Pages 47 to 63

TRUCK DESIGN FACTORS AFFECTING DIRECTIONAL BEHAVIOR IN BRAKING

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ABSTRACT

With the advent of new truck regulations in the United States for braking in a turn, there is renewed interest in the directional behavior during braking. Asymmetries in the design and operation of trucks influence their ability to hold the desired path during braking maneuvers. This paper describes an analytical and experimental study of truck directional behavior in braking, examining the relative influence of various design factors that affect the behavior.

The analytical portion of the research was conducted using a modified version of the 3-dimensional model of the 2-axle truck incorporated in the TruckSim® software. The modifications allowed for asymmetry in the loading and brake operation and included a comprehensive steering model. The steering system includes compliance steer from the reaction of the tire forces and moments on the geometry of the road wheels, as well as steer from geometric sources such as bump (jounce), roll, front axle wrap-up, and frame compliance.

Experimental tests were conducted with a 2-axle truck modified to allow asymmetric loading in controlled patterns with the ability to distribute the load deceleration forces selectively between the frame rails.

Asymmetry in the steering system of trucks with I-beam front axles is seen in both the analytical and experimental tests to cause a deviation to the right in straight ahead open-loop tests even with symmetric loading and brake forces. Lateral offset in the load alters the directional behavior such that a load bias to the left causes even greater deviation to the right, and vice versa. Steer from frame shear deflections associated with the offset loads were found to have little influence on open-loop directional behavior. Good agreement was obtained between the model and experiments in closed loop tests of the steady state steering corrections necessary to maintain a straight path.

The simulation model was used to explore the influence on directional behavior from design and operating variables that could not be easily and reliably controlled on the test vehicle. Results are presented to show the influence of brake torque variations and steering system design properties.

Content Back

INTRODUCTION

For a number of reasons truck manufacturers have taken renewed interest in truck directional behavior during braking. In contrast to straight-line braking performance specified in past Federal Motor Vehicle Safety Standards for air-braked trucks, the new standards specify minimum braking performance on a 500 foot (152.4 m) radius turn [1]. In addition, with the continuing improvements in motor vehicle design customers are becoming more conscious of nuances in performance, such as directional deviations during braking. Thus, manufacturers are placing more emphasis on robust design that will reduce the sensitivity to factors that contribute to deviations during braking, such as asymmetries in design and operation.

It is common in the analysis of vehicle dynamic behavior to consider a vehicle to be symmetric in the lateral direction. In practice, precise symmetry is not achieved on trucks as a result of asymmetries in:

- Design,
- Performance of components,
- The environment, and
- Loading.

While the first three factors are known to the designer such that they can be accommodated in the design, the variations in loading while in use can be significant. Virtually all vehicles in use have some difference in mass distribution from side to side. However, offset loads are common in certain truck vocational applications, particularly the maintenance trucks used by the utility industries (power, telephone, etc.) as a result of side mounting of special hardware, such as aerial cranes.

Although the load asymmetries affect all areas of vehicle performance, they are particularly noticeable during braking. Lateral offset in the center of gravity (CG) of the vehicle produces a yaw moment about the CG during braking which results in a tendency for the vehicle to yaw in the direction of the light-side loading.

Little has been done to quantify the magnitude of these effects nor to determine how the sensitivity is affected by vehicle properties.

This paper describes an analytical and experimental study of truck directional behavior in braking, examining the relative influence of various design factors that affect the behavior. For this purpose we developed a computer simulation model for two-axle trucks, including the relevant features needed to examine the mechanics involved.

THE TRUCK MODEL

Numerous computer simulation models of trucks have been developed over the years. The various models for braking and handling developed at the University of Michigan Transportation Research Institute (UMTRI), culminating in the “Phase 4” model [2] have been largely replaced by the TruckSim® model [3]. TruckSim® relies on the same modeling approach used in earlier models, but incorporates a graphical user interface to simplify use of the models along with automated animation and plotting of results.

Briefly, the vehicle(s) are represented by rigid bodies for the sprung and unsprung masses, with solid axles and the UMTRI “spring” model [4] to duplicate the hysteretic behavior of leaf springs common on trucks. In order to explore factors that may interact with braking on vehicles with offset loads, the model (implemented in TruckSim®) was expanded to include effects as described below [5].

Offset Loads

We modified the TruckSim® program to allow for offset loads on a 2-axle truck because the asymmetry affects directional response during braking through several mechanisms.

1) Even with equal brake forces on left and right side tires of the truck, there will be a moment imbalance about the CG that can cause the vehicle to yaw during the stop in the absence of corrective steering.

2) The longitudinal forces produced by the offset load may not be equally distributed on the frame rails of the vehicle. The compliance of the frame permits longitudinal movement on one side of the frame relative to the other (parallelogramming) which in turn may steer the axles relative to the longitudinal axis of the vehicle.

3) Tire load balance (left to right) is upset by offset loads, and may change during braking as a result of forward load transfer. Since steering reactions are dependent on tire loads, steer inputs may be generated during braking.

Steering System

The steering system in TruckSim® is modeled as a compliant linkage subject to geometric steer effects, and acted upon by the forces and moments generated at the tire contact patches to cause compliance steer. An overview of the system is provided in the diagram shown in Figure 1.

The user may specify steering control input either as a table of steer angle versus time, or a path to be followed by the truck. Steering commands (through the steering wheel) are applied to the left wheel and transmitted to the right-side wheel through the tie-rod linkage. The system is modeled by a lookup table defining the left and right wheel steer angles at increments of steering wheel angle. The table models the effects of the ratio in the steering gear, Pitman arm, drag link, and steering arm; and it replicates the differential angles (Ackerman geometry) achieved via the tie-rod linkage (tie rod and tie-rod arms). The table is input with degrees steering wheel angle as the independent variable and the left and right wheel steer angles as dependent variables.

The path follower course is defined by the lateral position (Y) as a function of longitudinal distance (X) traveled (from start of the simulation run) that is to be followed by the front axle of the vehicle. A user-specified preview time represents the forward distance (in time) considered in determining the appropriate steering action, and a driver lag value characterizes the driver behavior [6].

Deviations of the steer angle due to geometry errors (bump, roll, wrapup, and frame compliance steer as described below) are computed and applied to the road wheels. In addition, the torque reactions about the kingpin axes arising from forces and moments in the tire contact patches are computed and applied to duplicate compliance steer.

Geometry Errors

Bump Steer—Because the steering command must transfer across the suspension via the drag link, steering inputs can be induced at this point arising from four effects. The first is bump steer, familiar to most truck engineers as the unintended steering effect associated with movement of the suspension in the jounce or rebound directions. Bump steer is replicated by specifying a bump steer coefficient (degrees of left wheel steer angle per inch of suspension jounce deflection).

Roll Steer—Steer deviations may also result from roll of the chassis on both the front and rear suspensions. On front axles, the effect is related to bump steer, but may also include some effects due to axle rotation in the horizontal plane as a result of differential suspension deflections during roll. Therefore, roll steer is modeled by including a roll steer coefficient defined as degrees steer per degree of frame roll relative to the axle for both the front and rear axles.

Wrapup Steer—A third steer effect arises from wrapup of the front axle (in the pitch direction) when the brakes are applied. The effect is modeled by a wrapup coefficient with units of degrees steer per degree of axle wrapup relative to the frame. Wrapup steer also requires calculation of the wrapup angle of the axle relative to the frame. The behavior is modeled by specifying a wrapup compliance for the axle, and using this in combination with the brake torque to determine the angle.

Frame Compliance Steer—A steer effect may arise from frame deflections in the horizontal plane when brake forces are not equal on both sides of the vehicle. The deflection, often called match-boxing or parallelogramming, causes front and rear axles to rotate relative to the longitudinal axis of the vehicle. The effective steer on the front axle is complicated by the fact that the mechanism interacts with the steering system such that the resulting steer angle is not directly related to yaw rotation of the front axle, but includes potential steering system interactions. Therefore, front wheel steer is again modeled by a simple coefficient defining degrees of steer per unit of total brake force differential between left- and right-side wheels.

Once these four steering deviations are calculated, they are summed and added to the steer angles of both the left and right front wheels. These represent a geometric intervention between the steering wheel and the road wheels accounting for steer angles that are slightly different than those commanded by the driver because of these effects.

Frame deflections also alter the alignment of rear axles causing a rear steer effect. This is modeled as a rear axle steer coefficient with units of degrees steer per unit of total brake force differential between the left and right wheels.

Steering Reactions

The front wheels experience forces and torques that act in combination against the compliances of the steering system to induce steer angles. The reactions from each force and moment are computed and applied to each wheel to determine the steer deviation that results. This deviation is then used to modify the steer angle determined from the effects listed above. Modeling for these reactions is described below.

Vertical Force—The vertical force acts normal to the road surface. The force produces a moment around the kingpin axis because the axis is not perpendicular to the road surface. The relevant dimensions and angles are shown in **Figure 2**.

During the simulation the model calculates dynamic values of the tire load, radius and axle attitude relative to the ground. Then to calculate the steering reactions from the vertical load, the simulation first determines the instantaneous scrub based on the current caster and lateral inclination angles. The component of the vertical force acting about the kingpin axis is the sine component for the total angle of the kingpin axis with respect to the vertical. The moment arm of the sine component about the kingpin axis is the cosine component perpendicular to the scrub arm.

Lateral Force—The lateral force acting on the tire produces a moment around the kingpin due to the longitudinal offset of the lateral force behind the kingpin ground intercept.

Longitudinal Force—The longitudinal (brake) force produces a moment about the kingpin due to the moment arm, which is the lateral scrub.

Aligning Moment—The cosine component of the aligning moment acts directly about the kingpin axis. Since the aligning moment is normal to the road surface the appropriate angle is the total angle of the kingpin.

Steer Deviations—The steering deviations arising from these reactions in the tire contact patch are equal to the sum of the torques times the compliance. The left wheel is exposed to the total torque from reactions in its own contact patch and those on the right wheel reacted through the tie-rod. The steering deviation of the right wheel is driven only by the torques on that wheel.

Once these calculations are completed the steering angle for each wheel is calculated by adding the command steer angle to the deviations resulting from bump steer, roll steer, wrapup steer, frame compliance steer, and compliance steer.

Brake System Model

The brake system model in the simulation replicates both air and hydraulic brake systems. In the generic brake system shown below(**Figure 3**), brake pedal input from the driver is translated into an application pressure sent to the individual brakes. The brake actuation accomplished by driver input at the brake pedal is defined in the simulation by the application pressure as a function of time. The brakes develop a torque applied to the wheels, producing a brake force at the ground and deceleration of the wheel. The functional difference between air and hydraulic brake systems is primarily a matter of the timing with which brake system pressure travels through the lines and applies the brakes.

Brakes and Proportioning

The brakes are modeled as a device that generates torque in the direction opposite to wheel rotation (thus they also work when the truck is traveling backwards). The torque is specified as a function of the application pressure. The brake torque is applied to the differential equation for the wheel spin degree of freedom, causing it to slow down. Subsequently, the tire model develops a tractive force opposing the brake torque.

A pressure-sensing proportioning valve (typically applied to the rear brakes) can be replicated by using a nonlinear brake lookup table. A pressure-sensing proportioning valve works to retard the further buildup of brake pressure to the selected brakes once the application pressure reaches a breakpoint value [7]. The influence of the proportioning valve is replicated by the shaping the brake torque lookup table appropriately. Typically it is modeled as a bi-linear table. The initial slope—from zero to the breakpoint pressure—is the "torque gain" of the brake. That is, it is a fundamental property of the brake reflecting the torque produced per unit of application pressure. Above the breakpoint pressure the slope of the line is reduced in accordance with the proportioning valve gain.

Anti-Lock Brake System

The model provides capability to replicate the behavior of a generic anti-lock brake system (ABS). Properties for the ABS are entered for each axle individually. The defining parameters are the wheel slip limit at which the brakes are turned off and the wheel slip limit at which they are reapplied.

SIMULATIONS

The computer simulation provides a powerful tool for confirming our understanding of the prevailing mechanics on a vehicle, as well as to quantify the relative influence of each mechanism. The simulation was first used to explore which were most important to directional behavior in braking, and then compared to the behavior experimentally observed on an actual truck.

It should be noted that even a truck that is "perfectly" balanced will not stop straight without steering corrections because of the asymmetry in the steering system. Of course, that can be rectified in the simulation model by specifying zero steering system compliance. That case was examined and served as the starting point to explore the magnitudes of the effects that would be present on real trucks.

Figure 4 illustrates one such study in which the simulation was operated in the path follower mode. Essentially, a straight path was specified and the driver model steering corrections to follow that path were examined.

The "perfect" truck in Figure 4 has complete symmetry and no steering system compliance allowing it to stop without necessity of steering corrections. A typical real effect in trucks is windup steer in the front suspension which may necessitate some steering correction when the brakes are applied. That case, denoted in the figure as "perfect w/0.77 windup steer" has a windup steer coefficient corresponding to 0.77 degrees of road wheel steer angle per degree of front axle windup in braking. During the stop it deviates to the left, requiring a 10 degree correction to the right at the steering wheel to maintain the straight path. When the 3600 kg ballast load is moved 350 mm toward the left side of the vehicle, it tends to veer to the right during the stop requiring a 30 degree steering correction to the left to keep on the path. Note that by adding 0.77 deg/deg of windup steer, the influence of the offset load is compensated and the vehicle stops straight with minimal steering wheel input.

The same conditions were simulated with a fixed steering wheel angle to examine the severity of the path deviation that resulted. The measures of performance for this maneuver are potentially the path deviation, yaw rate during the stop, or lateral acceleration experienced during the stop. **Figure 5** shows the lateral acceleration response for the same conditions listed in Figure 4. The "perfect" truck of course stops with no lateral acceleration present. The "perfect w/0.77 windup steer" experiences a lateral acceleration to the left, causing a deviation to the left. With the load offset to the left the vehicle experiences a lateral acceleration to the right, and a deviation in this direction. Using 0.77 deg/deg windup steer counterbalances the influence of the offset load, and a straight stop is obtained.

EXPERIMENTAL TESTS

A series of brake stop tests were conducted on a medium duty truck with loads offset laterally as a basis for verifying the accuracy of the simulation. Suspension properties and frame shear compliance of the truck were measured at UMTRI to acquire the necessary information to describe the truck in the simulation.

The photographs of the truck that follow show the configuration of the truck and an example of how the loads were offset during the test program.(Figure 6) (Figure 7)

The general specifications of the truck were:

Wheelbase	3.96 m (156 inches)
Front axle rating	3682 kg (8,100 lb)
Rear axle rating	8636 kg (19,000 lb)
Front brakes	Disc, 9038 N-m (80,000 in-lb) maximum torque
Rear brakes	Disc, 11,863 N-m (105,000 in-lb) maximum torque
Tires	225/80R22.5, Michelin XZU4 all weather

The following table describes the location of the weights on the test truck for the various cases run. The case numbers indicate the loading conditions discussed here. In the figures presenting the results a letter "A" following the case number denotes a closed loop test (driver is attempting maintain a straight path), and "B" denotes open loop, where the steering wheel is fixed in the straight-ahead position. Only cases relevant to this paper are described.

Table 1 Truck Loading Conditions.

Case No.	Description	Location of weights	Front Wheel Weights	Rear Wheel Weights
1	GVWR, No offset	3636 kg on each rail	1705R, 1686L	4264R, 4282L
2	Test weight, no offset	1818 kg on each rail	1523R, 1505L	2695R, 2714L
3	Test weight, base frame shear stiffness	3636 kg on each rail	1668R, 1373L	3327R, 2050L
5	Test weight, base frame shear stiffness	3636 kg on right rail	1409R, 1614L	2064R, 3336L
8	Test weight, high frame shear stiffness	3636 kg on left rail	1668R, 1373L	3327R, 2050L
10	Test weight, high frame shear stiffness	3636 kg on right rail	1409R, 1614L	2064R, 3336L
12	Test weight	3636 kg along centerline, attached to left rail	1523R, 1505L	4264R, 4282L
13	Test weight	3636 kg along centerline, attached to right rail	1523R, 1505L	4264R, 4282L

We designed the vehicle tests to evaluate effects of laterally offset loads in combination with frame shear stiffness. These were "open loop" tests, where the driver held the steering wheel fixed during the stops, and "closed loop" tests, where the driver attempted to maintain a straight line during the stop. The open loop tests are useful in determining vehicle characteristics and the closed loop tests are useful for determining which trucks are easiest for drivers to control.

We ran stops at a constant deceleration of 3 meters per second square (0.31 g) from 64 km/h to 43 km/h. Repeat tests were conducted in each direction along the test track and at various intervals during the day to identify any trends due to crosswind direction, brake temperature buildup and initial temperature. None of these effects proved significant, so the test data shown below is an average of 20 runs per case. To make sure there were no trends developing over time, we repeated case 3 after case 6 and repeated some cases for demonstration purposes after all data were collected. We collected the data on magnetic tape and digitized for computer analysis, although strip charts were also used so the technicians could be sure all channels were recorded properly.

The data recorded included:

- vehicle speed
- yaw velocity
- steering wheel angle
- brake pedal force
- longitudinal deceleration
- brake light switch (used to trigger the digitizing software).

For data analysis, steering wheel angle and yaw velocity were normalized to the deceleration level since deceleration varies slightly from one stop to the next. The normalized values were averaged and then summarized graphically.

COMPARISON OF RESULTS

The following plots compare test and simulation results. The first graph shows yaw velocity (per g of longitudinal deceleration) at the point 1 second after brake application. Positive yaw velocity corresponds to the truck deviating to the left.**(Figure 8)**

The model shows less bias to the right than the test vehicle. This may be due to some asymmetric characteristics of the vehicle not accurately reflected in the model. One example might be unequal brake torques (left versus right) in the test vehicle. Even with even braking, the model shows a bias to the right, which is expected due to asymmetry in the steering geometry and tie rod compliance.

From the driver's perspective, the following graph shows how much steering wheel motion (per g of deceleration) is required to hold a straight path for the closed loop cases.**(Figure 9)** The steering wheel angle is averaged over the deceleration period and normalized by the longitudinal acceleration. Test and simulation results are plotted together.

The trends are the same but the simulated vehicle has less bias to the right, which requires less steering to the left (or more steering to the right) to hold a straight path.

CONCLUSIONS

- With the steering wheel held rigid, laterally offset loads can cause a vehicle to turn toward the lighter side during modest deceleration (3 m/s/s) with a 3.96 m wheelbase.

- With a centered load, the vehicle may tend to turn toward one side during a stop. (See cases 1, 2, 12, and 13). This bias will be superimposed on the path deviation caused by laterally offset loads during brake stops. (See cases 3, 5, 8, and 10). This bias can be caused by brake/steer, bounce/steer, tie rod compliance, and brake imbalance.

- Both the simulations and the tests showed that steer effects due to the frame shear compliance of the test vehicle were negligible. (Compare case 3 to case 8 with weight on the left side and compare case 5 to case 10 for weight on the right side).

Introduction

Content

Introduction Content

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Introduction Content

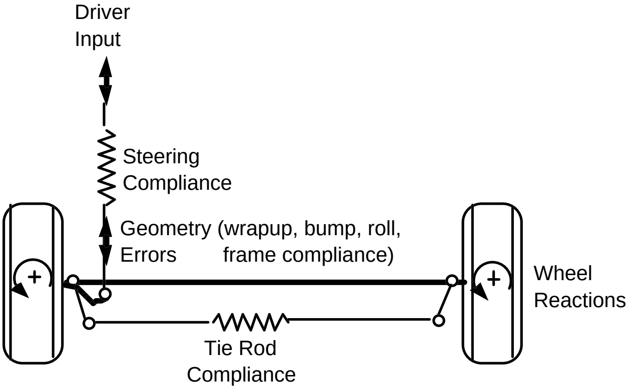

Fig. 1 Overview of the steering system model

Return to paper

Content

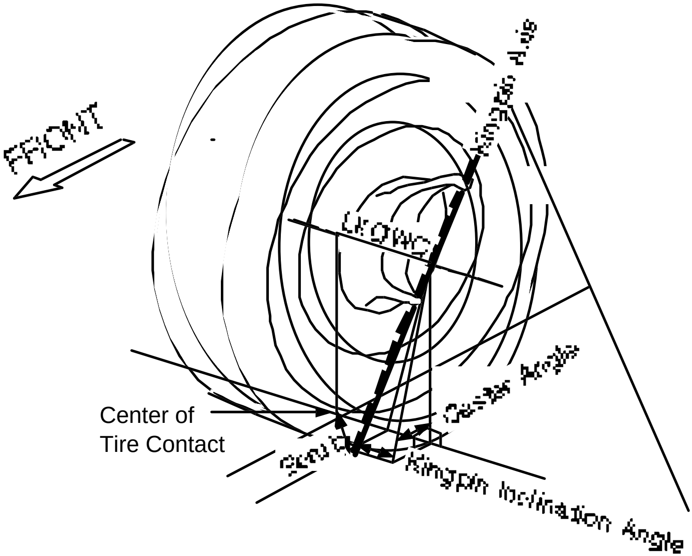

Fig. 2 Geometry at the right road wheel

Return to paper

Content

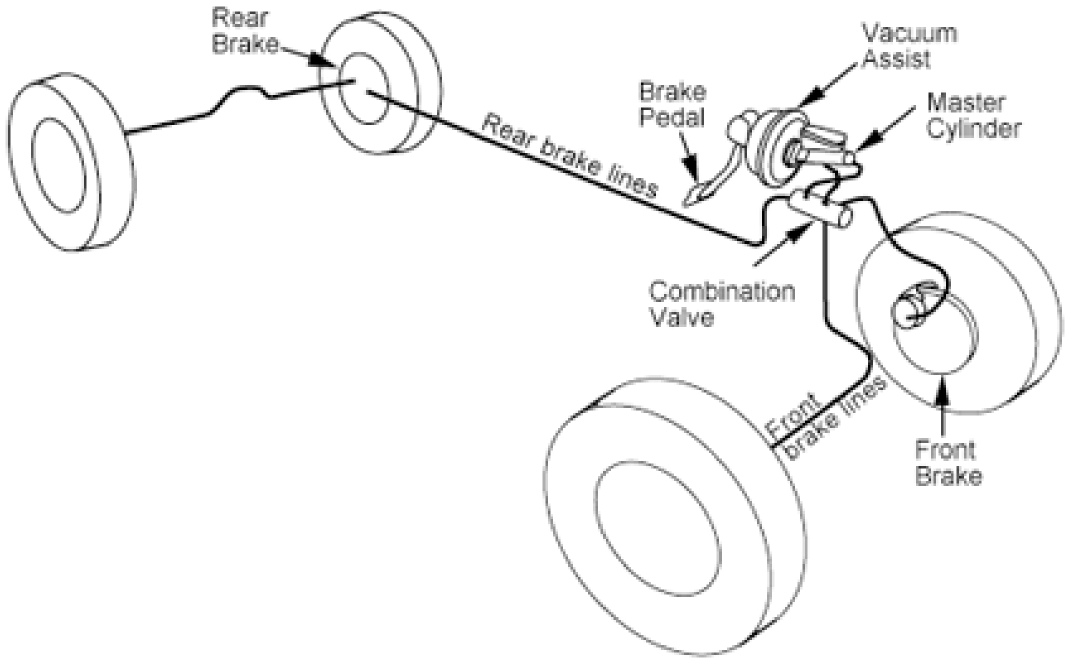

Fig. 3 Brake system layout

Return to paper

Content

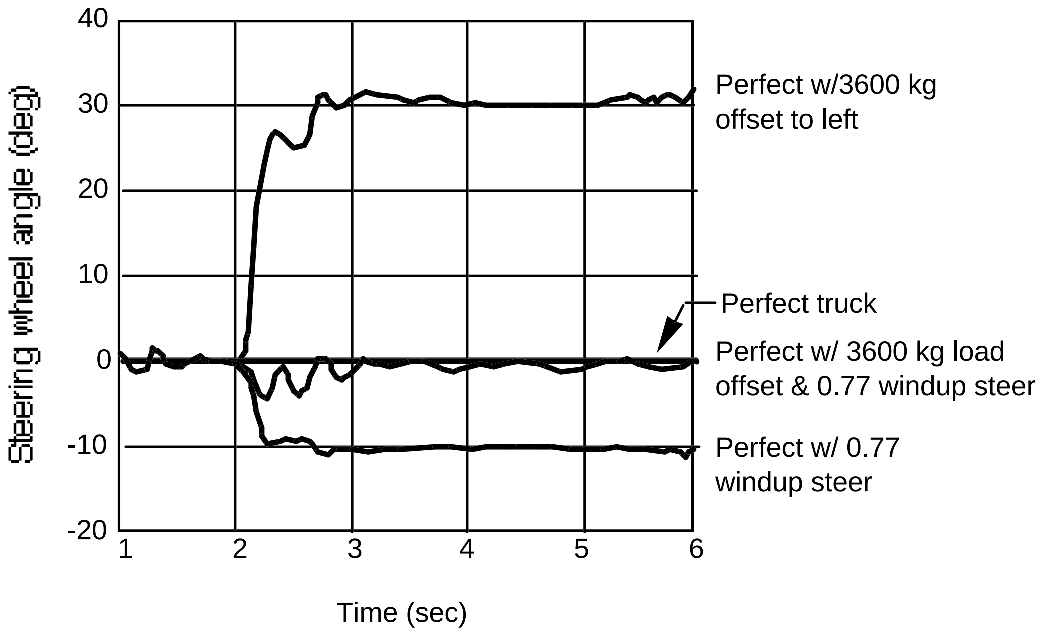

Fig. 4 Steering corrections necessary to follow a straight path (simulated)

Return to paper

Content

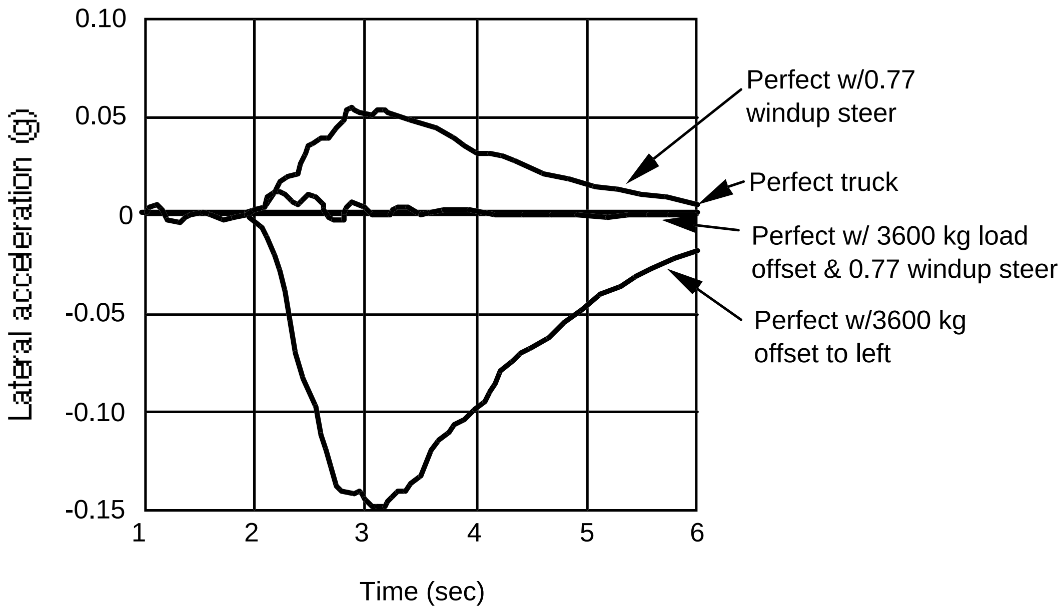

Fig. 5 Lateral accelerations with no steering corrections (simulated)

Return to paper

Content

Fig. 6 Front view of the test truck

Return to paper

Content

Fig. 7 The test truck with offset loads

Return to paper

Content

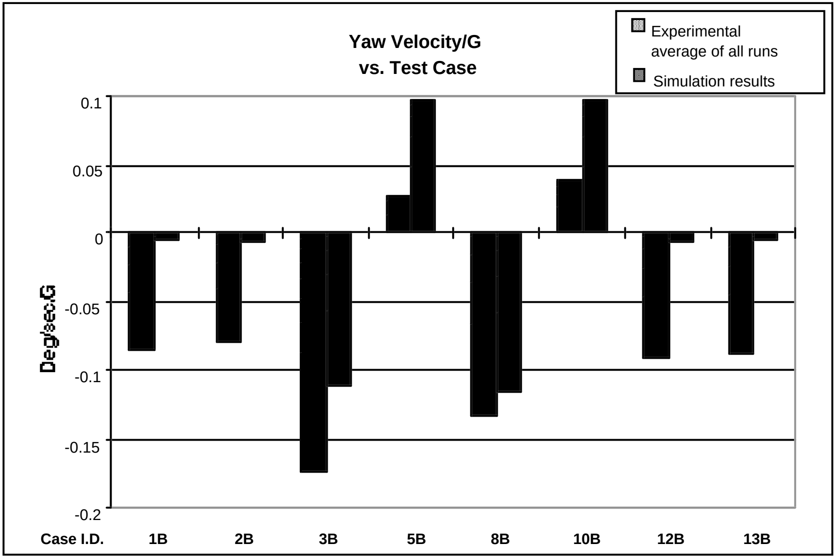

Fig. 8 Yaw response comparison of experimental and simulated open-loop tests.

Return to paper

Content

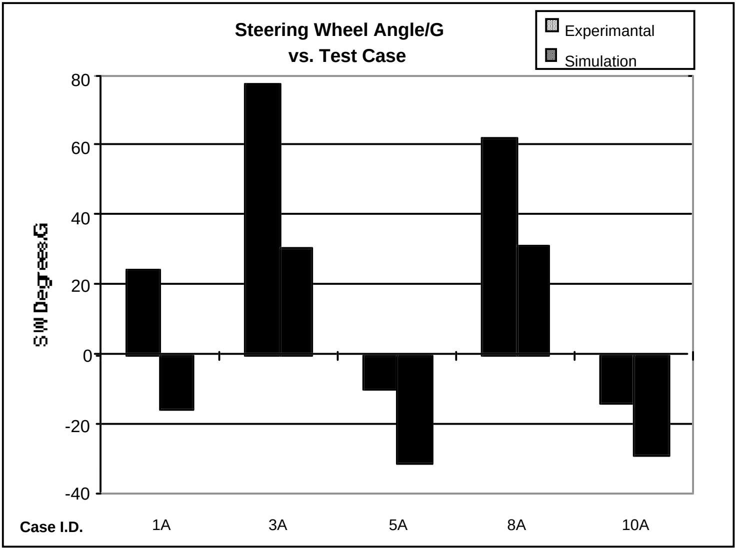

Fig. 9 Steering wheel angle comparison of experimental and simulated closed-loop tests.

Return to paper

Content