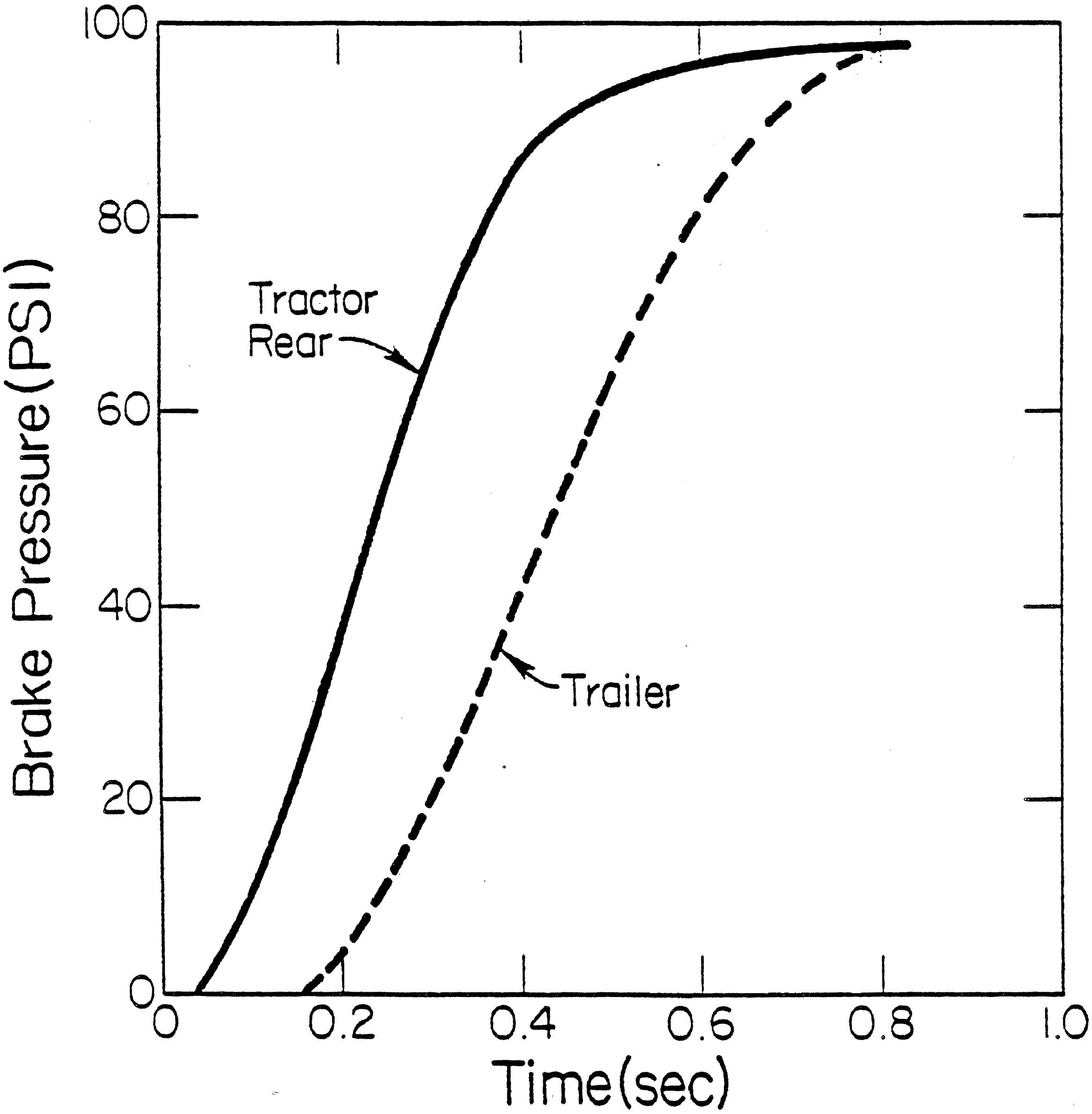
the moving parts of the brake have small weight compared to the forces applied to them, the dynamics of their motions are negligible compared to the response times of the air delivery system.

Air systems consist of compressors, reservoirs, lines, valves, actuators (brake chambers), connectors, and possibly pressure control devices such as proportioning or antilock systems. The characteristics of most of these elements combine to determine the time response of brake torque to changes in pressure. Figure 2.4.3 illustrates the time lag and rise time features characterizing the nature of the response of brake chamber pressure to an increase in treadle pressure. These characteristics depend upon the diameters and lengths of the air lines, the crack pressures and response characteristics of the various valves, and the volumes of the brake chambers. Small diameter lines with many bends tend to increase response time due to an increase in the resistance to air flow. On the other hand, large diameter lines and large volume chambers take longer to fill than smaller volumes. Hence there is a tradeoff between resistance to flow and volume. (The response shown in Figure 2.4.3 is faster than that of many tractors which would typically reach 60 psi at 0.35 sec.)

Many brake systems are equipped with quick release valves so that pressure can be rapidly reduced. However, if the brake is hysteretic, the torque will not decrease until the pressure has dropped considerably. A slow fall rate in brake torque can be detrimental to directional stability if the front brakes on a tractor or straight truck release much more quickly than the rear brakes on the unit. If the vehicle has been overbraked and then, upon release of the brakes, the front tires develop side forces before the rear tires, the vehicle will tend to jackknife or spin out. The jackknife response may be rapid enough that even a fraction of a second delay in the return of side force at the rear tires can have a significant effect.

2.4.1.3 Thermal capacity and cooling rate. These properties are somewhat different than the others in that they contribute to a situation in which vehicle perfomance is degraded by an overload of a component. Nevertheless, the brake will normally recover after it has been overheated. For heavy trucks the thermal capacity of their brakes is such that these vehicles should proceed at low speed down long, steep grades if they are heavily laden. The cooling rate at which heat can be convected away is a significant factor in situations in which the brake is used for minutes at a time. It has an insignificant effect in brake applications lasting for a few seconds such as in a typical or emergency stop.

The factors that affect the long-term heating and cooling of the brake are its mass, the specific heat of the energy-absorbing material, and the convection factor which is a function of the air flow over the exposed area of the brake. These factors combine to determine the bulk



Source: NHTSA (VRTC) measurements

Figure 2.4.3 Brake pressures versus time - tractor-trailer combination temperature of the brake as a whole. Empirical results show that the cooling rate is approximately a linear function of velocity for typical drum brake installations. Also, the bulk temperature varies in an exponential manner at constant velocity. The time constant, τ, involved in temperature changes, is related to the physical properties of the brake by the following simplified equation:

$$\tau = m\, c_p/h(v)$$

where,

m is the mass of the brake

c_p is the specific heat, and

h(v) is the cooling coefficient.

(Typical values for τ and h, expressed in convenient units, are presented in Section 2.4.3.3.)

2.4.2 The Importance of Brake Properties to Vehicle Maneuvering Behavior. The maneuvers considered with respect to braking are constant deceleration, braking while turning, and mountain descents. Each of these maneuvers emphasizes a different aspect of braking performance. In constant deceleration braking, the issue is the proportioning or balance of braking torque (effectiveness) from brake to brake. The timing of the rise and fall of torque is important in braking while turning and other transient braking maneuvers that are likely to require modulation of the braking effort in order to maintain directional control. The heat transfer and thermal properties of brakes are challenged in long, steep mountain descents. Table 2.4.1 provides a concise summary of the importance of the pertinent mechanical properties of brakes on vehicle dynamic performance in the subset of maneuvers that involve braking.

Since it is the balance of braking force amongst the various wheels that determines braking efficiency in a constant deceleration situation, one cannot say whether an increase or decrease in the effectiveness of a particular brake would be the better choice. Ideally, the braking force should be proportional to the load carried by that wheel. Currently, trucks in the United States tend to have their brakes proportioned according to their static loads when the vehicle is fully laden (at gross axle weight ratings). This practice leads to poor proportioning (low efficiency) when the vehicle is empty. In Europe, load-sensing proportioning of air pressure is used to limit these effects. In the United Kingdom, antilock braking is allowed in place of load-sensing proportioning in semitrailers.

Furthermore, load transfer from the rear to the front of a vehicle during deceleration causes front-wheel loads to increase and rear wheel loads to decrease. To compensate for this situation, the front effectiveness may be increased and the rear effectiveness may be decreased. The practice of disconnecting front brakes (which will not be allowed under the new BMCS rules) runs counter to this, with the result that the vehicles without front brakes have a greater potential for jackknifing or spinning than those vehicles with front brakes proportioned according to the instantaneous vertical load. On the other hand, if the front brakes lock the front wheels, steering control is lost. Clearly, the balance of braking effectivenesses is important in emergency braking situations.

Table 2.4.1

The Importance of the Pertinent Mechanical Properties of Brakes in Maneuvers Involving Braking

Pertinent Mechanical Properties of Brakes	Constant Deceleration Braking	Braking while Turning	Mountain Descents
effectiveness	Hi	Hi	(Could be High)
torque rise and fall characteristics	-	Hi	-
thermal capacity and cooling	-	-	Hi

The effectivenesses of the brakes have a "high influence" on the outcome of severe braking while turning maneuvers (see Table 2.4.1). In this situation it is again the balance of brake torques that is the important issue. If a particular brake is asked to provide a braking force that is larger than the vertical load and tire/road friction can support, the associated wheel will lock causing directional control problems. Since this is a "systems" problem rather than a component problem, it is not possible to state a simple generality for the effectivenesses of individual brakes—other than to say that their effectivenesses should be proportional to the instantaneous loads that they are carrying.

Brake torques should rise rapidly and uniformly throughout the braking system. Timing differences between the various brakes can be disconcerting to the driver and in extreme situations may cause the vehicle to be directionally unstable. These differences are most important in maneuvers involving steering because the loss of side force due to overbraking will have the greatest influence in these situations.

If one of the brakes has a much longer delay than the others in the reduction of brake torque, the side force capability of the associated tire will not return at the same time as it does for the other tires. This can cause a moment imbalance that is disturbing to the driver and, if the road is slippery, can lead to momentary loss of control. *Large differences in the timing of brake release (due to long delay times at particular brakes) cause large directional disturbances in vehicle heading when the vehicle is in a turn.*

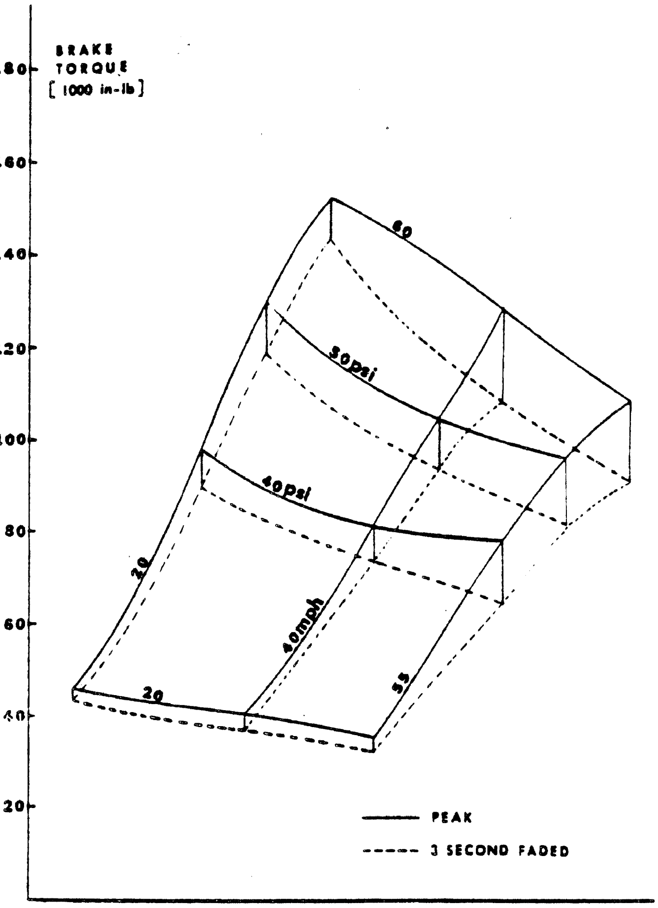
The mountain descent tests the thermal properties of the vehicle's brakes. The temperature of a brake is a measure of the internal energy stored in the brake. If the thermal capacity is lower, the temperature will be higher for the same amount of stored energy. During a long mountain descent, the brake stores energy internally and it also dissipates energy or else it would burn up. *The combination of high thermal capacity and adequate cooling through convection are needed to prevent high temperatures leading to brake fade and possibly fires.*

Since the brakes are applied at a low level to maintain speed on a grade, a pressure balance is needed to keep certain brakes from doing most of the work and overheating. This pressure balance can be attended to by matching the chamber pushout pressures and valving crack and differential pressures throughout the brake system. (This assumes that differences in work balance between axles due to brake gain and hysteresis are not factors.)

2.4.3 Typical Values of the Pertinent Mechanical Properties of Brakes. This section presents a brief summary of the range of values for the air brakes employed on heavy trucks. Typical values of torque rise and fall characteristics and heating and cooling properties are also stated.

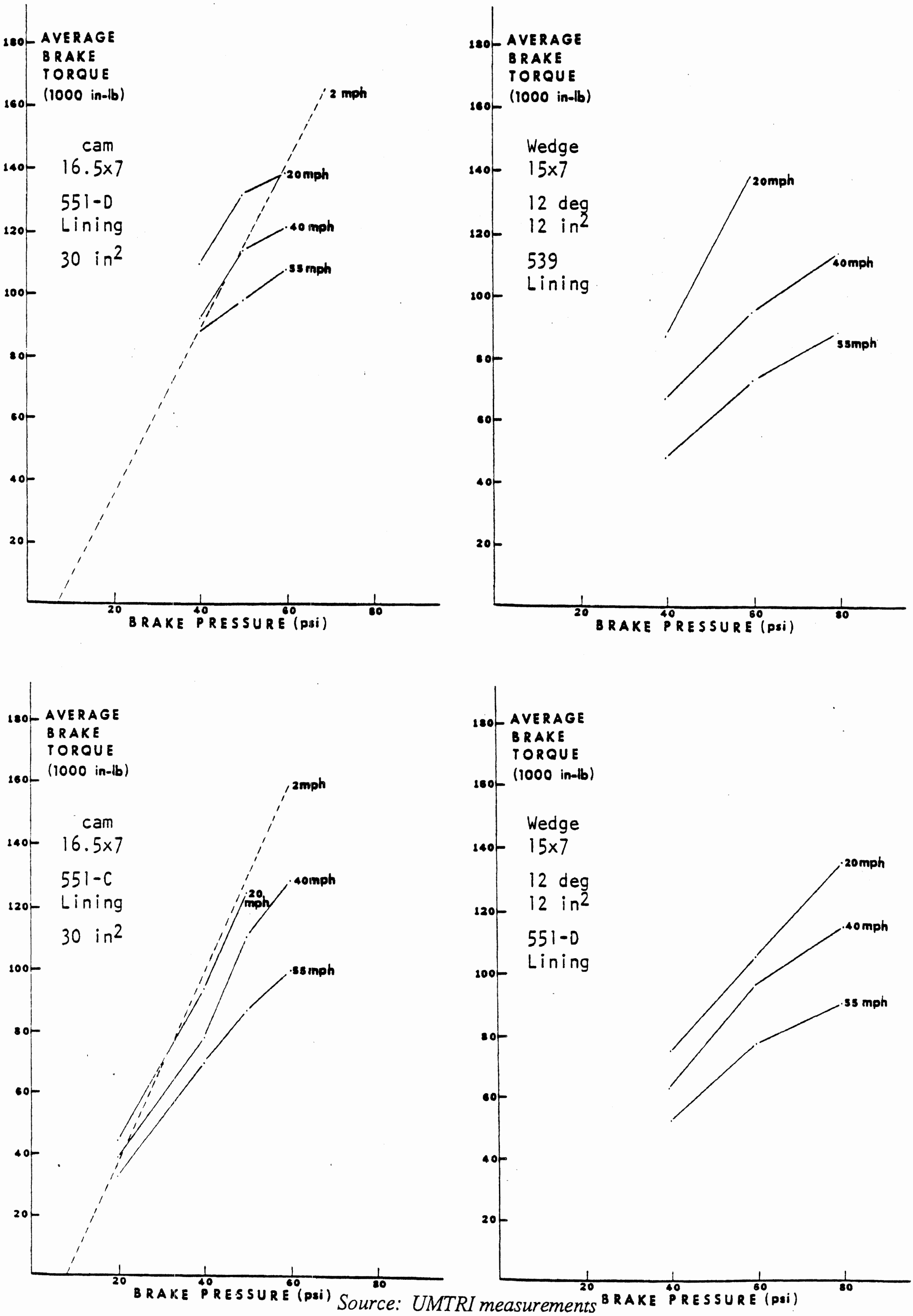
2.4.3.1 Brake gain (effectiveness). Brake torque characteristics vary with initial velocity, pressure, and temperature-fade during a stop (see Figure 2.4.4). These independent variables have different amounts of influence, depending upon the type of brake (see Figure 2.4.5). In order to summarize this diversity of torque characteristics, the average torque during a simulated stop from 50 mph will be used. (The "simulated stop" is performed with an inertial dynamometer.)

Figure 2.4.6 illustrates the range of brake gains typical of heavy-vehicle brakes currently in service in the U.S. The gains of front brakes are lower than those of rear brakes by a factor of 1/2 to 2/3. The tractor rear brakes and the trailer brakes cover the same range of gains, but trailer brakes sometimes have more "power" in terms of longer slack arms and larger air chambers than



Source: UMTRI measurements

Figure 2.4.4 Brake torque characteristics



Source: UMTRI measurements

Figure 2.4.5 Influence of velocity on average brake torque

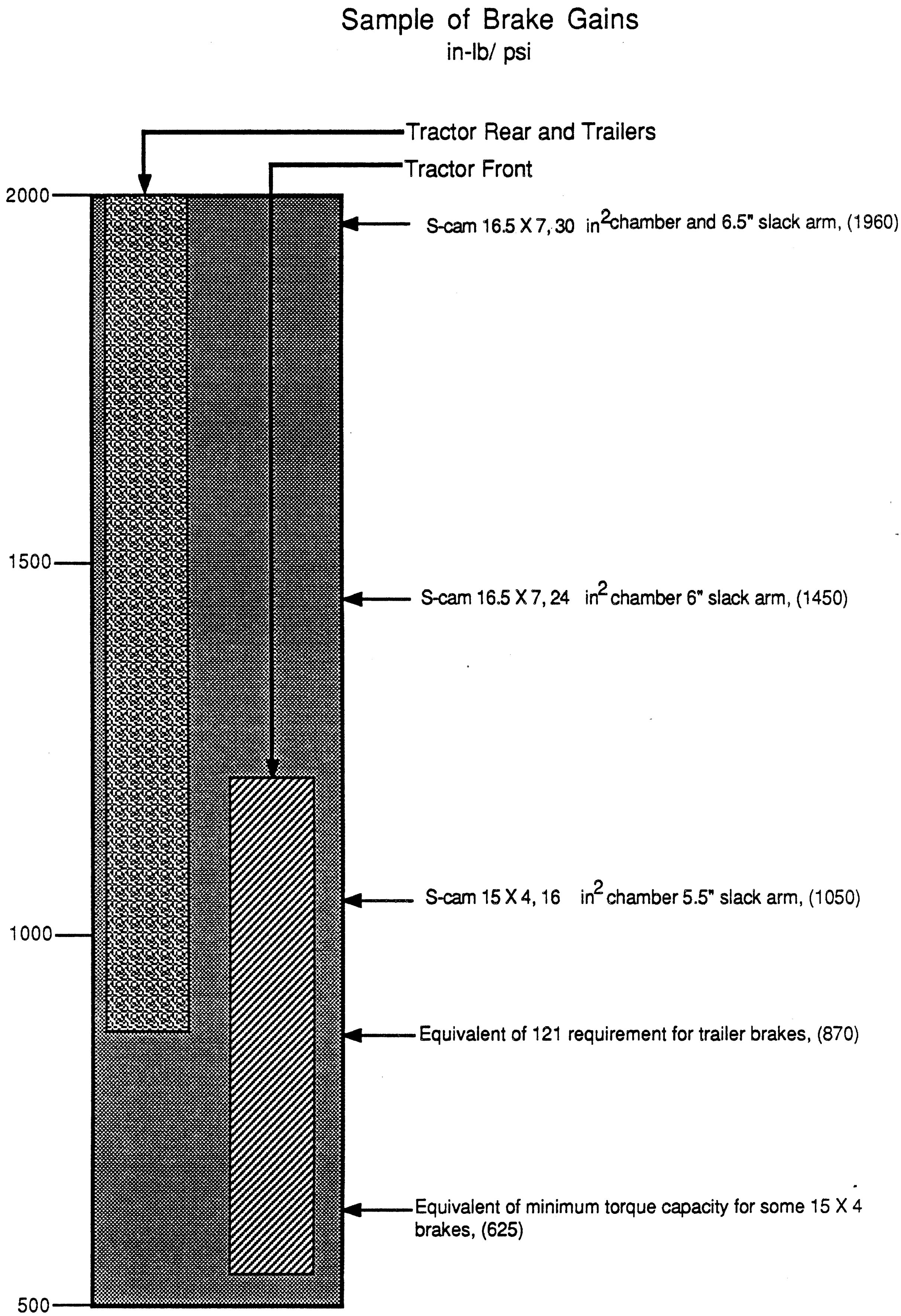


Figure 2.4.6 Estimates of brake gain approximating effectiveness at high pressure and 50 mph initial velocity

those installed on the tractor. The minimum retardation required in FMVSS 121 sets the lower bound for trailer brakes and a correspondingly equivalent gain factor is indicated in Figure 2.4.6. This lower bound is roughly equivalent to the European specification for semitrailer brakes.

2.4.3.2 Torque rise and fall characteristics. Federal Motor Vehicle Safety Standard 121 specifies brake actuation times of 0.45 sec for trucks, 0.35 sec for converter dollies, and 0.3 sec for trailers. These are likely to be faster than times found in service if maintenance is not excellent, but they are representative of the capabilities of well adjusted brake systems. Brake release times are specified in 121 to be 0.55 sec for trucks and 0.65 sec for trailers and dollies.

2.4.3.3 Thermal capacity and cooling rate. The thermal capacity of the brake depends upon the mass and specific heat of the material heated. Typical values for brake weights are 65 lb for front drum brakes and approximately 103 lb for tractor-rear or trailer drum brakes. These values, along with empirically determined convection coefficients [4], lead to the following factors that may be used to estimate average brake temperatures during mountain descents:

heating/cooling time constant, τ, = $1/(1.23 + 0.0256v)$ hrs.

where, v is velocity in mph.

cooling coefficient, $h(v) = 0.1 + 0.002v$ hp/degree Fahrenheit

These quantitities provide the mechanical parameters needed for computing bulk temperatures during mountain descents at constant velocities. They have been determined from experiments with a particular vehicle, but they have proven to be useful for making first order estimates of typical brake temperatures. Specifically, the following equation can be used to predict brake temperatures during a descent of a fixed grade at a constant velocity [4]:

$$T = T_i e^{-t/\tau} + ((HP_B/h(v)) + T_a)(1 - e^{-t/\tau})$$

where T = temperature in degrees Fahrenheit at time t in hours
T_i = initial temperature
HP_B = braking horsepower
T_a = ambient temperature
$h(v)$ = cooling coefficient.

Braking horsepower is determined by the product of brake torque and drum speed. Since the cooling coefficients and time constants may be roughly comparable throughout the braking system of a typical truck,the brake that provides the greatest torque (the most effective brake at low pressure) will usually be the hottest brake.

2.5 Frames

2.5.1 The Mechanical Properties of Frames. On a truck, the frame is the primary load-carrying member that spans between the support points of the front and rear axles. The frame also serves as the foundation structure to which most major components are attached. Utility vans (in the medium truck class) are about the only exception to this rule by virtue of their single piece body structure, which may also be designed as a load-carrying structure.

Truck frames are commonly of ladder-type construction, consisting of C-channel side rails running the length of the vehicle, with various types of cross members that vary in shape and type according to their function. A typical frame for a heavy truck is shown in Figure 2.5.1. The frame is designed to provide adequate strength to resist the vertical bending moment produced by the loading. The strength in the vertical direction derives from the material and section of the C-channel side rails. The frame's rigidity in other directions is determined by the number, type, and location of the cross members. While the frame must have adequate stiffness to provide support to the components which are attached, frame design practices normally strive for an overall torsional flexibility to minimize stresses which limit its service life.

2.5.2 Properties Important to Cornering and Braking. The torsional stiffness of the frame about its longitudinal axis is the only mechanical property identified as having influence on cornering or braking behavior (see Figure 2.5.2). Its influence is primarily limited to cornering maneuvers, as indicated in Table 2.5.1.

In cornering, load is transferred from the inside to the outside wheel sets. Because tire cornering stiffness is sensitive to load, the lateral load transfer causes the front and rear axles to lose cornering force. A greater loss on the front axle favors plough-out (an understeer response), while greater loss on the rear axle(s) favors spin-out (an oversteer response).

The lateral load transfer from the inside to the outside wheels is due, in part, to moments imposed on the axle through the suspension. Neglecting the acceleration forces acting on the unsprung mass (which are relatively small), a moment balance on an axle/suspension system yields the following equation for the difference in load between the inside and outside wheels:

$$\Delta F_z = K_f \,\text{ø} / t + M\,(U^2/R)\,h_r / t$$

where,

ΔF_z = difference in load between the outside and inside wheels

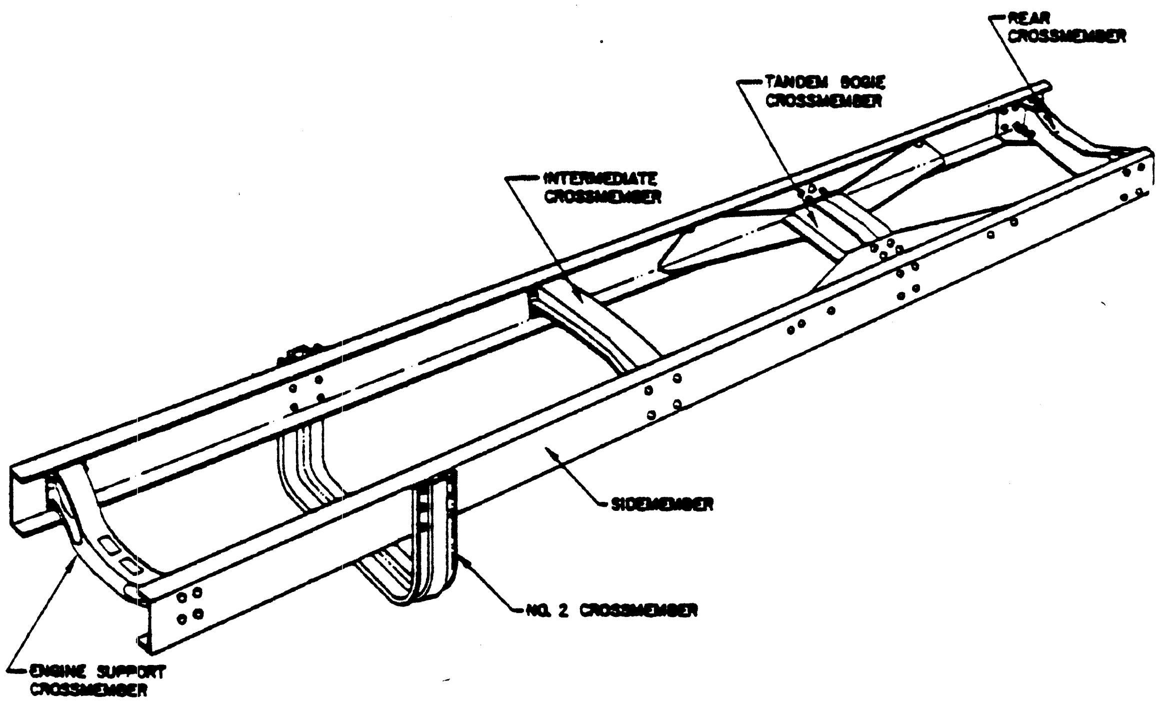


Figure 2.5.1 Typical highway truck frame

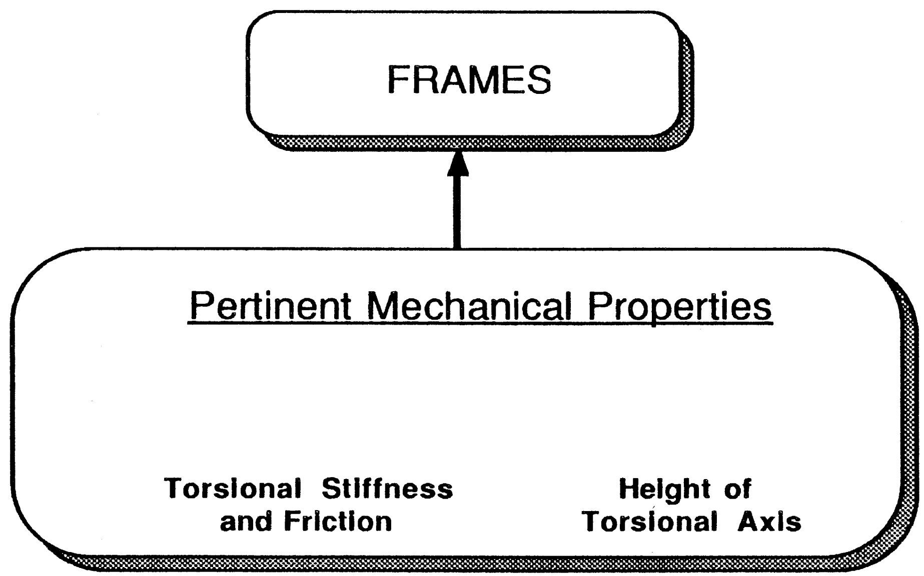


Figure 2.5.2 Pertinent mechanical properties of frames

Table 2.5.1

The Importance of the Pertinent Mechanical Properties of Frames
on Vehicle Dynamic Performance

Pertinent Mechanical Property of Frames	Low Speed Tracking	Hi Speed Tracking	Roll Stability	Yaw Stability	Response Time	Rearward Ampl	Braking Efficiency	Transient Braking	Downhill Braking	Response to Disturbances
Torsional Stiffness	Low	Low	-	Low	-	-	-	Low	-	-

K_f = suspension roll stiffness

ϕ = frame roll angle

M = mass carried on the axle

U^2 / R = lateral acceleration at that point on the vehicle

h_r = suspension roll center height

t = lateral spread between the inside and outside wheels

Thus it is seen that the lateral load transfer derives from two mechanisms:

1) Roll of the frame acting through the roll stiffness of the suspension

2) The lateral force applied to the suspension at its roll center.

If the frame on a truck were perfectly rigid, the lateral load transfer due to frame roll at each of the suspensions would be in exact proportion to the suspension roll stiffnesses. With torsional compliance, the roll angle above the front and rear suspensions is not necessarily equal; thus the lateral load transfer due to this mechanism requires more complicated modeling for accurate prediction. The sprung mass of a truck derives from a series of major components distributed along its length. In cornering, these produce a roll moment that is distributed to the front and rear wheels in proportion to the stiffness from each mass element to the appropriate suspension.

The importance of the truck frame as an influence on directional response has only been recognized in recent years. Frame torsional compliance has been included in the more comprehensive computer simulation models [8,9], although it would be inappropriate to say that its significance is fully understood at this time. Experimental measurements of the influence of frame stiffness on understeer have been examined on only one vehicle in the published literature [10]. *This study found that increasing the frame stiffness produces an understeer effect on a COE tractor pulling a flat-bed trailer.* The compliance of the trailer frames for typical van and flat-bed trailers was not particularly important to directional response. The influence of tractor frame compliance, however, was small due to the low roll stiffness of the tractor front suspension. Only when the front suspension roll stiffness was significantly increased was the understeer effect great enough to keep the tractor in the understeer condition throughout its cornering range.

2.5.3 Typical Values of These Mechanical Properties. Although the torsional compliance of a frame is commonly measured by frame builders [11], the actual stiffness achieved on a fully dressed tractor is usually much higher. The addition of suspension cross members, fuel tanks, cab, battery boxes, and other appurtenances adds significantly to the stiffness. The stiffness will vary along the length of the tractor due to presence of cross members and the rigidity contributed by miscellaneous off-frame components.

Measurements of the torsional compliance on fully dressed tractors have been attemped several times at UMTRI [12]. The first attempt at measurements on a tractor were rather crude and did not accurately measure the torsional compliance properties of interest. The vehicle was a COE tractor, 142 inches in wheelbase, with a 10-inch bolted frame. The value obtained is only an approximation, and is probably an underestimate due to the simplicity of the method. Typically, the measurements show a fairly linear stiffness enclosing a hysteresis loop that is 10,000 to 20,000 in-lb in magnitude. From the measurements, it is estimated that the torsional stiffness between the front and rear suspensions is in the range of 10,000 to 15,000 in-lb/deg. On the second tractor measured, a more valid method was used with more attention given to the detail of the process. It was a COE tractor, 152 -inches in wheelbase, with a bolted frame. The torsional stiffness between the front and rear suspension points, measured with moment inputs at those points, was 17,100 in-lb/deg. These data are summarized in Figure 2.5.3. There are few guidelines for estimating appropriate torsional stiffness values for a tractor. The values in Figure 2.5.3 may be used as reference points for estimating properties of other vehicles. The compliance (the inverse of the stiffness) would be expected to increase proportionally with the length of the tractor wheelbase, thus estimates for vehicles with wheelbases that are significantly different should be adjusted accordingly. Because of the roll compliance typical in front suspensions of American trucks and tractors, frame compliance does not have a large influence on directional response. Consequently, the exact value selected may not have a great influence on performance. However, on certain European trucks, auxiliary roll stiffness is incorporated into the front suspensions. With the higher suspension roll stiffness that may result, frame stiffness could play a greater role in limit performance.

Incorporating the effects of frame torsional compliance into models for estimating its effect on directional response has only been done using comprehensive computer simulation models [8]. For that purpose, a second parameter is required. The torsional stiffness, as described above, is one input. The other is the vertical position of the torsional axis. This latter parameter is nominally the elevation at the vertical midpoint of the frame. The top-of-frame (TOF) dimension is established by SAE standards for fifth-wheel height. Thus manufacturers of medium and heavy-duty trucks strive for a TOF that is 40 inches above the ground, varying plus or minus a couple of

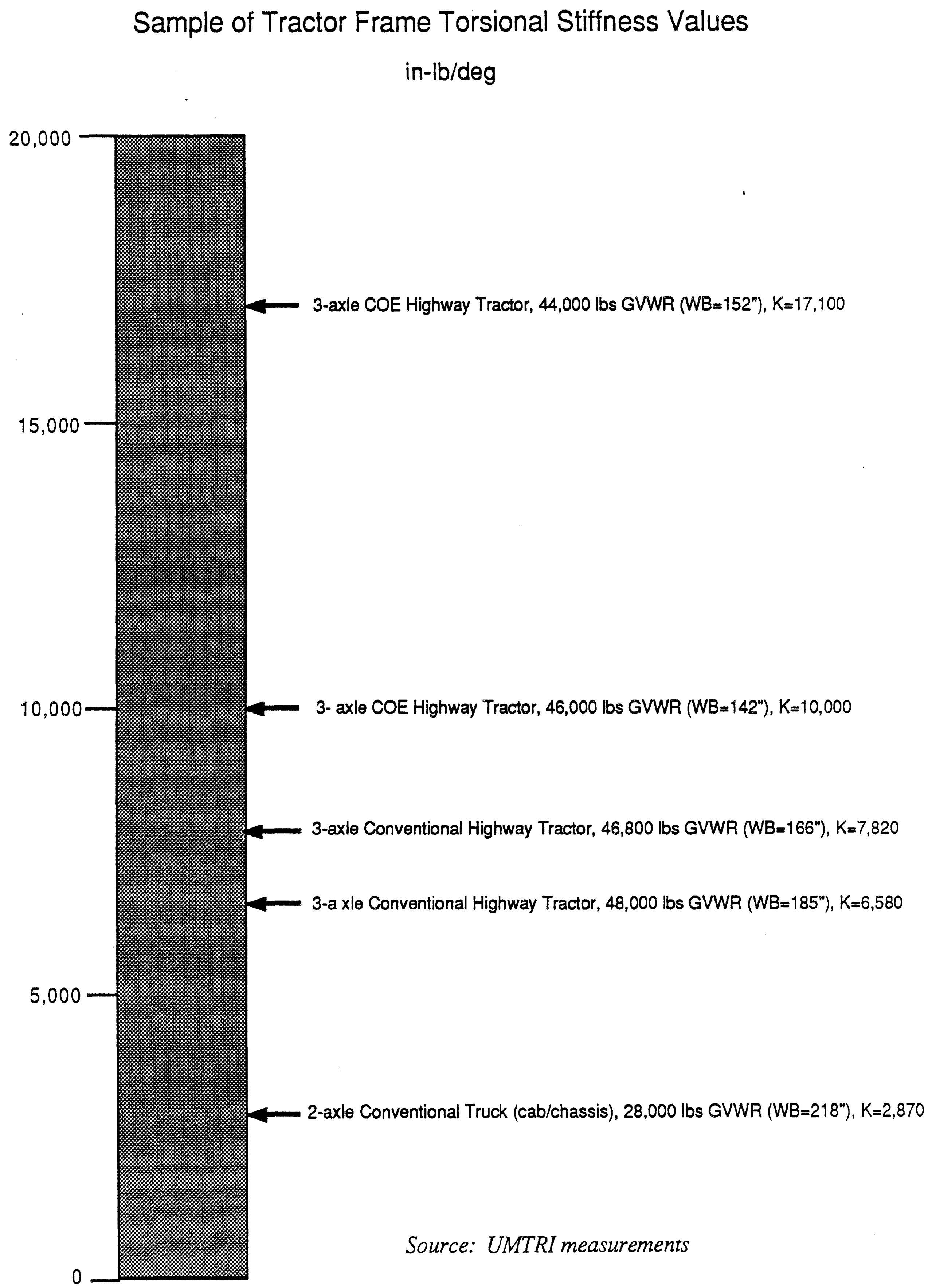


Figure 2.5.3 Tractor frame torsional stiffness

inches depending on load, suspension, and tire size. Frames range from 9 to 10 inches in section height, thus the midpoint of the frame (the torsional axis location) is nominally 35 inches above the ground ($\pm$ 2 inches). As used in the Phase 4 simulation program, the location is entered as a vertical distance above the rear suspension reference point (its roll center height), and should be entered accordingly.

2.5.4 Methods for Experimental Measurement of Properties. Direct measurement of frame torsional stiffness is a complex process. One end of the vehicle must be elevated and supported such that it is free to roll. A moment is applied at the suspension points while the roll angle is measured at these same locations and other points of interest. One difficulty in valid measurement comes from the large masses mounted off of the frame, particularly the cab and rear axles. A roll angle at these points causes an offset of the center of gravity of these components, which alters the actual moment sustained in the frame. Care must be taken to prevent or compensate for these effects. Typically, the frame at the front axle of the vehicle may be constrained at zero roll angle to avoid this influence from the cab and engine, while the rear axles are completely removed from the vehicle to prevent their influence on the measurement as the rear of the frame is twisted. A second problem in measurement is the application of the torsional moments. Ideally, they must be applied at the suspension locations to most closely duplicate the mechanics by which the frame compliance affects the vehicle in cornering.

2.6 Hitches

This brief section discusses hitching devices that are commonly employed in commercial vehicles in the United States. There may be more to say later, if advanced dolly concepts and hitching devices become more prevalent here.

2.6.1 Descriptions of the Pertinent Mechanical Properties of Hitches. The pertinent properties of a hitch are primarily the degrees of freedom of motion allowed between the bodies that are connected together by the hitch. Currently, fifth wheels, turntables, and pintle hitches are used in joining the units of combination vehicles. With regard to vehicle dynamics, each of these devices provides its own types of constraints on rotational motion.

A conventional fifth wheel allows a semitrailer to yaw with respect to the unit towing it (either a tractor or a converter dolly). When a semitrailer is lined up with its tractor, the fifth wheel allows pitch motion but it does not allow roll motion between the tractor and the semitrailer. However, if the semitrailer is at 90 degrees to the tractor, the fifth wheel restrains the semitrailer in pitch but not in roll. Due to the fact that the landing plate on the fifth wheel is mounted on the tractor, it transmits moments about the roll axis of the tractor. If the roll axis of the semitrailer is not parallel with that of the tractor, the moment transmitted between the tractor and the semitrailer has components about both the roll and pitch axes of the semitrailer. The magnitudes of these components depend upon the articulation angle between the tractor and semitrailer, with the roll component being proportional to the cosine of the articulation angle and the pitch component being proportional to the sine of the articulation angle.

In contrast, a turntable allows only yaw motion between a towing unit (usually a dolly) and the semitrailer. When a turntable is employed, the dolly is referred to as a "fixed dolly," and the dolly and semitrailer are usually "married" to form a full trailer. A fifth wheel is used in "converter" dollies that are employed in flexibly converting semitrailers into full trailers. Since the turntable restrains both towing unit and semitrailer in both roll and pitch, towing unit pitch moments are transmitted between these units.

A pintle hitch provides no restraints on rotational motion. It simply connects two units together and they are free to yaw, roll, and pitch with respect to one another.

In fixed dollies with turntables, a pitch hinge is used to prevent pitch moments from causing vertical loads to be applied to the next foreward unit at the pintle hitch. With a fixed dolly, pitch moments due to braking are not transfered to the unit ahead of the full trailer. On the other hand, converter dollies with short drawbar lengths transmit significant portions of pitch moments due to braking. This pitch moment is reacted primarily by an increase in vertical load on the pintle hitch.

2.6.2 The Importance of Hitch Properties to Vehicle Maneuvering Behavior. The most important mechanical property of a hitch is whether it transmits a roll moment. *The fifth wheel (or turntable) has an advantage over a pintle hitch in that the fifth wheel allows adjacent units to aid each other in preventing rollover.* This is particularly significant in dynamic situations in which first one unit and then the next is moving laterally such as in an emergency lane change.

The emergency braking performance of vehicles employing fixed dollies is different from that of vehicles employing converter dollies because of the fore/aft load transfer occurring between the converter dolly and the unit preceding it. Ideally, brake proportioning might be controlled to compensate for the load transfer. Load-sensing proportioning systems and/or antilock systems are needed to do this efficiently for vehicles in both their loaded and unloaded states, regardless of the type of dolly involved.

The excessive free-play in a hitch can degrade the performance of the vehicle. Free-play in pintle hitches is known to cause hunting oscillations of full trailers. These oscillations are disconcerting to observe and subject the vehicle to undesirable motions. Air-loaded pintle connections are sometimes used to eliminate this type of free-play. Another type of free-play occurs in fifth wheels when a vehicle has rolled to a large angle in a severe maneuver. For some vehicles this free-play lowers the vehicle's rollover threshold, but for many vehicles this free-play occurs after the vehicle has surpassed its rollover threshold and is in the process of rolling over.

2.6.3 Values of the Pertinent Mechanical Properties of Hitches. In analyzing vehicle performance, the equations of motion have to be chosen to represent the constraints imposed by different types of hitches. This means that hitches are different from other components in that there are no values to assign to their most important characteristics, but the form of the analysis changes depending upon the type of hitch involved.

3.0 GEOMETRIC LAYOUTS AND MASS DISTRIBUTIONS OF MAJOR UNITS

3.1 Geometric Layout

3.1.1 Geometric Layout Parameters The strictly geometric parameters which are classified as "layout parameters" locate the axles and hitch couplings on the vehicle. These dimensions, in turn, have profound importance for the dynamic response of vehicles simply because all external forces and moments to each vehicle unit are applied through the tires and the hitch couplings. In the discussion below, the dimensions of interest will be organized according to the longitudinal, lateral, and vertical placement of both axles and hitches.

Shown in Figure 3.1.1 is a diagram illustrating the pertinent mechanical properties related to geometric layout. We see that the parameters of primary importance are covered under four items, namely:

• Wheelbases -- That is, the overall spread between axles, on a power unit, and between forward hitch point and the axle center on a semitrailer or dolly. In all cases, wheelbase dimensions are measured either to the center of single axles, or to the center of the spread dimension on a two-axle tandem.

• Track Width -- Width-related dimensions are measured from the center of an axle to the wheel centerlines. These dimensions locate either the centerline of single tires, such as on steering axles, or the center of the dual tire pair with an accompanying dual spread dimension.

• Longitudinal Location of Hitches -- The hitch elements are classified either as fifth-wheel coupling or pintle-type hitches. Longitudinal locations are specified in terms of distance from axle centers, although most mathematical models of vehicle response require eventual conversion to distances relative to the center of gravity.

• Elevation of Hitches -- The vertical location pertaining to fifth wheels is defined as the elevation of the top surface of the fifth-wheel plate above the ground. The elevation of pintle hitches describes the height of the center of the tow-bar eye, or equivalent joint center. (These heights vary with vertical load, but only by a small amount because truck suspensions and tires are typically very stiff.)

Table 3.1.1 indicates that the value for certain of these dimensions derives both from the basic design of the vehicle and from the status of certain location adjustments. The two primary adjustment devices in common use are the "slider-bogie" trailer axle assembly and the "sliding fifth

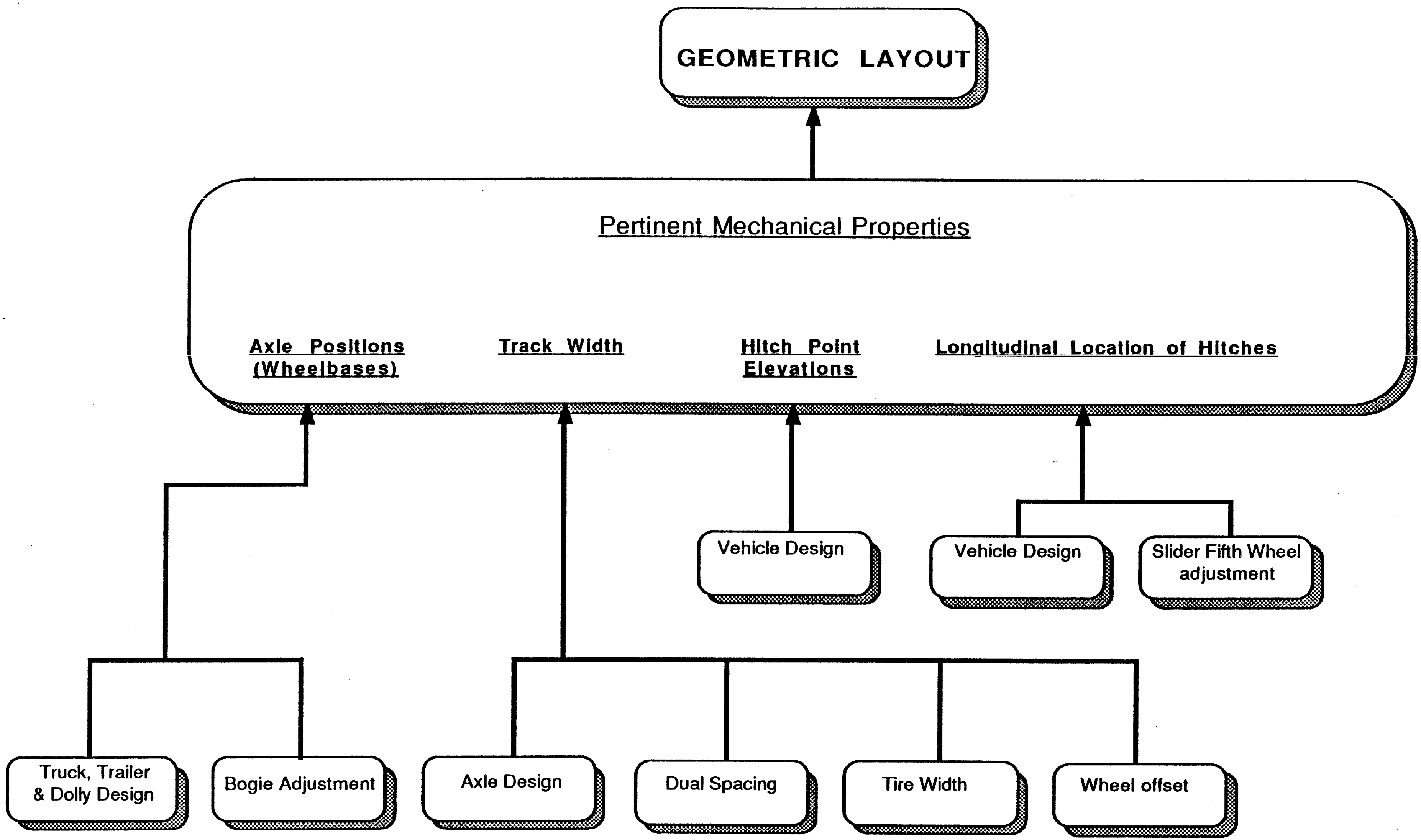


Figure 3.1.1 Factors influencing the pertinent mechanical properties of geometric layout

Table 3.1.1

The Importance of the Pertinent Mechanical Properties of the Geometric Layout on Vehicle Dynamic Performance

Pertinent Mechanical Property of Geometric Layout	Low-Speed Tracking	Hi-Speed Tracking	Roll Stability	Yaw Stability	Response Time	Rearward Amp	Braking Efficiency	Transient Braking	Downhill Braking
Wheelbases									
Truck/Tractor	Med	Low	-	Med	Med	-	Low	Low	-
Trailer	Hi	Hi	-	-	Med	Hi	Low	-	-
Dolly	Med	Med	-	-	-	Med	Low	-	-
Track Width	-	-	Hi	Med	-	-	-	Med	-
Fifth wheel offset - Tractors	Low	-	Low	Low	-	-	Med	Med	-
Pintle overhang - Trucks & Trailers	Low	Low	-	-	-	Hi	-	-	-
Fifth wheel height - Tractors	-	-	Low	-	-	-	Low	-	-

wheel." Both devices provide for substantial changes in the respective longitudinal dimensions for purposes of (a) redistributing axle loads, (b) altering ride vibrations as experienced in the cab, and (c) changing low-speed tracking characteristics. Also, the precise track width dimension on a given vehicle will be influenced by axle selection as well as by details regarding the choice of tire, wheel, and dual spacer.

3.1.2 Importance of Geometric Layout to Vehicle Maneuvering Behavior In this section, the nominal level of importance of each geometric layout parameter will be estimated. The pertinent mechanical properties shown above in Figure 3.1.1 will be broken down into the respective applications to truck, trailer, and dolly unit such that the importances of each particular parameter may be noted. Shown in Table 3.1.1 is the breakdown of estimated importance levels ascribed to each dimension.

3.1.2.1 Wheelbase of truck or tractor. The table shows that the wheelbases of trucks and tractors have a significant influence on a number of response characteristics. This observation is of special interest since commercial trucks and tractors are manufactured in a broad range of wheelbase values in order to meet the varied demands of trucking. The significant influences are as follows:

Low-Speed Offtracking -- The wheelbase of any element in a combination vehicle has a direct effect on the offtracking. The level of importance ascribed to the tractor in a vehicle combination is entered as "medium" in the table, reflecting the fact that the respective contributions of vehicle units are related to the square of the wheelbases. Since tractor wheelbase is a relatively small number in comparison with typical trailer wheelbases, its importance to the net offtracking response is moderate. *Increased tractor wheelbase increases low-speed offtracking.*

High-Speed Offtracking -- As in the case of low-speed offtracking, the tractor wheelbase, as well as that of all other units in the vehicle combination, contributes to high-speed offtracking. Since the tractor wheelbase is relatively short, again, the level of influence is low. *Increased tractor wheelbase will increase high-speed offtracking.*

Steady-State Handling Qualities (Yaw Stability) -- The wheelbase of trucks and tractors is directly instrumental in determining the yaw stability characteristic insofar as the critical velocity associated with the oversteer regime of behavior depends upon wheelbase. *Increased wheelbase increases the velocity at which an oversteer vehicle becomes yaw-unstable.* (To the extent that increased wheelbase implies more torsional compliance in the frame, frame compliance matters are pertinent here.)

Transient Turning at High Speed (Response Time) -- The transient response of the vehicle depends intimately upon the wheelbase dimension. The natural frequency in yaw will be related to both the wheelbase and to the yaw moment of inertia which generally follows wheelbase, depending upon the distribution of payload. The damping in yaw is related to the square of the wheelbase. *Increases in wheelbase typically produce a lower yaw natural frequency and increased yaw damping, depending upon loading.*

Constant Decel Braking (Braking Efficiency) -- The wheelbase of the tractor determines the magnitude of the load transfer, from rear tractor axles to front, that accrues during braking. Thus, the suitability of the brake torque distribution among tractor axles is determined, in part, by the role which tractor wheelbase plays in establishing axle loads. *Increased tractor wheelbase reduces load transfer among tractor axles during braking.*

Braking in a Turn (Transient Braking) -- The wheelbase of the tractor influences braking performance in a turn, again, insofar as it determines the amount of longitudinal load transfer between tractor axles. *Increased wheelbase will reduce load transfer among tractor axles during braking in a turn.*

3.1.2.2 Wheelbase of trailers. The wide range of trailer wheelbases seen in common commercial service is noted in Table 3.1.1 to have an important influence on a number of response characteristics, as follows:

Low-Speed Offtracking -- Trailer wheelbase influences the low-speed offtracking of the combination vehicle in relation to the square of the wheelbase. The level of influence is rated "high" since the wheelbase values of the longer trailers used in the U.S. tend to dominate the "sum of squares" outcome in typical vehicle combinations. *Increased trailer wheelbase increases low-speed offtracking.*

High-Speed Offtracking -- Trailer wheelbase determines high-speed offtracking since (a) it establishes the magnitude of the low-speed offtracking for a given turn radius and (b) it determines the outboard increment in radius associated with a given slip angle at the trailer tires during high-speed cornering. Given the cornering stiffnesses of the trailer tires, there exists an intermediate value for trailer wheelbase for which high-speed offtracking maximizes.

Transient Turning at High Speed (Response Time) -- Since the wheelbase of the trailer determines the magnitude of the "lag time" which exists between lateral motions at the fifth wheel and the yaw response of the trailer, the trailer wheelbase influences the lag in roll moments being experienced at the tractor. Accordingly, trailer wheelbase influences the response time aspects of the tractor's response which are related to lateral load transfer among left- and right-side tires on the tractor. *Increased trailer wheelbase increases yaw response times on the tractor.*

Obstacle Evasion (Rearward Amplification) -- Trailer wheelbase is a highly important parameter in determining the rearward amplification response of a combination vehicle. Longer trailer wheelbase gives rise to a more well-damped yaw response, thus tending to avoid the overshoot types of lateral motions at rear pintle hitch locations. *Increased trailer wheelbase reduces rearward amplification.*

Constant Decel Braking (Braking Efficiency) -- Trailer wheelbase is instrumental in determining the load transfer occurring between trailer axles and the fifth wheel (and thus, onto tractor axles) during braking. The extent of this load transfer will influence the balance of brake torques vs. wheel loads at the respective axle positions along the vehicle. *Increased trailer wheelbase decreases load transferred onto the tractor during braking.*

3.1.2.3 Wheelbase of dollies. The wheelbase of the dolly is defined as the distance from the center of the pintle hitch to the center of the dolly axle (or tandem). Substantial influences of this dimension on vehicle behavior are seen in the following areas:

Low-Speed Offtracking -- The dolly wheelbase plays a moderate role in the total offtracking dimension, again determined by the influence of this length, relative to those of the other units in the vehicle combination, in the sum of squares. *Increased dolly wheelbase increases low-speed offtracking.*

High-Speed Offtracking -- In a high-speed turn, the dolly wheelbase has the identical influence on offtracking as that cited above for the trailer wheelbase.

Obstacle Evasion (Rearward Amplification) -- Recent analyses [18] indicate that there is a "worst" length for dolly wheelbase, but that *the influence of dolly wheelbase on rearward amplification is low.*

Constant Decel Braking (Braking Efficiency) -- A substantial amount of load is transferred across the typically short dollies used in conventional doubles combinations during braking. This mechanism basically involves lightening the load on the dolly axle and increasing load on the preceding trailer axle. The implications of load transfer again involve the balance of brake torques and axle loads, given the ultimate desire for good braking efficiency. *Increased dolly axle wheelbase reduces inter-trailer load transfer.*

3.1.2.4 Track width. The track width of any axle has importance to the behavior of the vehicle whenever the lateral transfer of load influences overall response. The cases having a significant sensitivity of this kind are as follows:

Static Roll Stability (Roll Stability) -- The track width multiplied by the vertical load on a given axle determines the maximum level of roll moment which can be developed to resist rollover. Clearly, if the load is fixed, extensions in track width constitute a fundamental means of improving static roll stability. Also, the track width establishes the lever arm at which the "tire spring" acts. That is, that portion of the total rolling of the sprung mass deriving from tire deflection is determined by (the square of) the track width. *Moreover, increases in track width constitute a powerful means of increasing roll stability.*

Steady-State Handling Qualities (Yaw Stability) -- Since track width determines the magnitude of the load transfer which occurs between left- and right-side tires while cornering, it will in turn influence the extent to which tire cornering stiffness is varied as a function of the severity of the turn. As was discussed in the section on tire parameters, it is the balance in such cornering stiffness adjustments (due to load transfer) at the respective front and rear axle positions on a truck which determine the tendency toward oversteer and yaw instability. *Increases in rear axle track width on a truck or tractor will reduce the tendency toward yaw instability. Uniform increases at both front and rear will have little effect on such tendencies.*

Braking in a Turn (Transient Braking) -- Insofar as track width determines lateral load transfer in a turn, the right/left differences in tire load which aggravate braking-in-a-turn performance are directly influenced by track width. *Increased track width will improve braking-in-a-turn performance.*

3.1.2.5 Fifth wheel offset on tractors. The longitudinal location of the fifth wheel relative to the center of the tractor rear axle(s) simply determines the static distribution of fifth wheel load among the tractor's front and rear axles. This outcome has the following results on vehicle response:

Low-Speed Offtracking -- For all practical purposes, the magnitude of the fifth wheel offsets normally employed on tractors in line-haul service has a negligible (although beneficial) influence on low-speed offtracking. One vehicle configuration in which an offset-like dimension becomes significant to low-speed offtracking is the stinger-steered auto transporter. Such vehicles benefit substantially in their low-speed offtracking behavior from the very rearward-placed fifth wheel position. On the strength of such cases it is meaningful to state that *increased fifth wheel offset (either rearward or forward) reduces low-speed offtracking.*

Static Roll Stability (Roll Stability) -- Because the steering axle on trucks and tractors is generally rather softly sprung relative to rear axles, the placement of more load on the steering axle tends to reduce static roll stability. That is, more forward placement of the fifth wheel implies removal of some load from the rear tractor axles which are more suitably suspended for reacting to roll motions. Accordingly, *increased forward placement of the fifth wheel degrades static roll stability.*

Steady-State Handling Qualities (Yaw Stability) -- The relative magnitudes of the tire loads accruing on the respective front and rear axles of a truck or tractor will influence the basic understeer level and thus, the margin of tolerance for tendencies toward oversteer and yaw instability. Movement of the fifth wheel forward shifts more load onto the front tires and causes them to experience a net loss in cornering coefficient relative to the rear tires. Thus, *increased forward placement of the fifth wheel mildly improves the understeer quality.*

Constant Decel Braking (Braking Efficiency) -- The static load distribution among the tractor axles is, of course, at the heart of the brake torque proportioning issue. To the degree that tractor front axles are typically underbraked, braking efficiency is probably most often enhanced by placement of the fifth wheel more nearly over the tractor rear axle(s). Thus, *increased forward offset of the tractor fifth wheel generally degrades braking efficiency.*

Braking in a Turn (Transient Braking) -- Insofar as the static distribution of loads between the front and rear axle influences the balance of loads with brake torques, fifth wheel location obviously influences braking in a turn in the same manner that it influences straight-line braking.

3.1.2.6 Pintle overhang on trucks and semitrailers. The overhang dimension is defined here simply as the distance from the rear axle (or tandem) center to the pintle hitch. The few important influences are as follows:

Low-Speed Offtracking -- Any offset between a hitch and the rear axle center of the vehicle on which it is mounted will reduce low-speed offtracking. As noted with the auto transporter, above, large offset dimensions can introduce a measurably beneficial result. Thus, a relatively large rear overhang dimension, such as accrues in the case of many truck/ full-trailers, serves to reduce low-speed offtracking. *Increased overhang reduces low-speed offtracking.*

High-Speed Offtracking -- Mechanisms which reduce low-speed offtracking automatically increase high-speed offtracking. Thus, *increased overhang increases high-speed offtracking.*

Obstacle Evasion (Rearward Amplification) -- The placement of a pintle hitch far to the rear on trucks or trailers causes larger-amplitude lateral motions as input to the following trailer. Since these lead units must yaw, or rotate about a vertical axis, in order to conduct an evasion maneuver, the lateral motion at the pintle hitch is directly influenced by the length of the "lever arm" on which the hitch is fastened. Although the dimension which is immediately relevant to this process is the length from the center of gravity to the pintle location, the overhang dimension is clearly involved as a portion of the total. *Increased pintle overhang increases rearward amplification.*

3.1.2.7 Fifth wheel height. *The height of the fifth wheel is of minor significance*, but shows up in two particular mechanisms discussed below:

Static Roll Stability (Roll Stability) -- During cornering at levels of severity approaching the rollover limit, a large roll moment is transmitted across the fifth wheel coupling. If this moment reaches a large enough level, the trailer's upper coupler plate will separate from the fifth wheel plate, opening up a "free-play" or "lash" angle on the order of 2 to 3 degrees. The height of the fifth wheel determines the elevation at which this free-play angle is introduced and, thus, influences the amount of lateral motion at the trailer center of gravity due to this angle. Since all lateral motions of the sprung mass center are destabilizing in roll, decreased fifth wheel height decreases the roll stability level.

Constant Decel Braking (Braking Efficiency) -- The magnitude of the load transfer between rear and front tractor axles, during braking, is determined in part by the height of the fifth wheel. That is, this height dimension establishes the height at which the trailer's longitudinal force reaction with the tractor is developed. Increased fifth wheel height causes an increase in the load transfer between tractor rear and front axles.

3.1.2.8 Height of pintle hitch couplings. The elevation of a pintle hitch coupling, within the range of values known to commonly occur, is not seen as having measurable significance to any vehicle performance characteristics listed in Table 3.1.1. The primary reason for this insensitivity is that pintle hitches are typically placed at about the height of suspension roll centers. Thus, vehicle rolling introduces virtually no lateral motion as an input tending to steer the dolly drawbar. Of course, if one were to build a vehicle with pintle locations placed well above the suspension roll center, it is possible that significant lateral motions at the pintle hitch might be developed.

3.1.3 Presentation of Characteristic Values In this section, values of the various geometric parameters of trucks and trailers will be presented. Of course, since geometric parameters are fixed primarily by the preferences of the trucking community together with existing size and weight laws, such parameters are readily changed and do not constitute "state-of-the-art" developments such as in the case of various component properties. Thus, the presented data simply represent the state of common practice in the United States around 1985.

3.1.3.1 Truck and tractor wheelbase. Shown in Figure 3.1.2 are wheelbases of trucks and tractors in popular use. The wheelbase dimension, again, defines the longitudinal distance from the center of the steering axle to the center of the rear axle or mid-spread location on a two-axle tandem. The data indicate the following:

• Heavy-duty trucks and tractors are built in a range of wheelbases extending from approximately 118 to 272 inches. (Some "axle-back, cab-forward" tractors can have wheelbases approaching 100 inches. Wheelbases are rarely greater than 240 inches.)

• Straight trucks cover the broadest range of wheelbases for any class of power unit, from 125 to 272 inches. The "average" straight truck has a wheelbase of 198 inches. A relatively long truck seen commonly in truck / full-trailer combinations in the Western U.S. employs a wheelbase value of approximately 235 inches.

• Three-axle tractors are also constructed over a wide range of wheelbases, extending from 134 to 268 inches. Within that range, the typical Cab-Over-Engine model would employ a wheelbase of approximately 142 inches. The typical long-nose Conventional cab would employ a wheelbase of approximately 209 inches. The "average" 3-axle tractor has a wheelbase of 195 inches. (Note that the "6 X 4" designation in the figure indicates that there are a total of 6 wheel positions, 4 of which are driven). Near the long end of the wheelbase range, at approximately 268 inches, are the 3-axle tractors meant for carrying dromedary freight units, with an aft-placed fifth wheel position. Another rather unusual tractor configuration is the approximate-186-inch-wheelbase unit used in stinger-steered auto transporter combinations.

• Two-axle tractors characteristically employ relatively shorter wheelbase values, ranging from 118 to 203 inches. The "average" 4 X 2 tractor has a wheelbase of 160 inches, although a popular vehicle in this category is the 2-axle tractor with approximately 121-inch wheelbase used to pull twin-28-ft (doubles) trailers.

Aside from the specific example vehicles which have been shown in Figure 3.1.2, various special-purpose vehicles have been built with wheelbase dimensions outside of the indicated ranges. It is also worthwhile to note that many truck and tractor configurations are available in wheelbase increments every one or two inches. Thus, the "average" and specific example values of wheelbase should be viewed as illustrative rather than definitive of a very standardized practice.

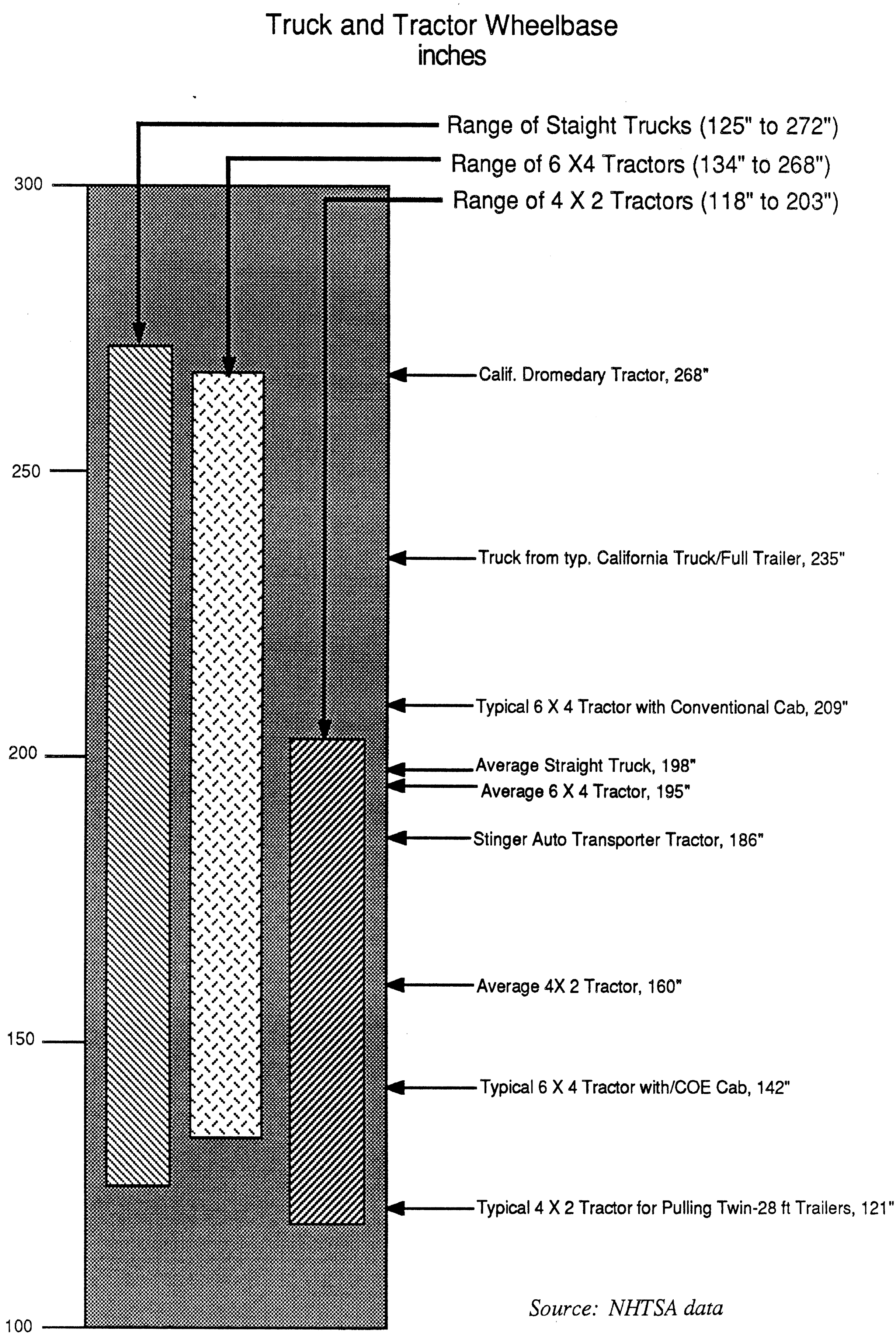


Figure 3.1.2 Truck and tractor wheelbase

3.1.3.2 Trailer wheelbase. Shown in Figure 3.1.3 are example wheelbase values for trailers in common service. (In this case, "wheelbase" is the distance from the kingpin to the centerline of the axle or axle set.) In general, one can say that typical single-axle trailers will employ a wheelbase which is equal to the overall length of the trailer minus 5.5 ft. Correspondingly, tandem-axle trailers will employ a wheelbase which is equal to the overall length minus 7.5 ft (when a slider bogie is in its aft-most position). Examples of common vehicles shown in the chart illustrate the following:

• Single-axle semitrailers are seen most commonly in 27- and 28-foot overall length, having respective wheelbase values of 21.5 and 22.5 feet.

• Full trailers in the western U.S., having single-axle dolly and trailer installations, employ wheelbase values of 18 to 20 feet.

• Common tandem-axle semitrailers exist in overall lengths of 40, 45, and 48 feet, having respective wheelbase values of 32.5, 37.5, and 40.5 feet. These trailers are commonly built with so-called "slider-bogie" installations permitting the tandem-axle assembly to slide fore and aft over some 7 to 9 feet. The indicated wheelbase values represent the rearmost position for the tandem bogie, such as is employed when the vehicle combination is loaded to near the maximum allowable gross weight. The figure also illustrates the range of wheelbase values which these respective trailers can employ as the bogie is adjusted forward from this rearmost setting.

• An example trailer wheelbase for a stinger-steered auto transporter combination (having 65-foot overall length) is approximately 28.6 feet.

Again, the reader should be cautious in applying specific numbers shown here, recognizing that trailers are built to specifications covering a virtually limitless number of specific wheelbase values. Also, it should be noted that trailers may not employ rearmost bogie locations which render wheelbase values equal to overall length minus 7.5 feet.

3.1.3.3 Dolly wheelbase (or tongue length). The distance from the pintle hitch to the center of the dolly axle (or tandem center) is termed the dolly wheelbase. For both single and tandem dollies used in the U.S. on close-coupled doubles and triples combinations, the typical dolly wheelbase dimension is approximately 73 inches. For the truck/full trailer combinations popular on the West Coast, the dolly tongue length is approximately 148 inches.

3.1.3.4 Track width. Shown in Figure 3.1.4 is an illustration of width-related dimensions for cases of (a) steering axles on trucks and tractors built to a 96-inch overall width allowance, (b)

Sample of Trailer Wheelbase- Kingpin-to-rear Axle (or Tandem) Center
feet

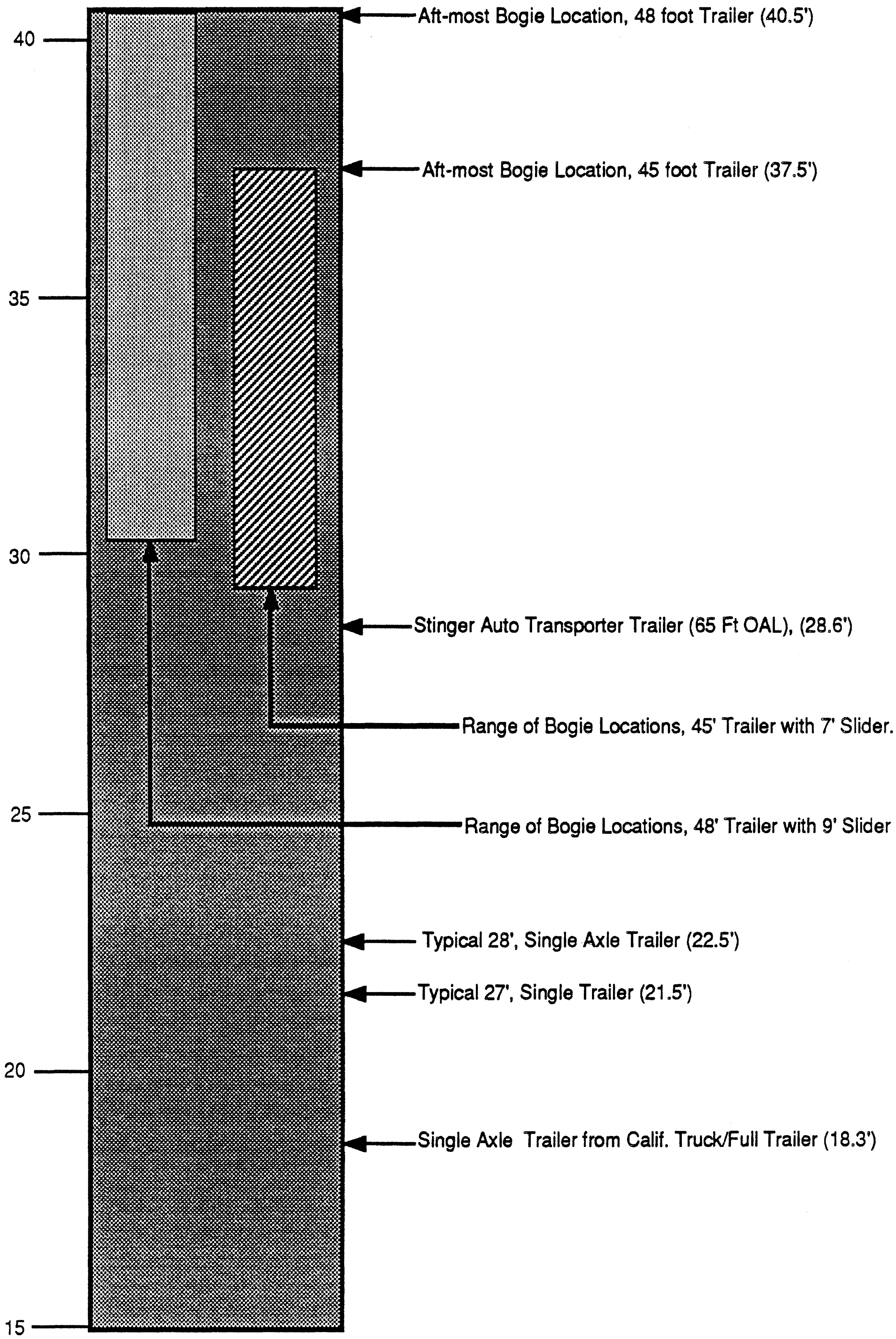
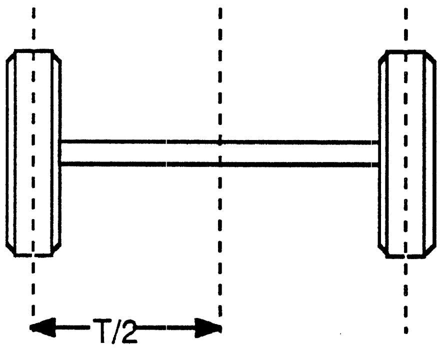


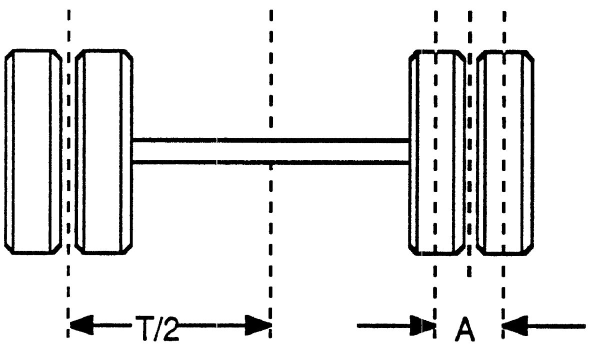
Figure 3.1.3 Trailer wheelbase- kingpin-to-rear axle (or tandem) center

TRACK WIDTHS



Conventional Axle, T/2 = 38.5 to 40.5

Truck and Tractor steering axles on 96" wide power units



W	T/2	A
96	35.5 to 36	12.5 to 13.0
102	38.5 to 39	12.5 to 13.0

Figure 3.1.4 Track widths

dual-tire-equipped axles (for both power units and trailers) on vehicles built to a 96-inch overall width allowance, and (c) dual-tire-equipped axles for a 102-inch overall width allowance. The figure illustrates the following:

• The half-track-width dimension for front axles varies from 36.5 inches to 40.5 inches, depending upon details of the design and upon the employment of conventional, as opposed to "centerpoint-type" steering layouts. Note that, overall, the effective width to the center of the single tires on steering axles is substantially less than is nominally permitted within the 96-inch layout.

• Dual-tire-equipped axles built to the respective overall vehicle widths of 96 or 102 inches typically achieve the overall dimension at the outside of the tires (not counting sidewall bulge under load).

3.1.3.5 Overhang and longitudinal hitch offset dimensions. Values for the so-called "offset" dimension of the fifth wheel placed on tractors and the "overhang" dimension for pintle hitches placed on trucks and trailers are shown in Figure 3.1.5. The figure shows the following:

• In common practice, conventional styles of tractor-semitrailers employ fifth wheel offsets ranging from 0 to approximately 24 inches forward of the rear axle, or tandem, center. Specific examples are listed for 3-S2 (three-axle tractor and two-axle trailer) and 2-S1-2 (doubles) combinations which have their fifth wheels located for achievement of the 80,000-lb gross weight condition. It is also noted that many road drivers prefer values of fifth wheel offset near zero in order to achieve an improved ride vibration condition in the cab.

• Aft-biased (or negative) values of fifth wheel offset are commonly seen in only two cases, namely, (a) dromedary-style tractor configurations, with offset values of approximately 46 inches and (b) stinger-steered auto transporters, with offset values of approximately 65 inches.

• Pintle overhang dimensions are approximately 28 inches, in the case of single-axle trailers, and 57 inches, in the case of tandem-axle trailers.

• Relatively long overhang dimensions are achieved on the trucks employed in truck/full trailer configurations. The example "California truck/full trailer" has a pintle overhang dimension of approximately 103 inches.

3.1.3.6 Elevation of hitches. Shown in Figure 3.1.6 are example values for the elevation dimension locating hitches. Since fifth wheels are generally installed directly above load-supporting axles, the elevation of the top surface of the fifth wheel depends upon the tire/wheel

Sample of Overhang and Longitudinal Hitch Offsets
inches

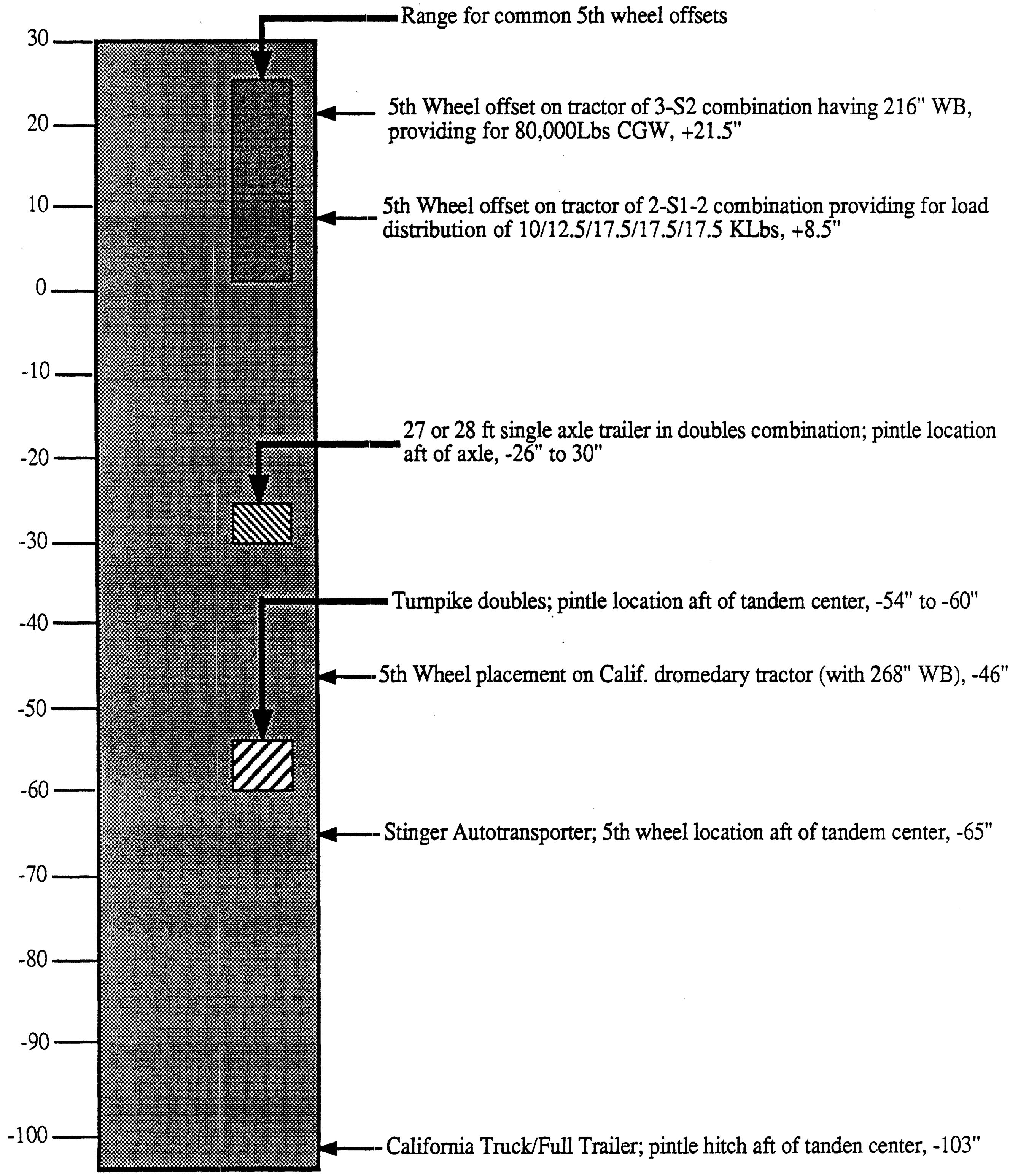


Figure 3.1.5 Overhang and longitudinal hitch offset dimensions from axle (or tandem) center to hitch centerline

Sample of Hitch Point Elevation above Ground
inches

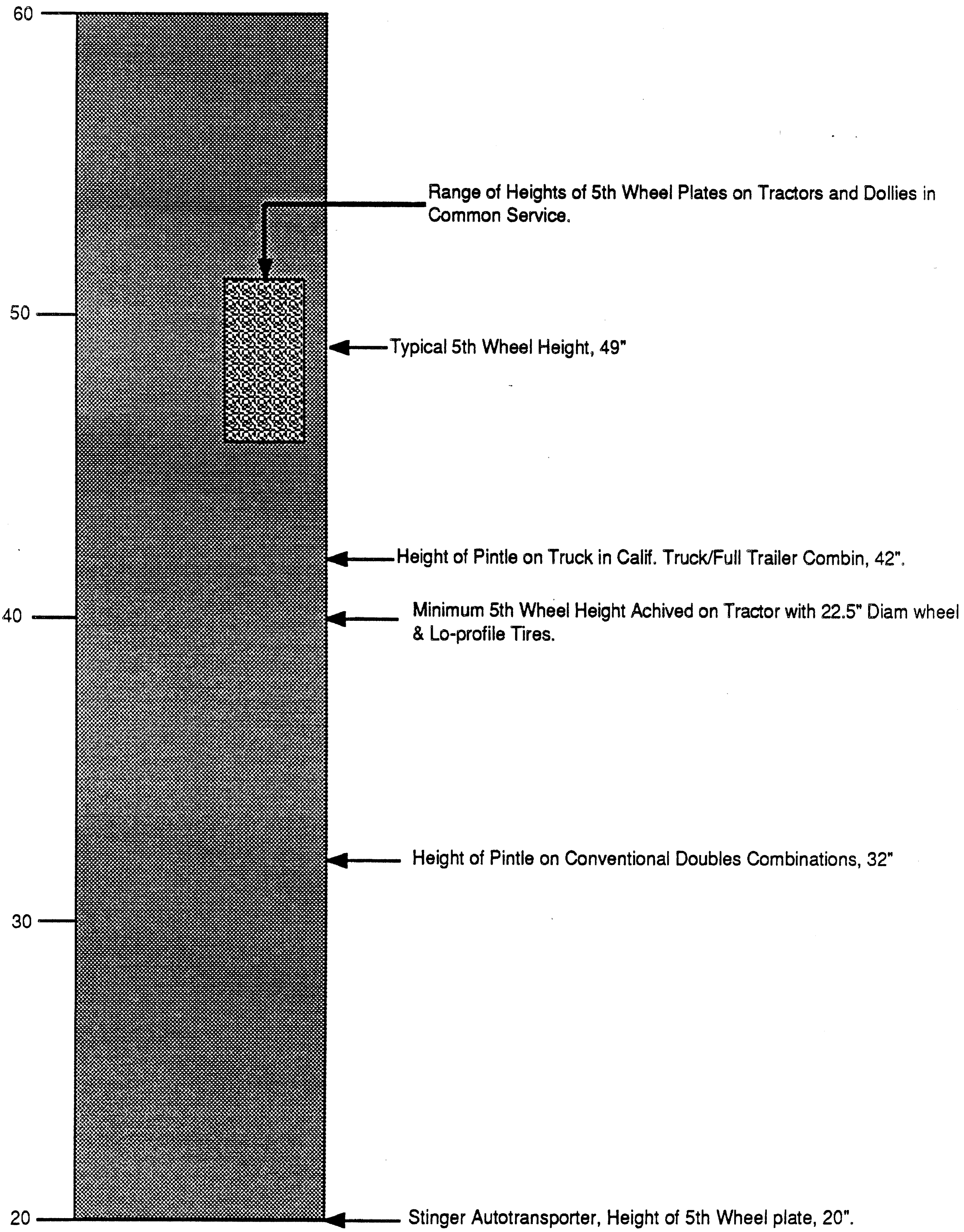


Figure 3.1.6 Hitch point elevation above ground

hardware which is chosen and upon details of frame and fifth wheel design. Pintle hitches, on the other hand, are installed at locations in which vertical position is relatively unconstrained. We see the following:

- Typical fifth wheel heights are in the range of 47 to 52 inches. A typical value for fifth wheel height would be 49 inches. A minimum fifth wheel height which has been achieved in the design of a tractor with 22.5-inch-diameter wheels and low-profile tires is 40.5 inches. In the stinger-steered layout of tractor configuration, a fifth wheel height of 20 inches is common.

- A typical height of pintle hitch in doubles and triples applications is approximately 32 inches.

3.2 Mass Distribution

3.2.1 Mass Distribution Properties. The mass distribution properties of rigid bodies are described by the body's mass (weight divided by the acceleration of gravity), its center of gravity (c.g.) location, and its moments of inertia. These quantities are also referred to as "inertial properties."

The moments of inertia are related to rotational motion in the same way that mass is related to translational motion, that is, the moments of inertia determine the tendency of a rigid body to resist changes in its rotational rates. There are roll, pitch, and yaw moments of inertia corresponding to rotations about the roll (longitudinal), pitch (lateral), and yaw (vertical) axes of the body, respectively. In general, the more spread out a mass is, the greater its moments of inertia will be.

A heavy truck can be viewed as an assembly of rigid bodies interconnected by suspensions and hitches. The suspensions connect the so-called "unsprung" masses to their "sprung" mass. Hitches interconnect the sprung masses of leading units to those of trailing units.

The vehicle itself is only a fraction of the total weight of a fully laden heavy truck. Clearly, the purpose of a heavy vehicle is to have enough capacity to carry a sizeable load. The ratio of laden to unladen weight is an indication of the vehicle's potential for productivity. The manner in which a vehicle is loaded has a large influence on its inertial properties.

Due to differences in mass distribution properties, the performance of an empty truck is much different from that of the same vehicle when it is loaded. In addition, the influence of load on tire properties is very important to dynamic performance.

Figure 3.2.1 illustrates the relationships of the pertinent inertial properties of entire units to those of their sprung and unsprung masses and payloads. Since the unsprung masses are remotely located with respect to the location of the total c.g. their mass is their most important inertial property. Their moments of inertia about their own c.g.'s are not large enough to be of major importance. The fore-aft c.g. locations of the sprung masses of empty trailers are centrally located, but because these units are long, their yaw and pitch moments of inertia are large. Since the widths and heights of trailers are considerably less than their lengths, their roll moments of inertia are small compared to their yaw and pitch moments of inertia, which are nearly equal.

Payloads, which are really a part of the sprung mass as illustrated in Figure 3.2.1, are often, but not always, centralized within the confines of a van. They are also much heavier than

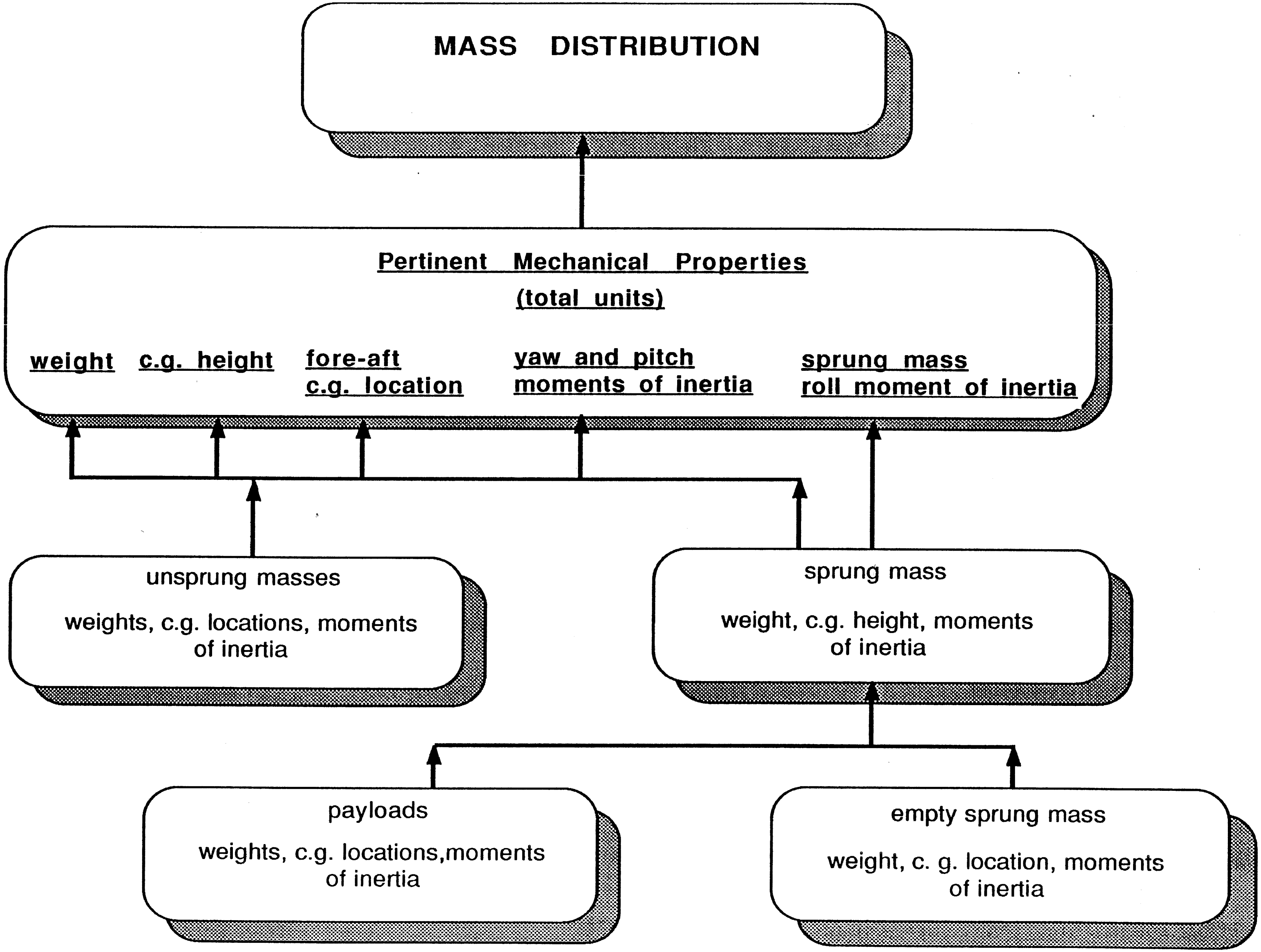


Figure 3.2.1 Breakdown of mass distribution properties

the "box" of the trailer. Consequently, even though the payload has a smaller volume than the box, it is as important, or more important, than the box of the trailer in determining the moments of inertia. Very dense loads are often carried in two parts located near the suspensions. Hence, they will have substantial yaw and pitch moments of inertia with respect to axes through the c.g., even though the moments of inertia of the payload would have been small if the total load were concentrated at a central location.

The sprung mass roll moment of inertia is presented as a pertinent mechanical property (see Figure 3.2.1) because of its importance to roll dynamics and dynamic rollover situations. Due to the suspension, the sprung mass will roll with respect to the unsprung masses to a much greater amount than it will pitch or yaw with respect to these masses. Hence, total moments of inertia can be used as a first-order approximation when considering yawing (steering) motions and pitching motions associated with braking, but roll studies are better handled using the rolling motion of the sprung mass.

3.2.2 <u>The Importance of Mass Distribution to Vehicle Maneuvering Behavior</u>. By examining Table 3.2.1, one can see that <u>weight</u> is given "Hi" importance in <u>all</u> of the vehicle maneuvers except <u>low-speed tracking</u> (which is only influenced by geometric layout as described in Section 3.1). <u>Weight</u> in combination with <u>fore-aft c.g. location</u> determines the loads on the various wheels. The wheel or axle loads are critical in <u>steady-turning</u> and <u>braking</u> situations because these loads have a major influence on the longitudinal and lateral forces that can be generated by the tires at the tire-road interface.

In non-steady, that is, transient, maneuvers the weight (or mass) of a unit determines the acceleration achieved per unit applied force. Hence, in addition to its influence on tire forces, weight is important in determining <u>response times</u>, <u>rearward</u> <u>amplification</u>, and performance in <u>transient braking</u> situations, and in <u>response to</u> <u>disturbances</u>.

In <u>downhill braking</u>, the <u>weight</u> of the vehicle contributes to the amount of potential energy that the vehicle possesses when it is at the top of a mountain. If speed is to be limited during a mountain descent, this potential energy must be dissipated (primarily by the brakes).

In certain simplified analyses, in which frictional forces at the road are taken to be proportional to load, the <u>weight</u> may be eliminated from consideration because its force and acceleration effects cancel each other. Hence, braking efficiency calculations may not depend upon weight directly, only upon the distribution of weight (i.e., wheel loads) in relation to the distribution of brake torques from wheel to wheel. Due to this relationship, the braking efficiency of a loaded vehicle is often much higher than that of an empty vehicle, when brakes are

Table 3.2.1

The Importance of the Pertinent Mechanical Properties of
Mass Distribution
on Vehicle Dynamic Performance

Pertinent Mechanical Properties of Mass Distribution	Low Speed Tracking	Hi Speed Tracking	Roll Stability	Yaw Stability	Response Time	Rearward Amp	Braking Efficiency	Transient Braking	Downhill Braking	Response to Disturbances
Weight	-	Hi	Hi	Hi	Hi	Hi	Hi	Hi	Hi	Hi
C.g. height	-	-	Hi	Hi	Low	Hi	Med	Med	-	Low
Fore-aft c.g. location	-	Hi	Med	Hi	Hi	Hi	Hi	Hi	-	Hi
Yaw moment of inertia	-	-	-	-	Hi	Med	-	Low	-	Low
Pitch moment of inertia	-	-	-	-	-	-	-	Med	-	-
Sprung roll moment of inertia	-	-	Med	-	Low	Low	-	Low	-	Low

proportioned in accordance with typical practice in the United States. *Increased weight without corresponding changes in tires and brakes leads to decreased dynamic performance.*

C.g. height is of primary importance in determining roll stability. Since yaw stability is influenced in an important manner by side-to-side load transfer during rolling, c.g. height is given "Hi" importance with respect to yaw stability. In this regard, c.g. height has some influence on rearward amplification, per se, but a high influence on the likelihood of a rollover during an avoidance maneuver. C.g. height is more important to roll than it is to pitch because truck units are usually much longer than they are wide. *Increased c.g. height decreases roll stability and can be detrimental to yaw stability.*

The fore/aft load transfer due to braking is not as influential as the side-to-side load transfer due to roll. Due to this, c.g. height is given "Med" (medium) importance (somewhat less than in roll-related situations) in braking situations (see Table 3.2.1).

Fore/aft c.g. location is a very important parameter related to the distribution of mass. The moments of inertia are not as important, because they do not significantly influence as many vehicle maneuvering situations as those influenced by c.g. location. As explained earlier, fore/aft c.g. location along with vehicle weight determines tire loads. In fact, tire loads (axle loads) and total weight are better known to truckers than c.g. locations because road-use laws are based on axle loads.

As can be seen by examining Table 3.2.1, fore-aft c.g. location has a large influence on all turning and braking situations except low-speed tracking and downhill descent, and it has a medium influence on static roll stability. Calculations indicate that *the manner in which vehicles are loaded deserves attention to ensure that loads are not biased excessively to the rear.*

The yaw moment of inertia, which is a measure of the extent of mass dispersion away from a vertical axis, resists the tendency of a vehicle to rotate about its vertical axis (turn). Hence, it is important to consider yaw moments of inertia when studying response times, rearward amplification, transient braking, and response to disturbances. *Increased yaw moment of inertia tends to decrease the responsiveness of vehicles.*

Pitch moments of inertia come into play in transients associated with the onset of braking or in situations in which braking effort is modulated such as in the use of antilock systems or if the driver attempts to pump the brakes. The pitch moment of inertia determines the timing or frequency of pitch motions.

The roll moment of inertia of the sprung mass is usually relatively small compared to the other moments of inertia. It influences roll motions and is important to the study of dynamic rollover and to a lesser extent in yaw transients. However, the suspension roll centers are usually far enough from payload c.g. heights that the resistance to roll motion about the roll centers is as much or more due to the c.g. height to roll center distance as it is due to sprung mass roll moment of inertia.

3.2.3 Characteristic Values of Mass Distribution Properties.

3.2.3.1 Rules of thumb for estimating inertial properties. In first-order analyses, the relative motions between sprung and unsprung masses are ignored and each unit of the vehicle is treated as a single mass. The "total" mass, c.g. location, and moments of inertia of a unit, consisting of sprung and unsprung masses, may be derived from the inertial properties of the constituent parts using principles of physics. Given guidance as to the inertial properties of sprung and unsprung masses, one can calculate the properties of various units.

For example, investigators have found the following rules of thumb to be useful for estimating the intertial properties of prototypical tractors. The sprung weight of the typical tandem-axle tractors can be estimated by the following formula:

$$W_s = 11{,}800 + 1{,}000\ [(L\text{-}190)/30]$$

where W_s is the sprung weight in pounds.

L is the wheelbase in inches.

This formula is based on the assumptions that (a) the sprung weight of a 190-in-wheelbase tractor is 11,800 lbs and (b) each added or subtracted 30 in of wheelbase translates into a change of 1000 lbs of sprung weight [13]. The next formula locates the horizontal position of the c.g. of the sprung weight with respect to the front axle of the tractor:

$$X_{11} = 55 + (L\text{-}190)/2$$

where X_{11} is the distance from the c.g. to the front axle, in inches.

The height of the sprung-mass c.g. is approximately 44 in for typical tractors.

The sprung-mass roll moment of inertia, I_{xx} (in. - lbs. - sec^2) can be estimated from the sprung weight by assuming a constant value of 29 in for the radius of gyration of the sprung mass, namely:

$I_{xx} = 2.178\ W_s$

The sprung-mass pitch and yaw moments of inertia (I_{yy} and I_{xx}) can be estimated by the empirical formula proposed on page 294 of reference [8], there:

$$I_{jj} = [(W_f + 0.4\ W_r)\ X_{11}^2 + 0.6\ W_r\ (L-X_{11})^2]/g$$

where j = x or y

$W_f = W_s\ (L - X_{11}) / L$

$W_r = W_s - W_f = W_s\ (X_{11} / L)$

g = gravitational constant, 386 in/sec^2

Since vehicle manufacturers often use the same chassis for either trucks or tractors, the approximations to the inertial properties of empty straight trucks can be obtained by combining the corresponding inertial properties of the truck body to those of a tractor with similar wheelbase.

The following conventions can be applied for estimating the inertial properties of current semitrailers. The empty sprung weight of tandem-axle semitrailers is approximately given by the following formula:

$$W_{se} = 5500 + (5300)\ [(L-27) / 21]$$

where L is the length of the semitrailer.

This formula is based on an interpolation scheme (see reference [14] volume 2, top of page 122) with parameters adjusted as follows:

W_{se} of a 48-ft tandem-axle semitrailer: 10,800 lbs

" " " 27' " " " : 5,500 lbs

For typical semitrailers with N axles (2 < N < 5), the above formula is generalized [13] to become:

$$W_{se} = 5500 + 5300\ [\ L-27) / 21] + 500\ [N-2]$$

In estimating moments of inertia, the following "reference" values are employed in extrapolating to other empty semitrailers of roughly the same size:

$$I_{xx} = 2.178\ W_s$$

The sprung-mass pitch and yaw moments of inertia (I_{yy} and I_{zz}) can be estimated by the empirical formula proposed on page 294 of reference [8], there:

$$I_{jj} = [(W_f + 0.4\ W_r)\ X_{11}^2\ + 0.6\ W_r\ (L\text{-}X_{11})^2]/g$$

where j = x or y

$$W_f = W_s\ (L - X_{11}) / L$$

$$W_r = W_s - W_f = W_s\ (X_{11} / L)$$

g = gravitational constant, 386 in/sec^2

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$$W_{se} = 5500 + 5300\ [\ L\text{-}27) / 21] + 500\ [N\text{-}2]$$

In estimating moments of inertia, the following "reference" values are employed in extrapolating to other empty semitrailers of roughly the same size:

Sprung I_{xxref} of 48-ft semitrailer: 80,000 in-lb-sec^2

Sprung I_{xxref} of 27-ft semitrailer: 55,000 in-lb-sec ^{2}

Sprung $I_{yyref} = I_{zzref}$ of 48-ft semitrailer 1,000,000 in-lb-sec^2

Sprung $I_{yyref} = I_{zzref}$ of 27-ft semitrailer 400,000 in-lb-sec^2

Using the above reference values, predicted moments of inertia are obtained as follows:

$$I_{xx} = I_{xx\,ref}\ (W_s\ /\ W_{sref})$$

$$I_{yy} = I_{zz} = I_{yyref}\,(Ws/W_{sref})\ (L/L_{ref})^2$$

where the subscript "ref" indicates values for the reference trailers selected for use in extrapolating to a trailer with different weight and/or length.

The c.g. height of the sprung mass of empty trailers is approximately 71 in. For empty 48-ft semitrailers, the c.g. of the sprung-mass is located approximately 300 in behind the kingpin, assuming the aft-most bogie location. Since the frame of a bogie slider assembly weighs approximately 2,000 lbs, the fore/aft c.g. location is influenced appreciably by the slider location.

For 27-ft or 28-ft trailers, the c.g. is located (in the fore/aft direction) somewhere near the center of the box because the masses of kingpin and landing gear structures tend to offset the mass associated with axle mounting hardware. Given that (a) the kingpin is 3 ft behind the front of the box, and (b) the axle is 3 ft in front of the rear of the box, the empty sprung mass c.g. is near the center of the wheelbase.

Typical weights for the unsprung masses (axles with tires, and brakes) are:

Tractor front axle (etc.), 1,200 lbs

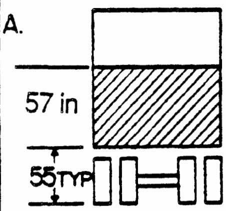
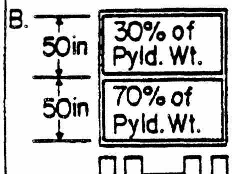
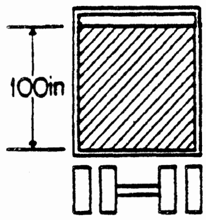
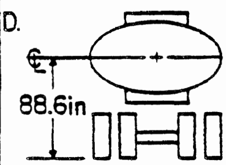
Tractor drive axle (etc.), 2,300 lbs

Trailer axle (etc.), 1,760 lbs

Payloads, of course, come in all sizes and densities. A few example arrangements of weight and c.g. height are illustrated in Table 3.2.2. Moments of inertia for payloads can be estimated using standard formulas. For example, the following formula applies to the uniformly dense, rectangular solid shown in Figure 3.2.2:

Table 3.2.2

Examples of Common Loading Cases with Accompanying Mass Center Height Parameters [15].

CASE	CONFIGURATION		WEIGHT (lbs)		MASS CENTER HEIGHT (Inches)		
			GVW	PAYLOAD	PAYLOAD	COMPOSITE SPR. MASS TRAILER & PYLD.	COMPOSITE SPR. MASS TRACTOR, TRAILER & PYLD.
A.		Full Gross, Medium-Density Freight (34.3 lb/ft^3)	80,000	52,200	83.5	80.0	75.0
B.		"Typical" LTL Freight Load	73,000	45,200	95.0	89.2	82.1
C.		Full Gross, Full Cube, Homogeneous Freight (18.7 lb/ft^3)	80,000	52,200	105.0	98.4	90.7
D.		Full Gross Gasoline Tanker	80,000	54,780	88.6	87.3	81.2

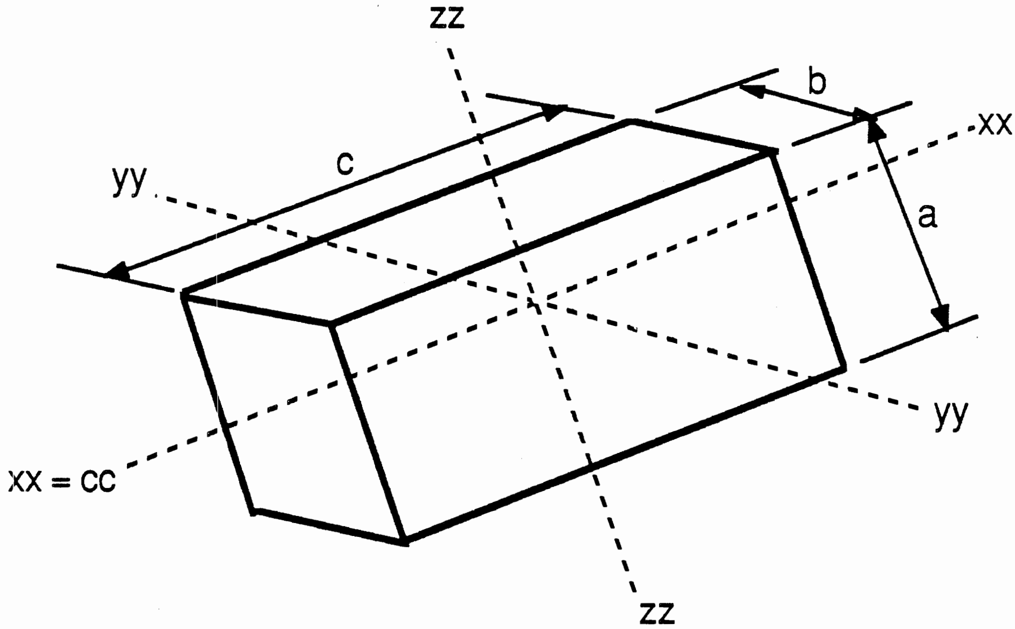


Figure 3.2.2 Dimensions used in estimating moments of inertia for a rectangular solid

$$I_{cc} = (W_p /386) (a^2 + b^2) /12$$

where I_{cc} is the moment of inertia about an axis through the c.g. and parallel to the edge with length c, edge lengths a and b are in inches, and W_p is the weight of the payload.

The above formula can be used to calculate roll, pitch, and yaw moments of inertia by using the appropriate dimensions corresponding to the appropriate axis of rotation, that is cc = xx, cc = yy, or cc = zz, respectively.

Application of rules of thumb for the constituent parts to the calculation of inertial properties for total units can be tedious. There are a few rules that can be used to estimate total unit properties for tractors and trucks. For pitch and yaw moments of inertia for tractors and straight trucks:

$$I_{yyT} = I_{zzT} = (L/2)^2 W_t/386$$

If wheel loads are known, the fore-aft c.g. locations can be readily calculated as follows:

$$X_{cg} = F_r L/W_t$$

where

X_{cg} is the distance from the c.g. to the front axle (or the kingpin of a semitrailer)

F_r is the load on the rear axle or tandem set

W_t is the total weight

L is the wheelbase

3.2.3.2 Ranges of characteristic values. The following charts summarize the ranges of characteristic values of pertinent mechanical properties pertaining to mass distribution. See reference [16] for information on the measured values of inertial properties presented in these charts.

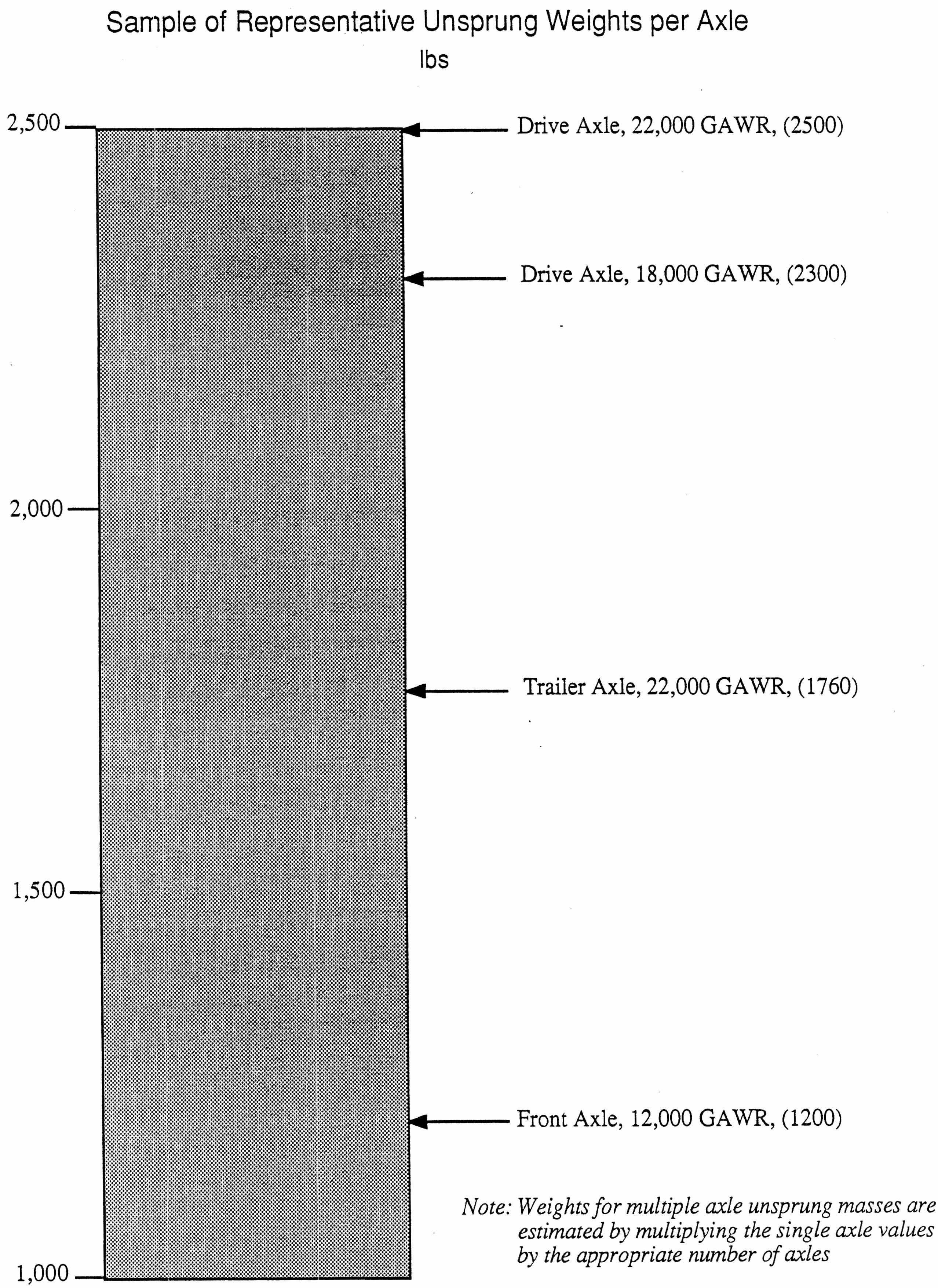


Figure 3.2.3 Representative unsprung weights per axle (includes axle, tires, brakes, etc.)

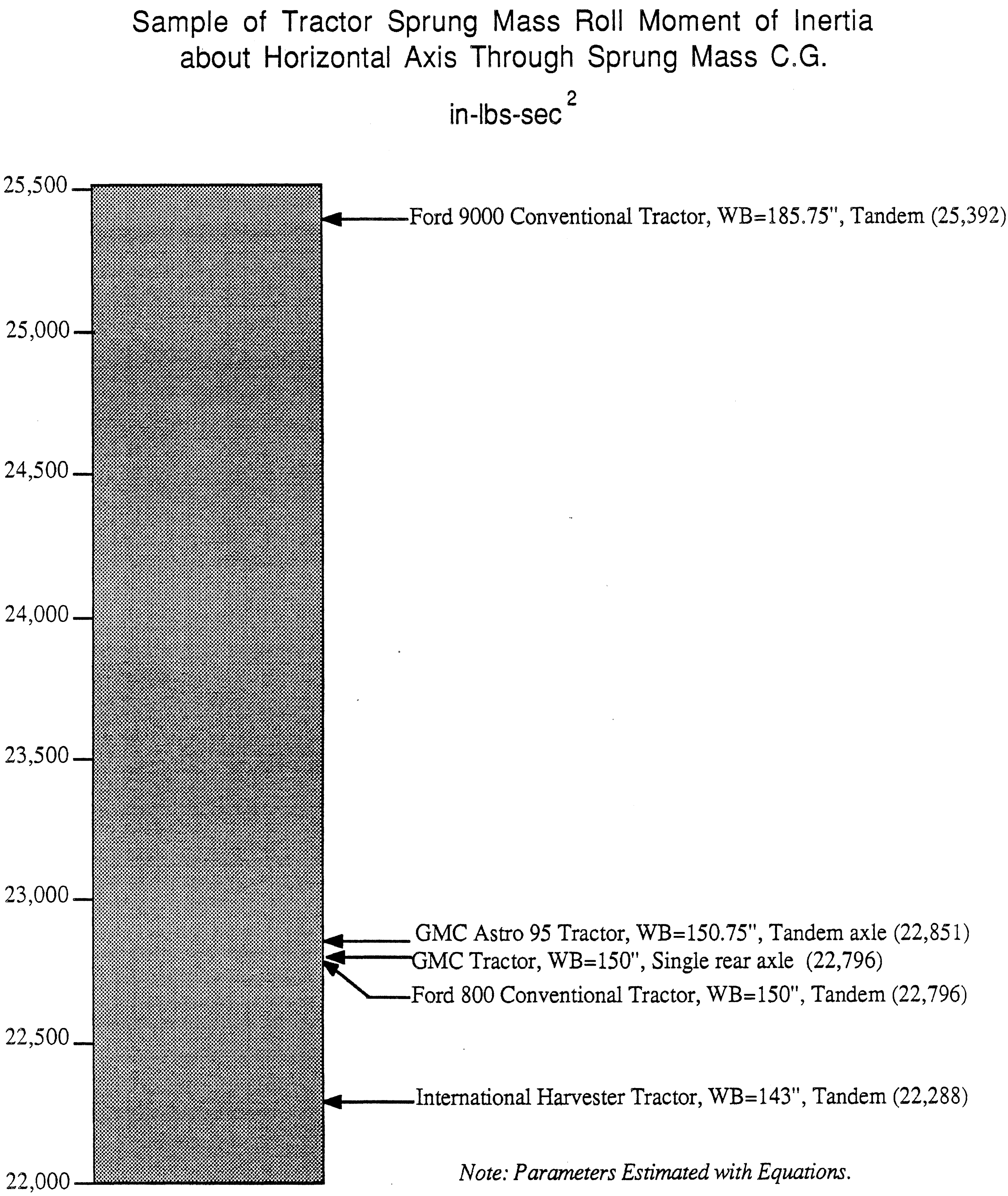


Figure 3.2.4 Tractor sprung mass roll moment of inertia about horizontal axis through sprung mass c.g.

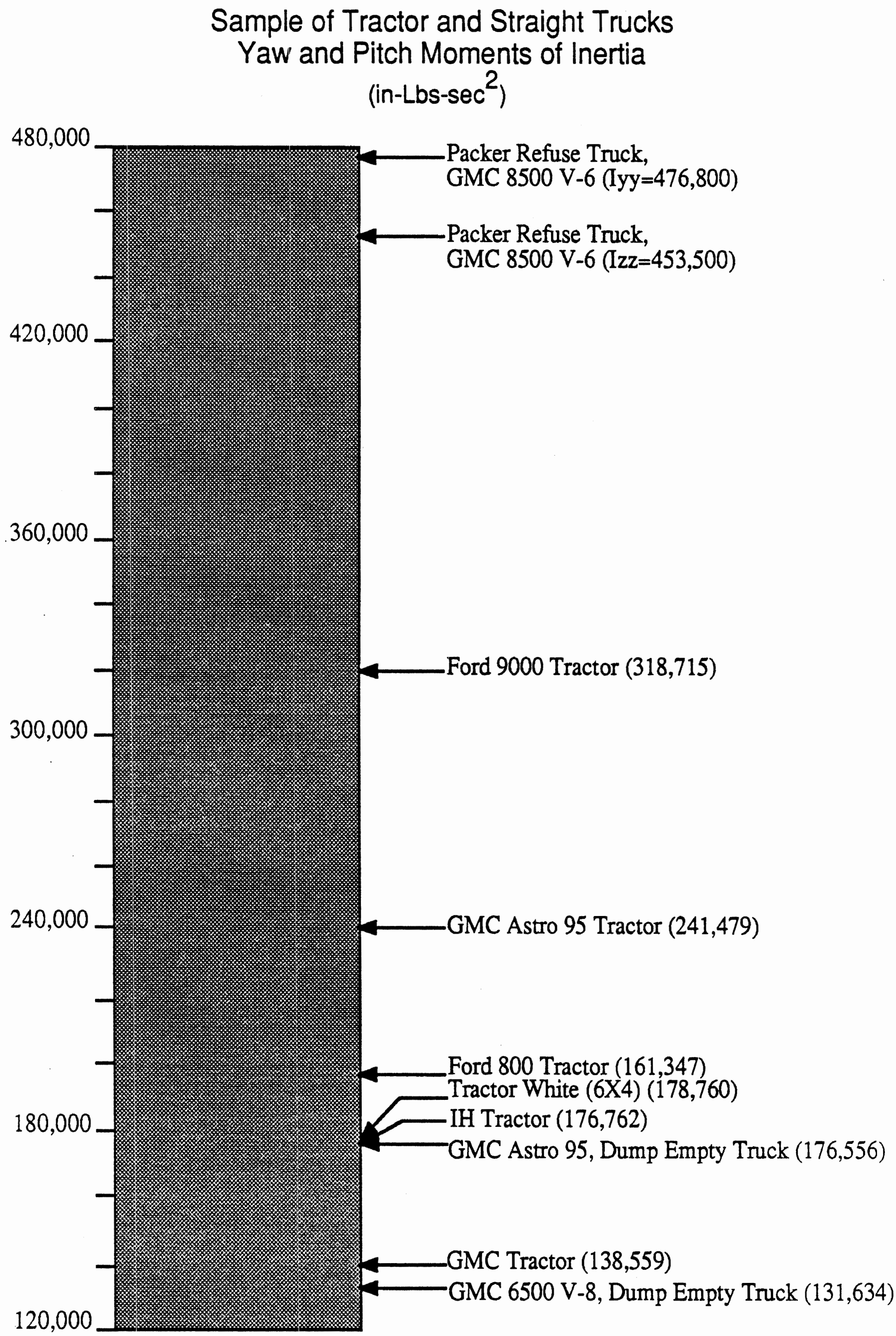


Figure 3.2.5 Tractor and straight trucks yaw and pitch moments of inertia about axes through total c.g. (unit unladen)

Sample of Tractor and Straight Truck Fore-aft C.G. Locations
(inches behind the front axle)

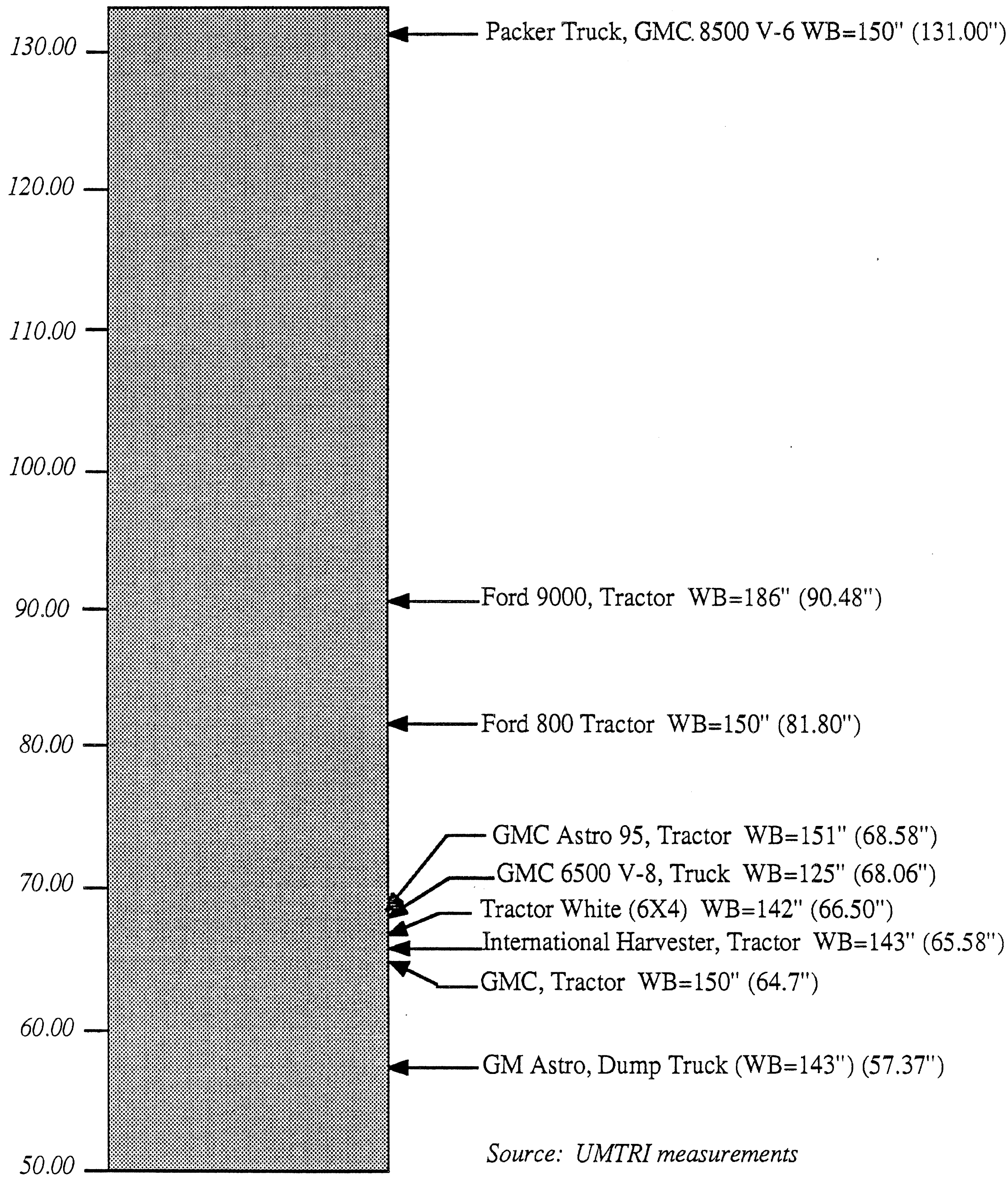


Figure 3.2.6 Tractor and straight truck fore-aft c.g. locations (total unit, unladen)
(see "Diesel Truck Index" [19] for more examples)

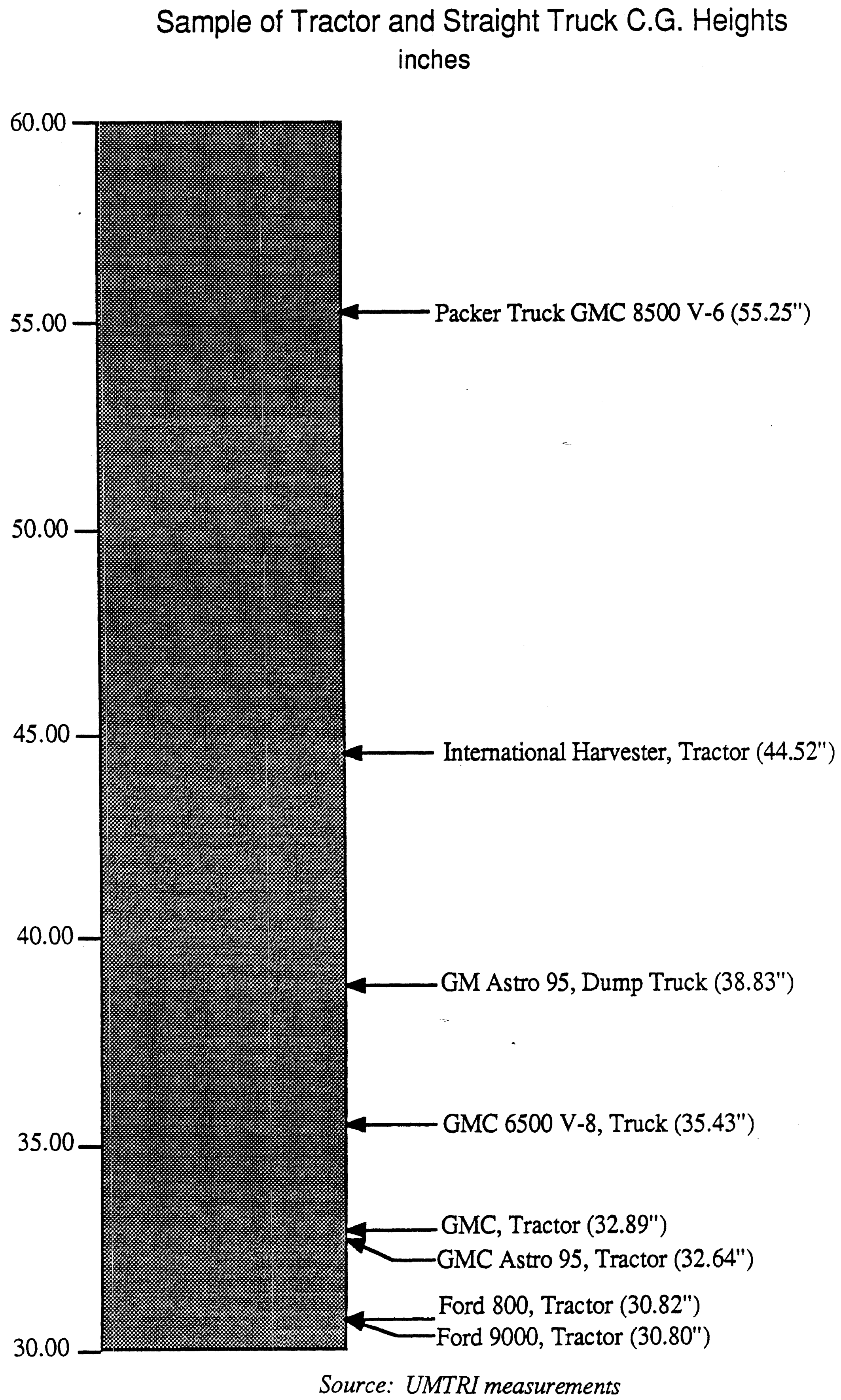


Figure 3.2.7 Tractor and straight truck c.g. heights (total unit, unladen)

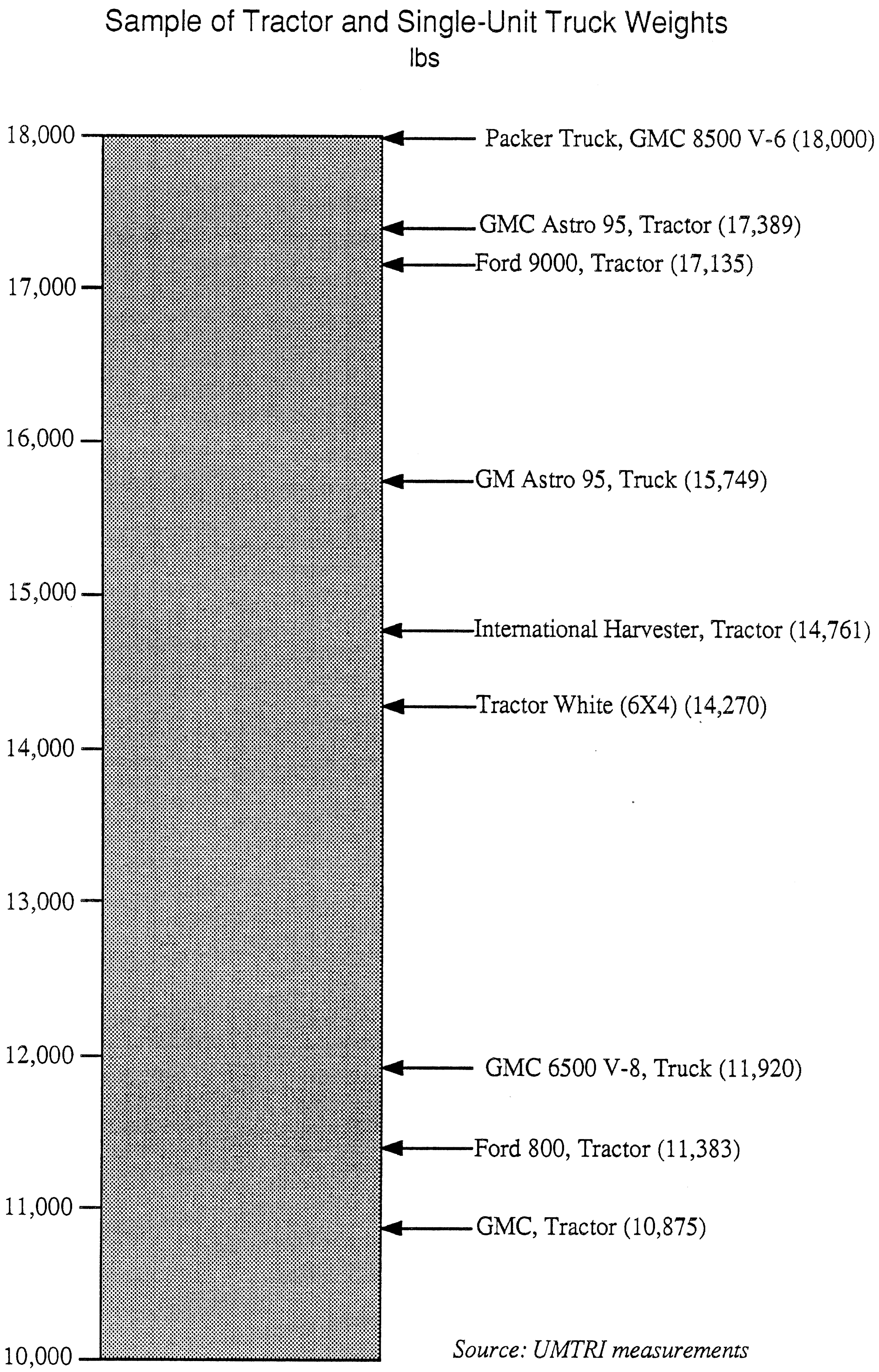


Figure 3.2.8 Tractor and single-unit truck weights (total unit, unladen)
(see "Diesel Truck Index" [19] for more examples)

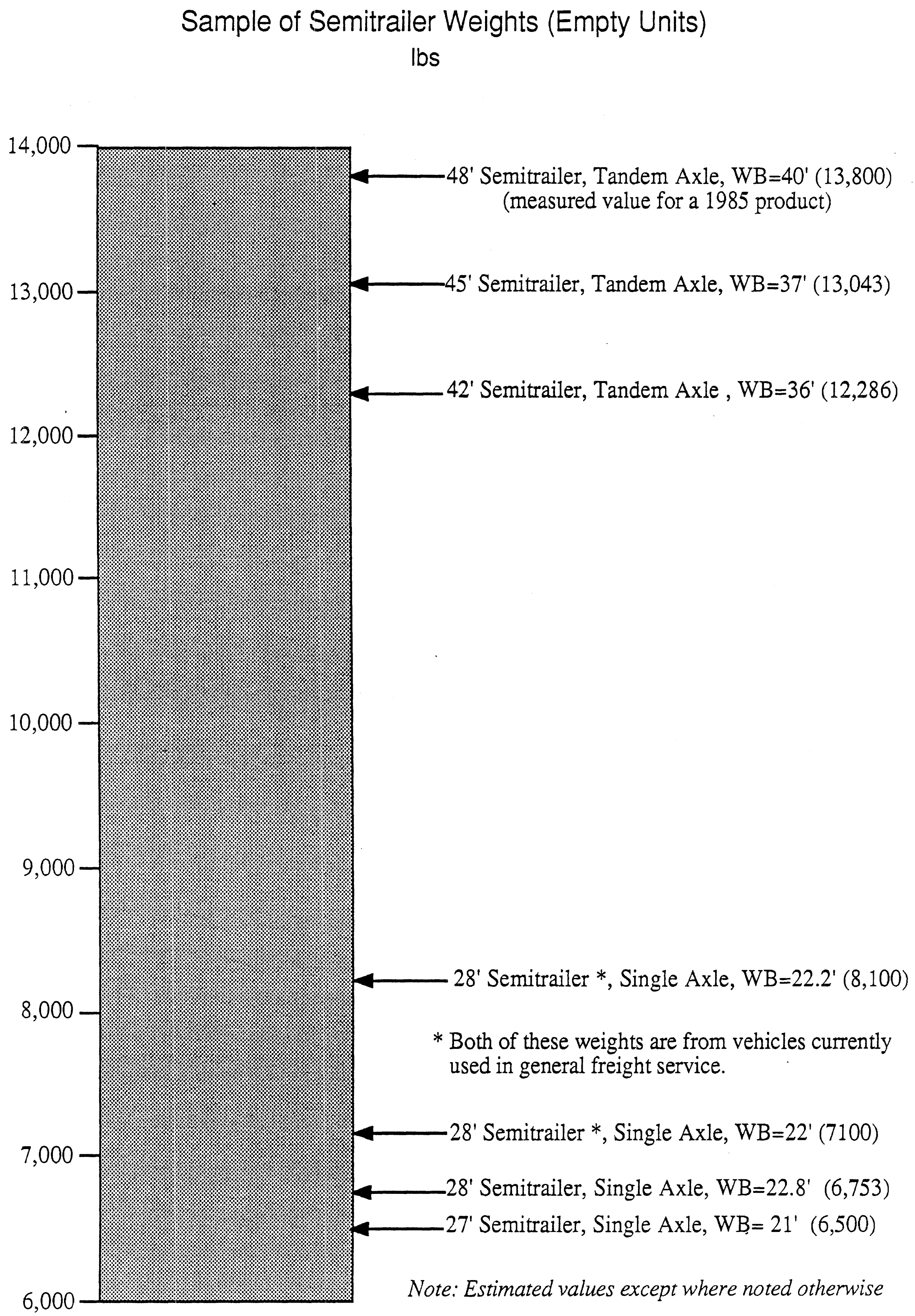
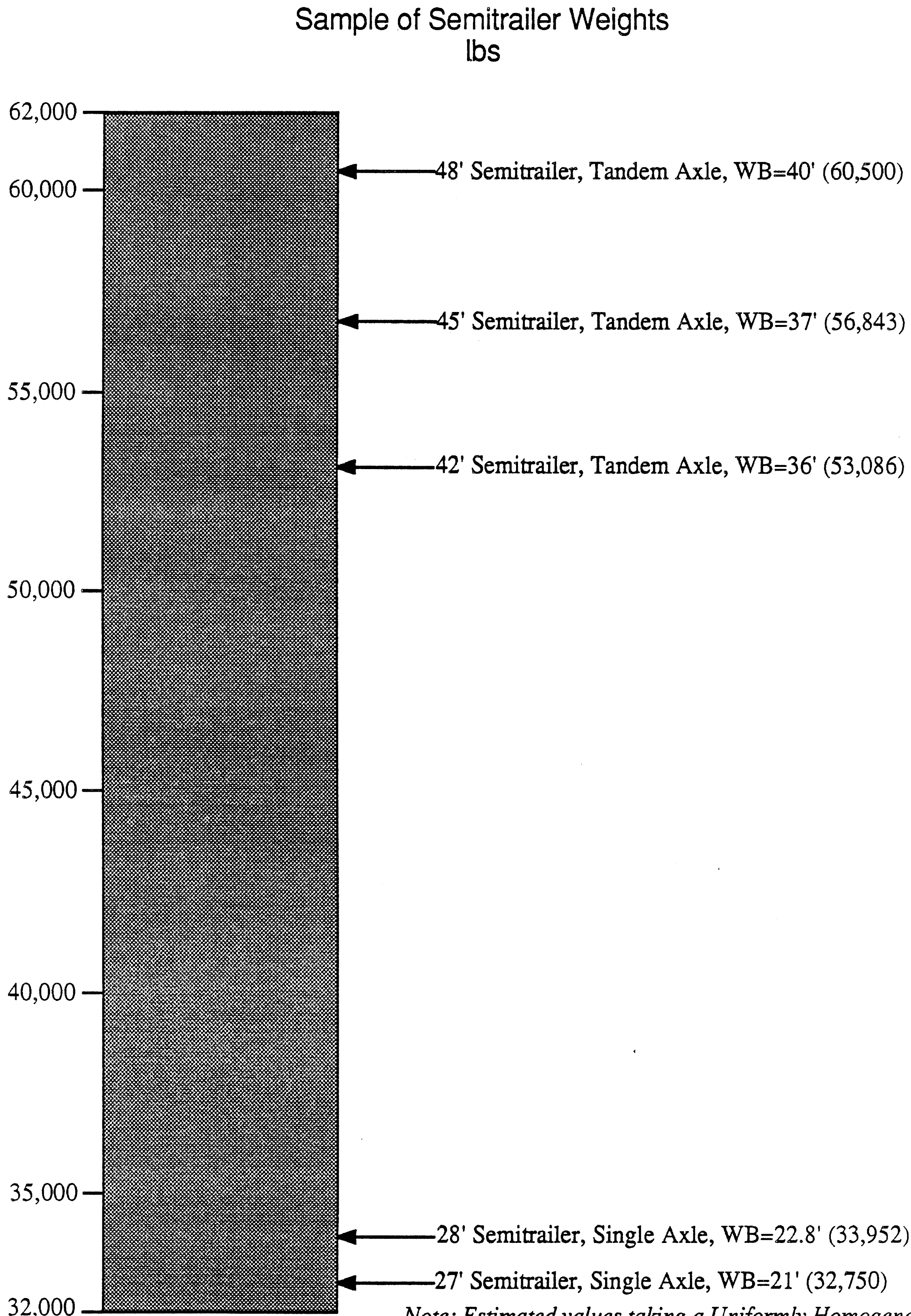
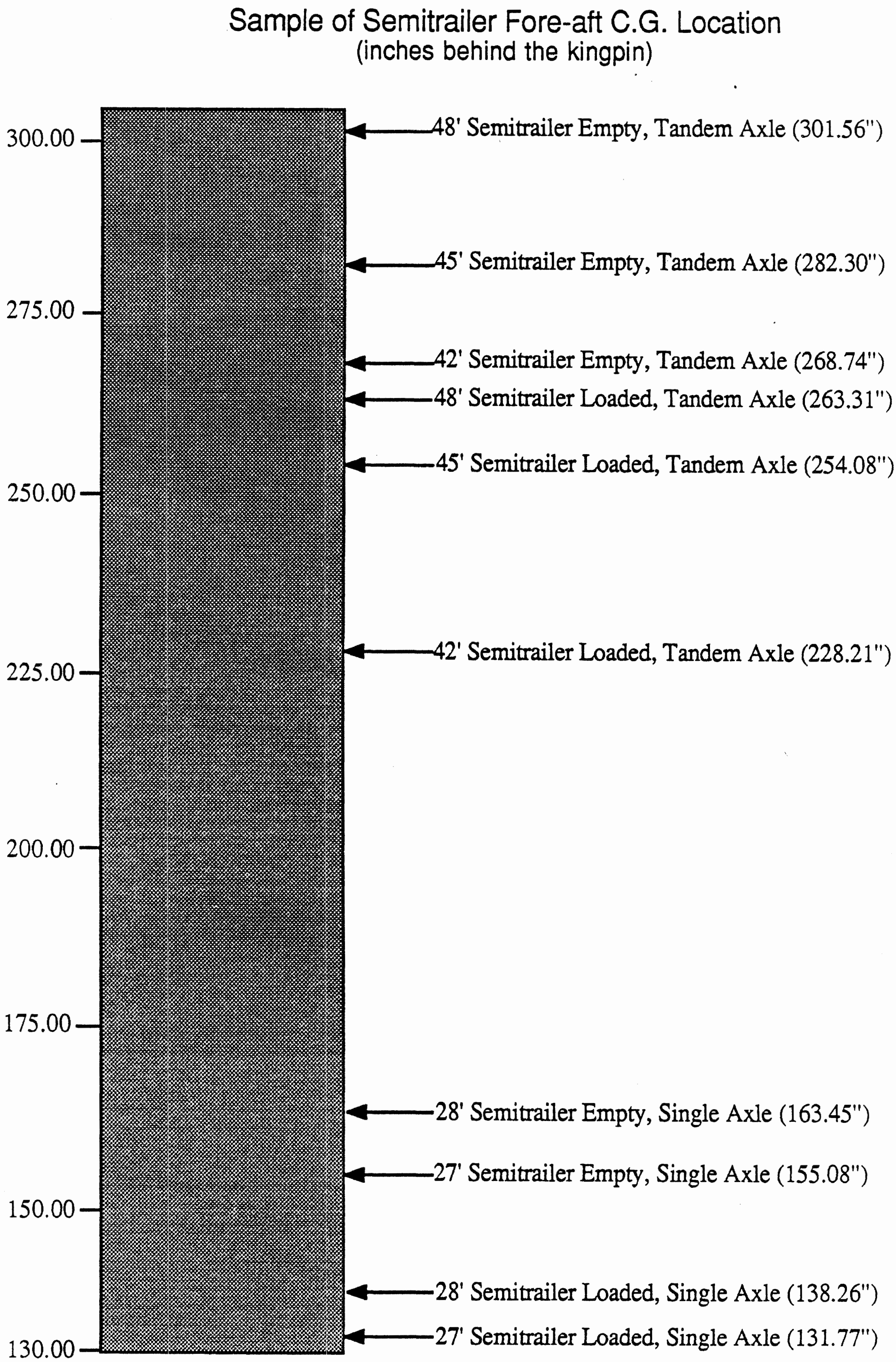


Figure 3.2.9 Semitrailer weights (empty units)



Note: Estimated values taking a Uniformly Homogeneous Freight with a density of; $\rho = 14.0 \, \frac{\text{lbs}}{\text{ft}^3}$

Figure 3.2.10 Semitrailer weights (loaded units)



Note: Estimated values taking a Uniformly Homogeneous Freight with a density of ; $\rho = 14.0 \frac{Lbs}{Ft^3}$

Figure 3.2.11 Semitrailers fore-aft c.g. location (inches behind the kingpin)

Sample of Semitrailers Yaw and Pitch Moments of Inertia (Empty Units)
in-Lbs-sec ^{2}

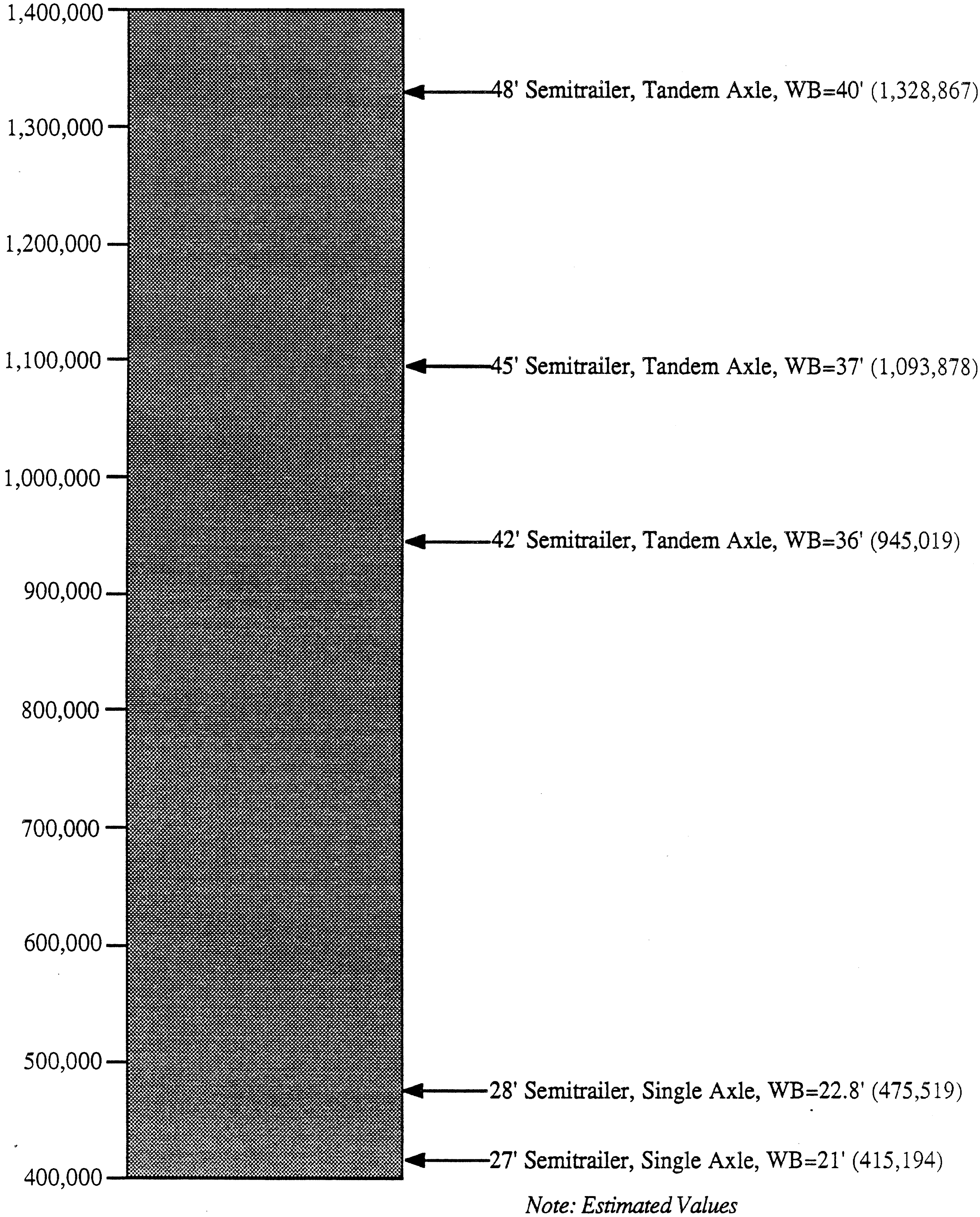
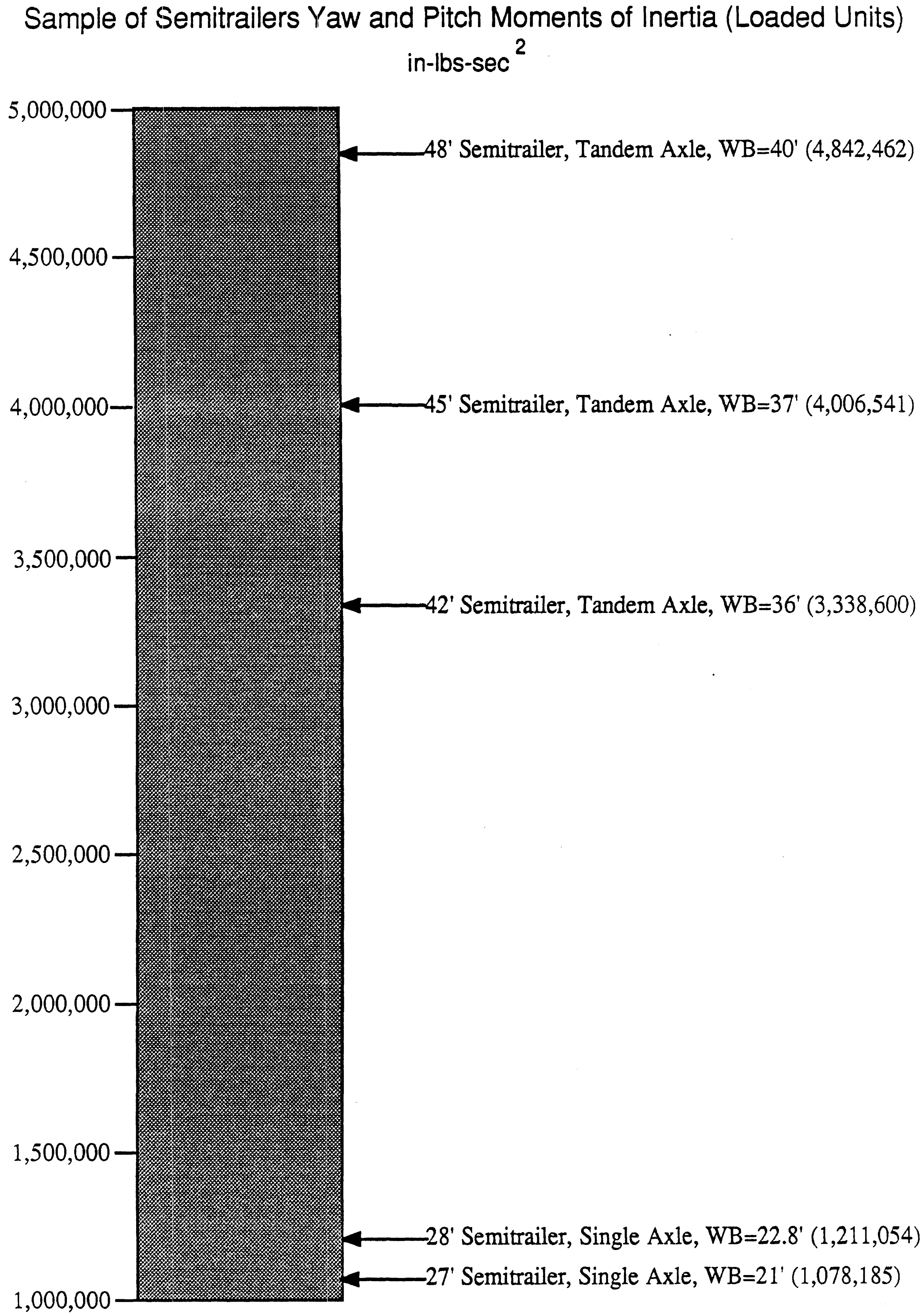


Figure 3.2.12 Semitrailers yaw and pitch moments of inertia (empty units)



Note: Estimated values taking a Uniformly Homogeneous Freight with a density of ; $\rho = 14.0 \frac{\text{Lbs}}{\text{Ft}^3}$

Figure 3.2.13 Semitrailers yaw and pitch moments of inertia (loaded units)

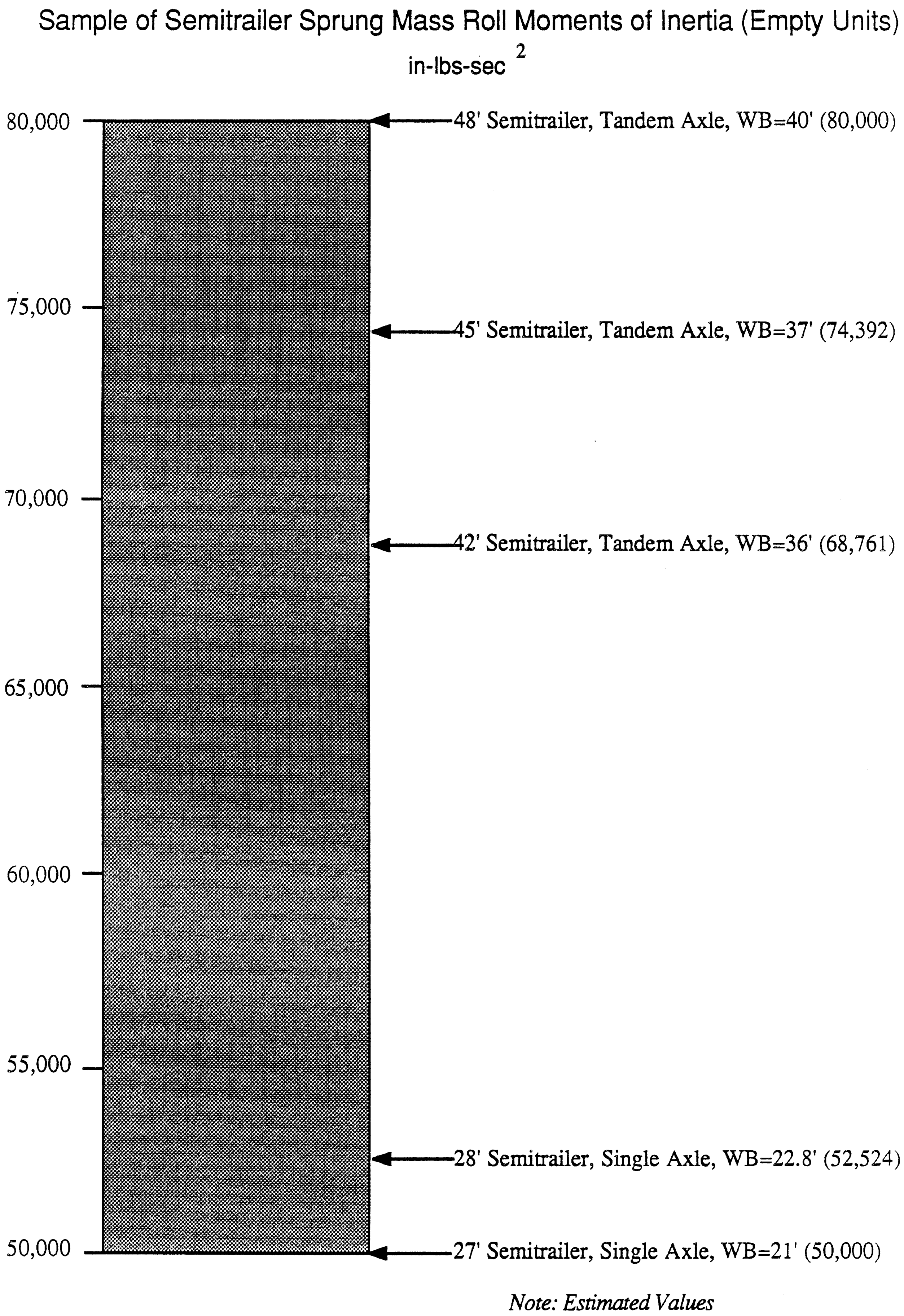


Figure 3.2.14 Semitrailer sprung mass roll moments of inertia (empty units)

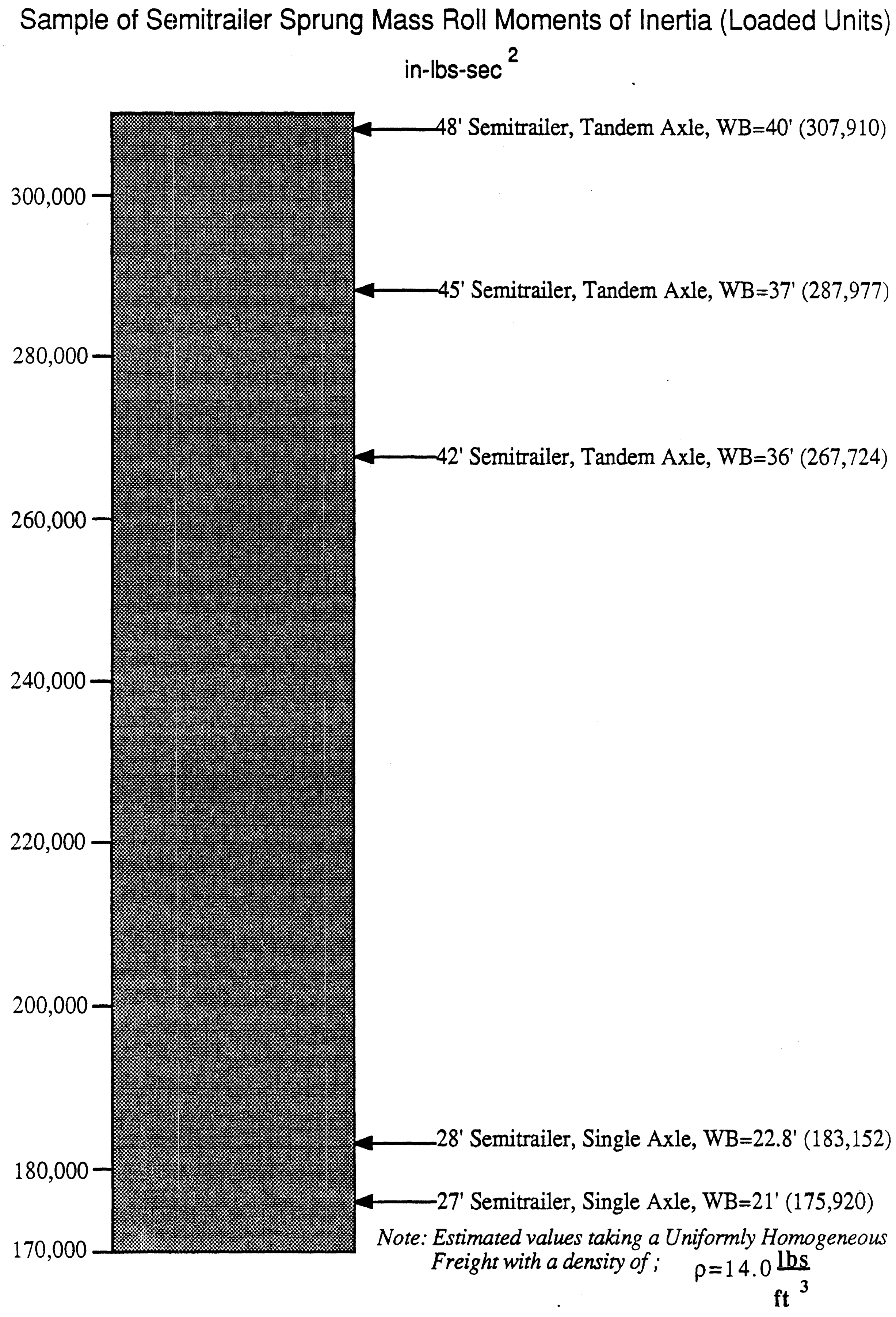


Figure 3.2.15 Semitrailer sprung mass roll moments of inertia (loaded units)

4.0 REFERENCES

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2) Fancher, P.S. (ed.), "Descriptive Parameters Used in Analyzing the Braking and Handling of Heavy Trucks," Final Report, University of Michigan Transportation Research Institute, Report Nos. UM-HSRI-81-19-1 through -5, October 1981.

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17) Radlinski, R. W., and Williams, S. F., "NHTSA Heavy Duty Vehicle Brake Research Program Report No. 5 --Pneumatic Timing," Report No. DOT-HS-806-897, December 1985.

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APPENDIX A

DATA SETS FOR BENCHMARK VEHICLES

The following items are printouts from computer files that are used in representing straight trucks, tractor-semitrailers, truck-full-trailers, doubles, triples, and B-trains [1].

Load File

Payload for a 27 ft full trailer
Factors influencing heavy truck dynamic performance – NHTSA project
Payload file
Filename = ST6T:Pl.Full.27

Payload Weight = 25300.000 lbs.

Center of Gravity Distance = 136.200 in.
Center of Gravity Lateral Offset = 0.000 in.
Center of Gravity Height = 83.110 in.

Roll Moment of Inertia = 65000.000 in-lb-sec^2.

Pitch Moment of Inertia = 375000.000 in-lb-sec^2.

Yaw Moment of Inertia = 375000.000 in-lb-sec^2.

Notes:

1. CG-Distance is measured along x-axis, positive aft of front articulation point (front axle for Unit 1, king-pin for a semi, pintle-hook for a dolly).
2. CG-Height and Z-Locations are with reference to the ground.

Load File

Payload for a 42 ft semitrailer
Factors influencing heavy truck dynamic performance - NHTSA project
Payload file
Filename = ST6T:Pl.Semi.42

Payload Weight = 52500.000 lbs.

Center of Gravity Distance = 218.060 in.
Center of Gravity Lateral Offset = 0.000 in.
Center of Gravity Height = 82.000 in.

Roll Moment of Inertia = 132000.000 in-lb-sec^2.
Pitch Moment of Inertia = 3050000.000 in-lb-sec^2.
Yaw Moment of Inertia = 3050000.000 in-lb-sec^2.

Notes:

1. CG-Distance is measured along x-axis, positive aft of front articulation point (front axle for Unit 1, king-pin for a semi, pintle-hook for a dolly).
2. CG-Height and Z-Locations are with reference to the ground.

Tire File

Tire model, cornering stiffness and aligning moment data for a benchmark (actual name unknown) tire on a wet surface
Factors influencing heavy truck dynamic performance - NHTSA project
Filename = ST6T:Ti.Low.Frict

Tire Radius = 19.5 in.

Polar Moment of Inertia = 103.0 in-lbs-sec^2.
Lateral Stiffness = 0.0 lb/in.
Vertical Stiffness = 4500.0 lb/in.
Camber Stiffness = 0.0 lb/deg.
Overturning Stiffness = 0.0 lb/deg.
Aligning Torque Stiffness = 1200.0 in-lb/deg.
Longitudinal Stiffness = 35400.0 lb/slip.
Cornering Stiffness = 676.0 lb/deg.
Peak Cornering Friction Coefficient = 0.3

Mu-y vs. Slip Angle Tables

Slip Angle [deg]	Mu-y
1.00	0.18
2.00	0.30
4.00	0.36
6.00	0.38
8.00	0.39
12.00	0.40
16.00	0.40

Velocity = 58.7 ft/sec.
Load = 3000.0 lbs.

Slip Angle [deg]	Mu-y
1.00	0.15
2.00	0.27
4.00	0.33
6.00	0.36
8.00	0.37
12.00	0.38
16.00	0.39

Velocity = 58.7 ft/sec.
Load = 6000.0 lbs.

Slip Angle [deg]	Mu-y
1.00	0.11
2.00	0.21
4.00	0.30
6.00	0.33
8.00	0.35
12.00	0.36
16.00	0.37

Velocity = 58.7 ft/sec.
Load = 9000.0 lbs.

Semi-Empirical Tire Model Parameters

Variable	Inital Value	D(var)/ Dload	D(var)/ Dvelocity
Nominal Cornering Stiff - lb/deg	676.00	0.13	0
Peak Friction Value	0.34	-9.93E-06	-2.69E-03
Locked Wheel Friction Value ..	0.23	-1.18E-05	-1.86E-03
Slip Value at Peak Friction-% ..	0.21	1E-05	-3.4E-04
Nominal Pneumatic Trail-in ...	1.50	1.25E-04	0
Lateral Stiffness-lb/in	5000.00	0	0
Nominal Vertical Load-lbs	4000.000	N/A	N/A
Nominal Velocity-ft/sec	58.700	N/A	N/A

$D^2(\text{Calfa})/\text{Dload}^2 - (\text{lb-deg})^{-1} = -1.4\text{E-}05$

Aligning Torque Table

	Slip Angle						
Load	1.00	2.00	4.00	6.00	8.00	12.00	16.00
3000	731.5	720.0	435.1	306.2	235.5	159.7	120.0
6000	1540.2	1919.4	1211.5	861.7	665.5	454.0	341.8
9000	2074.2	3706.7	2612.3	1907.0	1489.6	1027.0	777.0

Aligning Moment values are in ft/lbs.

Tire File

Tire model, cornering stiffness and aligning moment data for a benchmark (actual name unknown) tire on a dry surface
Factors influencing heavy truck dynamic performance - NHTSA project
Filename = ST6T:Ti.High.Fric

Tire Radius = 19.5 in.

Polar Moment of Inertia = 103.0 in-lbs-sec^2.
Lateral Stiffness = 0.0 lb/in.
Vertical Stiffness = 4500.0 lb/in.
Camber Stiffness = 0.0 lb/deg.
Overturning Stiffness = 0.0 lb/deg.
Aligning Torque Stiffness = 1200.0 in-lb/deg.
Longitudinal Stiffness = 35400.0 lb/slip.
Cornering Stiffness = 676.0 lb/deg.
Peak Cornering Friction Coefficient = 0.8

Mu-y vs. Slip Angle Tables

Slip Angle [deg]	Mu-y
1.00	0.18
2.00	0.35
4.00	0.64
6.00	0.75
8.00	0.81
12.00	0.87
16.00	0.89

Velocity = 66.0 ft/sec.
Load = 3000.0 lbs.

Slip Angle [deg]	Mu-y
1.00	0.17
2.00	0.34
4.00	0.62
6.00	0.73
8.00	0.79
12.00	0.85
16.00	0.88

Velocity = 66.0 ft/sec.
Load = 6000.0 lbs.

Slip Angle [deg]	Mu-y
1.00	0.15
2.00	0.29
4.00	0.56
6.00	0.68
8.00	0.74
12.00	0.80
16.00	0.83

Velocity = 66.0 ft/sec.
Load = 9000.0 lbs.

Semi-Empirical Tire Model Parameters

Variable	Inital Value	D(var)/ Dload	D(var)/ Dvelocity
Nominal Cornering Stiff-lb/deg	676.00	0.13	0
Peak Friction Value	0.78	-1.6E-05	0
Locked Wheel Friction Value ..	0.52	-1.4E-05	0
Slip Value at Peak Friction-% ..	0.25	0	0
Nominal Pneumatic Trail-in ...	1.00	1E-04	0
Lateral Stiffness-lb/in	1500.00	0	0
Nominal Vertical Load-lbs	4000.000	N/A	N/A
Nominal Velocity-ft/sec	66.000	N/A	N/A

$D^2(\text{Calfa})/\text{Dload}^2$ - $(\text{lb-deg})^{-1}$ = -1.4E-05

Aligning Torque Table

	Slip Angle						
Load	1.00	2.00	4.00	6.00	8.00	12.00	16.00
3000	478.8	958.0	1188.6	929.3	746.8	528.2	404.5
4000	676.1	1352.5	1753.3	1380.4	1112.5	788.9	604.7
6000	1056.1	2112.8	3171.2	2566.8	2090.8	1496.2	1151.6
9000	1464.1	2929.2	5865.6	5222.3	4397.6	3243.7	2520.2

Aligning Moment values are in ft/lbs.

Tare File

NHTSA Benchmark two axle tractor - Tare file
Tractor used in B-Train, Doubles and Triples combinations
Factors influencing heavy truck dynamic performance - NHTSA project
Filename = ST6T:Ta.Tract.Two

Tractor Trailer Sprung Mass = 9700.0 lbs.
Torsional Stiffness = 50000.0 in-lb/deg.
Torsional Friction = 11000.0 in-lbs.
Torsion Axis Height = 36.0 in.

Rear Suspension

Number	WheelBase to (Location)
1	132.0

Center of Gravity Position	
Sprung CG Distance	26.4 in.
Sprung CG Offset	0.0 in.
Sprung CG Height	44.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia	21112.0 in-lbs-sec^2.
Pitch Moment of Inertia	48989.0 in-lbs-sec^2.
Yaw Moment of Inertia	48989.0 in-lbs-sec^2.

Rear Hitch Location	
X Location	122.4 in.
Y Location	0.0 in.
Z Location	48.0 in.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Tare File

NHTSA Benchmark three axle tractor - Tare file
Tractor used for towing 42 ft semi-trailers
Factors influencing heavy truck dynamic performance - NHTSA project
Filename = ST6T:Ta.Tract.III

Tractor Trailer Sprung Mass = 9700.0 lbs.
Torsional Stiffness = 50000.0 in-lb/deg.
Torsional Friction = 11000.0 in-lbs.
Torsion Axis Height = 36.0 in.

Rear Suspension

Number	WheelBase to (Location)
1	144.0

Center of Gravity Position	
Sprung CG Distance	28.8 in.
Sprung CG Offset .	0.0 in.
Sprung CG Height .	44.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia .	21112.0 in-lbs-sec^2.
Pitch Moment of Inertia	58301.0 in-lbs-sec^2.
Yaw Moment of Inertia	58301.0 in-lbs-sec^2.

Rear Hitch Location	
X Location	129.6 in.
Y Location	0.0 in.
Z Location	48.0 in.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Tare File

NHTSA Benchmark three axle straight truck - Tare file
Truck used independently
Factors influencing heavy truck dynamic performance - NHTSA project
Filename = ST6T:Ta.Truck.St

Tractor Trailer Sprung Mass = 12200.0 lbs.
Torsional Stiffness = 50000.0 in-lb/deg.
Torsional Friction = 11000.0 in-lbs.
Torsion Axis Height = 36.0 in.

Rear Suspension

Number	WheelBase to (Location)
1	240.0

Center of Gravity Position	
Sprung CG Distance	72.0 in.
Sprung CG Offset .	0.0 in.
Sprung CG Height .	56.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia .	40000.0 in-lbs-sec^2.
Pitch Moment of Inertia	105000.0 in-lbs-sec^2.
Yaw Moment of Inertia	105000.0 in-lbs-sec^2.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Tare File

NHTSA Benchmark single axle semitrailer - Tare file
Trailer used in B-Train, Doubles and Triples combinations
Factors influencing heavy truck dynamic performance - NHTSA project
FileName = ST6T:Ta.Semi.One

Semi Sprung Mass = 4500.0 lbs.
King Pin Setting = 36.0 in.

Rear Suspension

Number	WheelBase to (Location)
1	252.0

Center of Gravity Position	
Sprung CG Distance	126.0 in.
Sprung CG Offset .	0.0 in.
Sprung CG Height .	60.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia .	49500.0 in-lbs-sec^2.
Pitch Moment of Inertia	360000.0 in-lbs-sec^2.
Yaw Moment of Inertia	360000.0 in-lbs-sec^2.

Rear Hitch Location	
X Location	288.0 in.
Y Location	0.0 in.
Z Location	44.0 in.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Tare File

NHTSA Benchmark tandem axle semitrailer - Tare file
42 ft trailer used in tractor semitrailer combinations only
Factors influencing heavy truck dynamic performance - NHTSA project
FileName = ST6T:Ta.Semi.Two

Semi Sprung Mass = 9000.0 lbs.
King Pin Setting = 36.0 in.

Rear Suspension

Number	WheelBase to (Location)
1	432.0

Center of Gravity Position	
Sprung CG Distance	216.0 in.
Sprung CG Offset .	0.0 in.
Sprung CG Height .	60.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia .	66667.0 in-lbs-sec^2.
Pitch Moment of Inertia	638021.0 in-lbs-sec^2.
Yaw Moment of Inertia	638021.0 in-lbs-sec^2.

Rear Hitch Location	
X Location	468.0 in.
Y Location	0.0 in.
Z Location	44.0 in.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Tare File

NHTSA Benchmark converter dolly - Tare file
Dolly used in Doubles and Triples combinations
Factors influencing heavy truck dynamic performance - NHTSA project
FileName = ST6T:Ta.Conv.Doly

Dolly Sprung Mass = 1000.0 lbs.

Rear Suspension

Number	WheelBase to (Location)
1	80.0

Center of Gravity Position	
Sprung CG Distance	80.0 in.
Sprung CG Offset .	0.0 in.
Sprung CG Height .	44.0 in.

Sprung Mass Moments of Inertia	
Roll Moment of Inertia .	1900.0 in-lbs-sec^2.
Pitch Moment of Inertia	2560.0 in-lbs-sec^2.
Yaw Moment of Inertia	2560.0 in-lbs-sec^2.

Rear Hitch Location	
X Location	80.0 in.
Y Location	0.0 in.
Z Location	44.0 in.

Notes:

1. Wheelbase-to and Hitch X-Location are, respectively, the longitudinal distances of the suspension's C.L. and the rear hitch C.L. measured from front axle on tractors.trucks, from front articulation point on dollies and semitrailers.
2. Hitch locations refer to 5th-wheel/turntable/pintle-hook/s on given Unit
3. Sprung CG Offset refers to lateral offset from the longitudinal centerline.
4. CG-Height and Z-locations are with reference to the ground.

Steering System File

Steering system used in all truck combinations
Factors influencing heavy truck dynamic performance - NHTSA project
ST6T:STEER.SYS

Steering Parameters	
Steering Ratio	28
Mechanical Trail ..	1 in.
Lateral Offset	3 in.
Steering Stiffness .	11000 in-lb/deg.
Tie Rod Stiffness .	11000 in-lb/deg.
Wrap Up Stiffness	150000 in-lb/in.

Wheel Brake file

Specifications for brakes used on tractor front axles
Factors influencing heavy truck dynamic performance - NHTSA project
Brake file
ST6T:WB.TKTR.FNT

Time Lag = 0.050 sec.
Rise Time = 0.250 sec.
Torque Coefficient = 1000.000 in-lb/psi.

Wheel Brake file

Specifications for brakes used on tractor rear axles
Factors influencing heavy truck dynamic performance - NHTSA project
Brake file
ST6T:WB.TKTR.R.1

Time Lag = 0.075 sec.
Rise Time = 0.250 sec.
Torque Coefficient = 1500.000 in-lb/psi.

Wheel Brake file

Specifications for brakes used on trailer/dolly axles
Factors influencing heavy truck dynamic performance – NHTSA project
Brake file
ST6T:WB.TRL.1

Time Lag = 0.175 sec.
Rise Time = 0.250 sec.
Torque Coefficient = 1500.000 in-lb/psi.

Spring File

Springs used on tractor front suspensions
Factors influencing heavy truck dynamic performance - NHTSA project
Spring files
ST6T:SP.TKTR.FNT
Coulomb Friction = 300.0 lbs.

Spring Table

Force [lbs]	Deflection [in]
-20000.0	-20.000
0.0	0.000
8645.0	7.200
25000.0	7.500

Spring Envelope

Compression Table		Extension Table	
Force [lbs]	Deflection [in]	Force [lbs]	Deflection [in]
-20000.0	-20.000	-20000.0	-20.000
0.0	0.000	0.0	0.000
9250.0	7.200	8040.0	7.200
25000.0	7.500	25000.0	7.500

Compression Coefficient = 0.080
Extension Coefficient = 0.080

Spring File

Springs used on single axle tractor rear suspensions
Factors influencing heavy truck dynamic performance - NHTSA project
Spring files
ST6T:SP.TKTR.R.1
Coulomb Friction = 1000.0 lbs.

Spring Table

Force [lbs]	Deflection [in]
-22500.0	-11.000
0.0	-0.900
0.0	0.000
3500.0	1.000
5750.0	1.500
8750.0	2.000
12250.0	2.500
16250.0	3.000
45000.0	4.000

Spring Envelope

Compression Table		Extension Table	
Force [lbs]	Deflection [in]	Force [lbs]	Deflection [in]
-26600.0	-11.000	-33000.0	-11.000
0.0	-1.000	0.0	-0.800
0.0	0.000	0.0	0.200
5300.0	1.000	4000.0	1.000
8650.0	1.500	6650.0	1.500
12650.0	2.000	10650.0	2.000
17300.0	2.500	15300.0	2.500
22600.0	3.000	20600.0	3.000
66000.0	4.000	53000.0	4.000

Compression Coefficient = 0.020
Extension Coefficient = 0.020

Spring File

Spring's used on a tractor's tandem axle rear suspension
Factors influencing heavy truck dynamic performance - NHTSA project
Spring file
ST6T:SP.TKTR.R.2
Coulomb Friction = 1000.0 lbs.

Spring Table

Force [lbs]	Deflection [in]
-22500.0	-11.000
0.0	-0.900
0.0	0.000
3500.0	1.000
5750.0	1.500
8750.0	2.000
12250.0	2.500
16250.0	3.000
45000.0	4.000

Spring Envelope

Compression Table		Extension Table	
Force [lbs]	Deflection [in]	Force [lbs]	Deflection [in]
-20000.0	-11.000	-25000.0	-11.000
0.0	-1.000	0.0	-0.800
0.0	0.000	0.0	0.200
4000.0	1.000	3000.0	1.000
6500.0	1.500	5000.0	1.500
9500.0	2.000	8000.0	2.000
13000.0	2.500	11500.0	2.500
17000.0	3.000	15500.0	3.000
50000.0	4.000	40000.0	4.000

Compression Coefficient = 0.050
Extension Coefficient = 0.050

Spring File

Springs used on single axle trailer and dolly suspensions
Factors influencing heavy truck dynamic performance - NHTSA project
Spring files
ST6T:SP.TRL.1
Coulomb Friction = 1000.0 lbs.

Spring Table

Force [lbs]	Deflection [in]
-40500.0	-11.000
0.0	-1.400
0.0	0.100
2812.5	0.500
7187.5	1.000
12175.0	1.500
17812.5	2.000
24062.5	2.500
56000.0	3.000

Spring Envelope

Compression Table		Extension Table	
Force [lbs]	Deflection [in]	Force [lbs]	Deflection [in]
-37000.0	-11.000	-44000.0	-11.000
0.0	-1.500	0.0	-1.300
0.0	0.000	0.0	0.200
3750.0	0.500	1875.0	0.500
8125.0	1.000	6250.0	1.000
13100.0	1.500	11250.0	1.500
18750.0	2.000	16875.0	2.000
25000.0	2.500	23125.0	2.500
62000.0	3.000	50000.0	3.000

Compression Coefficient = 0.020
Extension Coefficient = 0.020

Spring File

Spring's used on a trailer's tandem axle rear suspension
Factors influencing heavy truck dynamic performance – NHTSA project
Spring file
ST6T:SP.TRL.2
Coulomb Friction = 1000.0 lbs.

Spring Table

Force [lbs]	Deflection [in]
-32500.0	-11.000
0.0	-1.400
0.0	0.000
2531.0	0.500
6468.5	1.000
10968.5	1.500
16031.0	2.000
21656.0	2.500
50625.0	3.000

Spring Envelope

Compression Table		Extension Table	
Force [lbs]	Deflection [in]	Force [lbs]	Deflection [in]
-30000.0	-11.000	-35000.0	-11.000
0.0	-1.500	0.0	-1.300
0.0	0.000	0.0	0.200
3375.0	0.500	1687.0	0.500
7312.0	1.000	5625.0	1.000
11812.0	1.500	10125.0	1.000
16875.0	2.000	15187.0	2.000
22500.0	2.500	20812.0	2.500
56250.0	3.000	45000.0	3.000

Compression Coefficient = 0.050
Extension Coefficient = 0.050

Hitch File

Pintle hitch type articulation point
Factors influencing heavy truck dynamic performance – NHTSA project
Hitch file
ST6T:HI.PINTLE.HK

'A' Dolly Hitch

Hitch File

Fifth wheel type articulation point
Factors influencing heavy truck dynamic performance - NHTSA project
Hitch file
ST6T:HI.FIFTH.WL

Fifth Wheel Hitch

Suspension File

Tractor single axle rear suspension
Factors influencing heavy truck dynamic performance - NHTSA project
Suspension file
ST6T:SU.TKTR.R.1

Single Rear Suspension

Lead Axle	
Unsprung Mass	2300 lbs.
Roll Moment of Inertia ..	4458 in lbs sec^2
Center of Gravity Height	19.5 in.
Roll Center Height	29 in.
Track Width	72 in.
Dual Tire Seperation	13 in.
Spring Spread	38 in.
Auxiliary Roll Stiffness ..	6000 in-lbs/deg
Rollsteer Coefficient ...	0.000

Suspension File

Tractor's tandem axle rear suspension
Factors influencing heavy truck dynamic performance - NHTSA project
Suspension file
ST6T:SU.TKTR.R.2

Walking Beam Tandem Rear Suspension

Lead Axle	
Unsprung Mass	2300 lbs.
Roll Moment of Inertia ..	4458 in lbs sec^2
Axle Seperation	48 in.
Static Load Distribution .	50 %
Dynamic Load Transfer .	0%
Center of Gravity Height	19.5 in.
Roll Center Height	29 in.
Track Width	72 in.
Dual Tire Seperation	13 in.
Spring Spread	38 in.
Auxiliary Roll Stiffness ..	6000 in-lbs/deg
Rollsteer Coefficient ...	0.000

Suspension File

Single axle trailer and dolly suspension
Factors influencing heavy truck dynamic performance - NHTSA project
Suspension file
ST6T:SU.TRL.1

Single Rear Suspension

Lead Axle	
Unsprung Mass	1500 lbs.
Roll Moment of Inertia ..	4100 in lbs sec^2
Center of Gravity Height	19.5 in.
Roll Center Height	29 in.
Track Width	72 in.
Dual Tire Seperation	13 in.
Spring Spread	38 in.
Auxiliary Roll Stiffness ..	10000 in-lbs/deg
Rollsteer Coefficient ...	0.000

Suspension File

Trailer's tandem axle rear suspension
Factors influencing heavy truck dynamic performance – NHTSA project
Suspension file
ST6T:SU.TRL.2

Walking Beam Tandem Rear Suspension

Lead Axle	
Unsprung Mass	1500 lbs.
Roll Moment of Inertia ..	4100 in lbs sec^2
Axle Seperation	48 in.
Static Load Distribution .	50 %
Dynamic Load Transfer .	0%
Center of Gravity Height	19.5 in.
Roll Center Height	29 in.
Track Width	72 in.
Dual Tire Seperation	13 in.
Spring Spread	38 in.
Auxiliary Roll Stiffness ..	10000 in-lbs/deg
Rollsteer Coefficient ...	0.000

Suspension File

Tractor front suspension
Factors influencing heavy truck dynamic performance - NHTSA project
Suspension file
ST6T:SU.TKTR.FNT

Front Suspension	
Unsprung Mass	1200 lbs.
Roll Moment of Inertia ..	3719 in lbs sec^2
Center of Gravity Height	19.5 in.
Roll Center Height	23 in.
Track Width	80 in.
Dual Tire Seperation	0 in.
Spring Spread	32 in.
Auxiliary Roll Stiffness ..	1500 in-lbs/deg
Rollsteer Coefficient ...	0.000

Load File

Payload for a straight truck
Factors influencing heavy truck dynamic performance - NHTSA project
Payload file
Filename = ST6T:P1.St.Trck

Payload Weight = 28000.000 lbs.

Center of Gravity Distance = 220.630 in.
Center of Gravity Lateral Offset = 0.000 in.
Center of Gravity Height = 85.000 in.

Roll Moment of Inertia = 70000.000 in-lb-sec^2.
Pitch Moment of Inertia = 300000.000 in-lb-sec^2.
Yaw Moment of Inertia = 300000.000 in-lb-sec^2.

Notes:

1. CG-Distance is measured along x-axis, positive aft of front articulation point (front axle for Unit 1, king-pin for a semi, pintle-hook for a dolly).
2. CG-Height and Z-Locations are with reference to the ground.

Load File

Payload for a 27 ft semitrailer
Factors influencing heavy truck dynamic performance - NHTSA project
Payload file
Filename = ST6T:Pl.Semi.27

Payload Weight = 24600.000 lbs.

Center of Gravity Distance = 138.400 in.
Center of Gravity Lateral Offset = 0.000 in.
Center of Gravity Height = 83.920 in.

Roll Moment of Inertia = 65000.000 in-lb-sec^2.
Pitch Moment of Inertia = 375000.000 in-lb-sec^2.
Yaw Moment of Inertia = 375000.000 in-lb-sec^2.

Notes:

1. CG-Distance is measured along x-axis, positive aft of front articulation point (front axle for Unit 1, king-pin for a semi, pintle-hook for a dolly).
2. CG-Height and Z-Locations are with reference to the ground.