

## **Development of a Low Profile Work Zone Barrier Using Impact Finite Element Simulation**

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**SUBMITTED: NOV. 14, 2001**

**WORD COUNT : 5000 WORDS (EXCLUDING TABLES & FIGURES)**

**ABSTRACT**

A new low profile portable concrete barrier system developed for use in roadside work zone environments is presented. It is shown that by making extensive use of nonlinear dynamic finite element simulation, several cycles of conceptual design refinement can be accomplished using simulation rather than expensive full scale crash testing. Modifications to the suspension characteristics of the widely used C2500 reduced resolution finite element pickup truck model are presented. A discussion of the importance of properly accounting for frictional effects in vehicle impact simulations is given and example results for various frictional representations are presented. Full scale crash tests involving both a pickup truck and a compact car impacting the new barrier were successfully carried out. Despite its low profile, the barrier is able to successfully redirect a pickup truck without causing rollover. Additionally, because the new barrier utilizes inertial resistance to redirect impacting vehicles, its performance does not depend on anchorage of the barrier segments to the roadway.

## INTRODUCTION

In the past, the development of new roadside safety hardware systems such as railings, barriers, and energy absorbing crash cushions typically required iterative cycles of conceptual design and full scale vehicle crash testing. Much of the conceptual design process was based on sound understanding of engineering and mechanics principles, previous development experience, and intuition. However, with only these tools available, several cycles of concept development and crash testing were usually required to arrive at a successful design. Given the setup time and costs involved in full scale crash testing, and the matrix of testing conditions that must be examined for system approval (1), this process could be quite costly as well as time consuming.

Over the past decade, the roadside hardware design process has changed substantially as nonlinear dynamic finite element analysis (FEA) has found increasing utilization. A substantial portion of conceptual design can now be performed using computational simulation rather than experimental testing. Such use of simulation can result in more rapid system development, improved system performance, and reduced development costs. Ultimately, such systems must still be subjected to full scale crash testing to validate and certify their integrity, however far fewer cycles of development and testing are required if design flaws are identified based on simulation results.

This paper describes the process by which numeric simulation was recently used to carry out conceptual development of a new temporary concrete work zone barrier. Extensive use was made of the nonlinear explicit dynamic finite element simulation code LS-DYNA (2) to identify and correct design flaws early in the concept development stage. After arriving at what was considered to be a viable system design, full scale physical crash testing was successfully carried out to verify the performance of the system.

## EVALUATION OF A PROPOSED WORK ZONE CURB

A work-zone curb is a temporary barrier system used to separate traffic from individuals working in roadside work areas. The study described herein began when the authors were asked by the Florida Department of Transportation (FDOT) to evaluate methods for fastening a proposed low profile work zone curb systems to asphalt pavements (3). The proposed system consisted of multiple segments having a trapezoidal cross section and a vertical height of approximately 230 mm (9 in.). A thin coat of sprayed asphalt (i.e. a “tack-coat”) was going to be used to adhere the curb segments to the roadway. The goal was to quantify the shear strength requirements of the tack-coat. Impact conditions for which the curb system were to perform adequately were designated by the FDOT as a 2050 kg (4500 lbf) pickup truck striking the curb at 72 km/h (45 mi/h) at an angle of 15 degrees. If the tack-coat fastening method proved to have insufficient strength, then it would be the task of the researchers to develop a supplementary attachment system.

The low profile of the proposed curb system presented a separate set of potential problems however. Even if a fastening system could be developed which possessed sufficient strength, it would do little good if vehicles were able to cross the curb during impacts and enter the work zone. To investigate this possibility, multiple numeric simulations were conducted using two independent computer simulation programs to study the behavior of pickup trucks impacting the proposed curb. Initially, the highway vehicle object simulation model (4) software was used to simulate a typical Chevy C20 pickup truck impacting the curb at various speeds and

angles. Results from the HVOSM simulations revealed that the proposed 230 mm (9 in.) curb shape would not be able to reliably redirect an errant pickup truck away from the work zone even if the curb-to-roadway bond was infinitely strong.

This conclusion was further corroborated through simulations conducted using the nonlinear dynamic finite element analysis code LS-DYNA. Using a modified version of a public domain finite element model of a Chevy C2500 pickup truck (discussed in further detail later in this paper), the HVOSM results were confirmed by LS-DYNA simulations. Based on these results, the focus of the research effort was redefined. The new focus was to develop a portable concrete work zone barrier that could meet the following criteria.

- *Crash Test Validation* – System must meet the NCHRP 350 Level 2 requirements for longitudinal barriers. This requires full scale crash tests at 70 km/h (45 mi/h) for both a small car (820 kg, 1800 lbf, 20 degree impact angle) and a standard pickup truck (2000 kg, 4400 lbf, 25 degree impact angle).
- *Portable and Modular* – System must be composed of easily movable units that can be assembled in the field and modularly replaced as necessary (e.g. after an impact).
- *Low profile* – System must have a low profile (short height) to allow drivers unhindered visibility of pedestrian and vehicle cross traffic.
- *Minimal anchorage* – Performance of the system should not be highly dependent on anchorage of the barrier to existing pavement.
- *Minimal work zone intrusion* – System should allow minimal lateral barrier deflection into the work zone during an impact.

In order to design a barrier system that could achieve these goals with minimal crash testing costs, extensive use was made of the finite element simulation code LS-DYNA.

## **FINITE ELEMENT MODELING TECHNIQUES**

For a longitudinal barrier design to be acceptable, it must be capable of smoothly redirecting errant vehicles of various sizes away from the work zone being protected. NCHRP 350 requires crash tests be conducted using both a 2000P vehicle (a 2000 kg pickup truck) and an 820C vehicle (an 820 kg compact car). The 2000P vehicle delivers more energy on impact than does an 820C and has a higher center of gravity making it more prone to rollover. As a result, primary focus in the redefined FDOT research effort was placed on designing a new barrier system to successfully resist impact by a 2000P truck. It was assumed (correctly as will be demonstrated later) that if the barrier could pass crash testing with a 2000P truck, it would also pass testing with an 820C car.

### **Suspension modifications made to the NCAC C2500 finite element model**

Since the Chevy C2500 pickup truck is the most common vehicle used in 2000P tests, extensive use of a finite element model of a C2500 truck was used in this research. The National Crash Analysis Center (NCAC) has developed a number of finite element models for common test vehicles and has released these models into the public domain. The model used in this research was a modified version of the NCAC Chevy C2500 reduced resolution (version 8) pickup truck

model (5). The original C2500 model developed by NCAC has a great deal of merit and has been widely and successfully used by the roadside safety community for several years. However, in the course of conducting curb and barrier impact simulations, the authors identified some aspects of the model that could be improved. These changes enhance the applicability of the model to impacts involving significant deformation of the front suspension springs.

Specifically, in the original NCAC model the stiffness of the front suspension coil springs is too small, the pre-compression force in the springs (needed to account for gravity loading) is not present, and the specifications of contact surface definitions in the vicinity of the front suspension control-arms can be improved (3,6). These aspects of the model are not readily evident in impact conditions where the front suspension deformations are not of great significance. For example, in frontal or oblique impacts on vertical rigid walls, vehicle response is affected by structural deformation of the vehicle body more than by compression and recoil of the suspension springs. However, for low profile barriers and curbs, the deformation of the suspension springs is important in determining redirection and trajectory of the vehicle during and after impact.

Shown in Figure 1 are two views of the front wheel and suspension assembly of the reduced resolution C2500 truck model. Each front suspension assembly is composed of two control-arms: an upper and a lower rotating control arm. Discrete spring and damper elements connect the two control-arms together to simulate the effects of the coil springs and shock absorbers. Spherical joints connect the control-arms to the knuckle of the wheel assembly and revolute joints connect the arms indirectly to the chassis rail (allowing the arms to rotate about the revolute joints). In the NCAC model, the discrete spring elements have a stiffness of 14.4 N/mm (82.2 lbf/in) and zero initial pre-compression (i.e., no initial “spring offset”).

However, in a real truck under normal gravity loading, the front suspension *springs* (not the tires) of the vehicle will see approximately 7750 N (1742 lbf) of force. For the stiffness specified, this would require 538 mm (21.2 in.) of spring deformation to reach equilibrium with gravity loading. The reason that analysts using this NCAC model generally do not observe such large spring deformations under combined gravity and impact loading is revealed in Figure 2. The control-arms are attached to rigid plates through the revolute joints previously mentioned. These rigid plates are then attached to the deformable chassis rail of the pickup. However, there is only a very small clearance between the control-arms and the rigid plates. As soon as gravity is applied and the arms attempt to swing upward, they immediately come into contact with the rigid plates thus stopping the upward arm motion and preventing the discrete spring elements from being deformed. The vehicle comes into equilibrium with gravity not by compressing the springs, but by resting on the rigid plates. This is an acceptable condition for some types of impact conditions because the center of gravity (CG) of the vehicle is at the correct elevation. In many situations, the vertical position of the CG and deformation of the vehicle body (but not the wheels and suspension) are the primary factors affecting the outcome.

This modeling technique, however, is not appropriate for impact problems involving significant vertical movement of the front wheels (and thus compression and subsequent extension of the coil springs). For example, when a front wheel strikes a curb during an impact event, the suspension control-arms should rotate upward and compress the coil spring. It is clear that the original NCAC model was not intended for this type of application because the presence of the rigid plates prevents correct rotation of the control-arms. To enhance its applicability to impact conditions of this type, several modifications were made to the NCAC vehicle model. First, the contact definitions specified between the control-arms and the rigid plates were

removed from the model. By removing these contact definitions, the control-arms can now pass “through” the rigid plates. Effectively, the rigid plates are now only used as a connection point for the revolute joints but no longer prevent rotation of the control arms. Next, laboratory tests were conducted on typical C2500 coil springs to determine their actual stiffness. Results from the tests indicated that the load-deflection relationship for the coil springs was linear and that the linear stiffness was 166.5 N/mm (951 lbf/in).

Finally, the spring pre-compression required to balance gravity loading was determined. In the NCAC C2500 model, the position and orientation of the wheel and suspension components are modeled to match the *gravity-equilibrium configuration* of the vehicle. For example, the position of each front wheel relative to the vehicle body matches the position that the wheel would have if the truck were sitting at rest on a flat surface while subjected to standard gravitational body forces. In this configuration, the compressive force in each front suspension coil spring must be such that it produces the necessary equivalent force *at the centerline of the tire* so as to balance the reaction on the tire due to gravity loading. An initial discrete spring element “offset” (in LS-DYNA terminology) can be used to produce the required initial spring compressive force. It was determined that an initial compressive offset (i.e. compressive spring deformation) of 46.5 mm (1.83 inches) was required produce the spring force corresponding to the gravity equilibrium position.

### **Frictional Effects in Finite Element Impact Simulation**

In addition to the suspension modifications described above, the authors also gave special attention to the representation of frictional effects. Frictional contact forces developed between tires and barrier contact surfaces can affect whether or not a particular barrier system will successfully redirect an impacting vehicle. Predictions of vehicle redirection obtained from finite element simulation are influenced by several factors—frictional effects, barrier impact face geometry, barrier stiffness, barrier inertial properties, finite element representations of vehicle components, and selection of contact algorithms. Among these factors, the representation of friction is often given inadequate attention.

In the present study, individual contact and friction definitions were specified for steel-to-steel contact, steel-to-concrete contact, barrier-to-roadway contact, tire-to-roadway contact, and tire-to-barrier contact. Particular attention was given to modeling friction between the vehicle tires and the barrier segments (tire-to-barrier contact). In general, the frictional force between two bodies in contact is determined based on the normal force and coefficient of friction ( $\mu$ ) that are active at the contact zone. The value of  $\mu$  may be numerically modeled either as a fixed constant value or as a function of the velocity at which the surfaces slide past one another ( $v_{sliding}$ ). However, using the latter representation matches more closely with the viscous nature of frictional development in materials such as vehicle tires. Also, when a rotating tire impacts barrier at an oblique angle, different locations on the tire slide against the barrier at different velocities (especially in cases involving significant tire sidewall scrubbing). Thus, using a velocity dependent (i.e. viscous) frictional model is most appropriate, although constant friction coefficients are often successfully used for cases involving very low profile systems (e.g. short curbs).

In the present study, experimental data obtained from the literature were used to develop velocity dependent friction models for representing tire-to-barrier contact. Yager et al. (7) reported test results for tire-to-concrete friction measurements and discussed the influence of

parameters such as sliding speed, type and amount of surface contaminants, tire characteristics, and temperature. Their test data indicate that tire friction is sensitive to surface contamination, particularly for tire friction on dry concrete. Data presented by Gillespie (8) further corroborate sensitivity to surface condition. Viscous friction is also affected by tire properties such as tread pattern and hardness (9), roughness of the roadway surface (e.g. concrete texture) and tire wear (10).

Given this variability, the authors' recommendation is to bracket the expected range of frictional values with upper and lower bound limits and then to run simulations at each of these limits to evaluate system performance. If a constant friction model is being used, then establishing upper and lower bound limits amounts to choosing scalar values of  $\mu_{upper}$  and  $\mu_{lower}$ . For the system being developed in this study however, it was anticipated that during vehicle impacts, significant scrubbing contact would occur between the vehicle tires and the barrier. For this reason, a velocity dependent friction model was deemed to be more appropriate than a constant friction model. Using data for tire-to-concrete friction obtained from the references cited above and accounting for the typical dusty characteristics of roadside work zones, upper and lower bound friction curves were established for use in this study. The functional form of these curves is given by :

$$\mu = \alpha + \beta^{-\gamma(v_{sliding})} \quad (1)$$

where  $\alpha$ ,  $\beta$ , and  $\gamma$  are fitting parameters chosen to produce the desired velocity dependency of the friction coefficient. Figure 3 illustrates the upper bound ( $\mu_{curve-A}$ ) and lower bound ( $\mu_{curve-B}$ ) curves used herein.

Recognizing that the fundamental role of a longitudinal barrier is to *redirect* errant vehicles, consider the following example. A 2000P vehicle impacts a 225 mm (9 inch) work zone curb at a 15 degree impact angle. Two simulations are performed: one in which tire-to-curb friction is represented using a simple, constant value of  $\mu = 0.1$  and another in which the upper bound curve of Figure 3 ( $\mu_{curve-A}$ ) is used. In both cases, the bond between the curb and the roadway is assumed to be perfect, i.e. the curb itself does not slide. Predicted vehicle trajectories for both cases are illustrated in Figure 4. The first case predicts successful redirection of the vehicle. In contrast, the second case predicts that the vehicle over-rides the curb—a condition that is unacceptable if the curb is intended to protect the work zone. This example clearly points out that the magnitude of friction introduced into a simulation—whether introduced through the use of a constant friction model or a velocity dependent frictional model—can significantly alter the predicted results. Conducting separate impact simulations for a system using appropriately chosen upper and lower bound friction models can help ensure that the full range of post-impact outcomes has been considered.

Figure 5 expands on this example by illustrating the kinetic and internal energy histories corresponding to the two simulations cited above. It is evident that more energy is dissipated in the case where a higher level of friction is introduced (i.e.  $\mu_{curve-A}$  in Figure 3). In this case, the kinetic energy drops much faster than the internal energy rises—the difference being primarily attributable to the energy dissipated by frictional forces. In contrast, the case where  $\mu = 0.1$

indicates only a moderate drop in kinetic energy and relatively less energy dissipation. Differences in energy dissipation are very important because the segment-to-segment joint forces in sliding barrier systems (such as the one developed in this study) are substantially influenced by energy dissipation during the impact.

### **Modeling Limitations**

The C2500 suspension modifications and the newly developed friction model both represent improvements in the ability to perform accurate vehicle-to-barrier impact simulation. However, limitations still exist and should be noted. For example, in the reduced resolution C2500 vehicle model the control-arm assemblies are modeled using rigid material properties whereas in reality these are steel components capable of deformation under severe impact loading. In addition, the revolute and spherical joints that connect the components of the suspension system together do not account for the possibility of joint failure. Future efforts should focus on improving these aspects of the suspension model.

In regard to modeling frictional effects, limitations also remain. Frictional forces between contacting bodies depend not only on the coefficient of friction  $\mu$ , but also on the normal force acting on the contact surfaces. The friction model presented herein addresses the computation of  $\mu$ , but not the normal force. The magnitude of the normal force is a function of the stiffnesses of the bodies in contact (e.g. the tire and the barrier). It must be noted that the tire modeling techniques used in the C2500 model were never intended to be highly accurate. Instead, the tires are modeled using coarse, approximate meshes of shell elements that do not accurately represent the tire tread or sidewall stiffnesses. To further improve the modeling of frictional effects, the tires in the C2500 model should be modeled with a greater level of accuracy.

### **NEW BARRIER DEVELOPMENT USING FEA IMPACT SIMULATION**

Using the modified C2500 model and the new friction relationships just described, the authors began the process of developing a new low-profile portable concrete barrier system that would meet the design criteria previously outlined in this paper. Each cycle of the iterative development process consisted of a concept development followed by extensive computer simulations of the proposed design.

Prior to starting the development process in full, a literature review was conducted to determine if any presently available system would satisfy the required design criteria. Of all the systems reviewed, the segmental concrete barrier shape reported by Guidry and Beason (11) was given the most serious consideration as it was reported to have performed well in crash tests involving pickup truck impacts. It was also portable in the sense that it consisted of segments that would be connected together in the field. However, due to their cross-sectional shape and length, the weight of each concrete segment was deemed to be too heavy by FDOT. After reviewing several other systems, it was concluded that no currently available barrier system would meet all of the design criteria specified. Development of a new system was thus undertaken.

The first concept developed consisted of concrete segments similar to but lighter and shorter than those of Guidry and Beason. Their system consists of segments that are 508 mm (20 in) tall, 6.1 m (20 ft) long, have an inverted trapezoidal cross sectional shape (see Figure 6a), and weigh 49 kN (11 kips) each. The first concept developed by the authors used segments that were 457 mm (18 in) tall, 3.7 m (12 ft) long, had a truncated slanted triangular cross section (see

Figure 6b), and weighed only 21 kN (4.6 kips) each. Computer simulations conducted for the new system indicated suitable redirection capability when impacted at 72 km/h (45 mi/h) and 25 degrees by a 2000P vehicle. Initially the new design called for connection bolts installed through blocked out sections at the ends of the barrier segments. However, after building prototype formwork for this design, it was concluded that there was excessive reinforcing steel congestion and that a new connection method was needed.

A new connection detail in which external brackets were attached to the back (sloping) faces of the segments was then explored (see Figure 6c). By running connection bolts through external brackets on each set of adjacent barrier segments, the bolt block-outs were eliminated and the reinforcing cage was simplified. However, subsequent simulations revealed a serious flaw in this design—the bolts spanning between the brackets do not transfer adequate shear force. As a result, during an impact the barrier segments slide relative to one another sufficiently to create snagging points. Figure 7 shows a simulation in which portions of the vehicle cabin snag on protruding downstream barrier segments. The vehicle undergoes a harsh deceleration (resulting in a rapid decrease in longitudinal velocity between  $t = 0.32$  sec. and  $t = 0.38$  sec.) and virtually comes to a stop before finally separating from the snag point. Due to the potential for this type of snagging, this design concept was abandoned.

Eventually a final design evolved in which the connection bolts were still near the back face of the barrier but were now embedded in the concrete cross section. Figure 8 presents both detailed design drawings and photographs of the system. A load transfer assembly was designed that takes the load from the connection bolt and transfers it to the concrete via shear studs and additional embedded bolt length. This connection design is capable of carrying the tensile loads in the bolts but is also able to transfer shear from one segment to the next during impact, thus eliminating the snagging problem. In addition, the connection bolts provide sufficient system stiffness during impact such that the inertial resistance of the system *as a whole* is sufficient to redirect errant vehicles. Anchorage of the barrier system to the existing pavement is unnecessary, greatly simplifying field installation of the system.

A comprehensive series of impact simulations were performed using the final design concept in an attempt to ensure success during full scale crash testing. Simulations were performed using both the upper and lower bound friction curves of Figure 3 to evaluate the redirection capability of the barrier and to determine the critical design forces for the connection bolts and load transfer assemblies.

## FULL SCALE VALIDATION AND CRASH TESTING

In order to validate the field performance of the proposed system, which was developed based purely on finite element simulations, two full scale crash tests were performed. As recommended by NCHRP 350, the barrier assembly was impact tested using both an 820C compact car and a 2000P pickup truck. Ten barrier segments were fabricated and connected together to form a test article that was 36.6 m (120 feet) in length. This choice of system length was fully justified by the test results as the segments at both ends of the system did not displace at all during either the 820C or 2000P impact tests. The actual impact tests were conducted at the facilities of E-Tech Testing Services located in Rocklin, California, USA.

Results from the full scale tests are shown in Table 1. In both tests, the test vehicle exited the barrier at a very small angle and with minimal damage to the vehicle (both vehicles sustained a single suspension joint failure during impact). Lateral deflection of the barrier into a hypothetical work zone located behind the system was less than 191 mm (7.5 inches) despite the

fact that the barrier was not anchored to the roadway. Frictional resistance between the bottom of the barrier and the roadway at the test site represented worst case conditions as the roadway surface consisted primarily of loose chipped asphalt with a low coefficient of friction. Occupant risk measures for both tests were within allowable limits. Even with its low profile, the barrier was able to redirect the 2000P vehicle without the occurrence of rollover, barrier connection failure, or snagging.

Figure 9 shows a comparison of crash test results and simulation results (for  $\mu = \mu_{curve-A}$ ) for the 2000P impact condition. The two sets of results are generally in good agreement with the overall vehicle motions matching reasonably well and the exit angles being very similar. The most notable differences are in the behavior of the front tire during impact and the maximum roll angle sustained. In the simulation, the front impact tire steers more excessively than in the crash test resulting in a longer duration of contact between the tire and the barrier. As a result, more kinetic energy is dissipated through friction in the simulation and a smaller maximum roll angle is predicted. The steering angle discrepancy is believed to be caused by the properties of the steering control mechanism present in the model and is being investigated. Lateral barrier deflections predicted by the upper and lower bound simulations bracket the crash test results closely, and justify the methodology of system design based on simulation followed by crash test validation.

## CONCLUSIONS

By making extensive use of finite element impact simulation, a new low-profile portable work zone barrier system has been successfully developed and tested for the FDOT. Several cycles of design iteration were performed based purely on computational simulation thus substantially reducing both the time and costs associated with development and testing of the system. Modifications to the NCAC C2500 reduced resolution pickup truck model were made to expand its applicability to impacts involving significant movement of the front suspension assemblies. Appropriate consideration of frictional effects using upper and lower bound parameters was discussed and demonstrated.

Full scale crash tests involving both 2000P and 820C vehicles were successfully carried out and good agreement between the 2000P simulation results and crash test results were observed. Despite its low profile, the barrier was able to successfully redirect even the 2000P vehicle without causing a rollover. Additionally, because the new barrier utilizes inertial resistance to redirect impacting vehicles, its performance does not require anchorage of the barrier segments to the roadway.

## ACKNOWLEDGEMENTS

The authors wish to thank the Florida DOT for providing the financial support that made this research and development program possible. The material presented in this paper is based upon research that was supported by the FDOT under Contract Number BB-894. Special thanks are extended to John Shriner of the FDOT, Dhafer Marzougui of the National Crash Analysis Center (NCAC), and John LaTurner of E-Tech Testing Services for their assistance.

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**FIGURE 6 Barrier design concepts considered.**

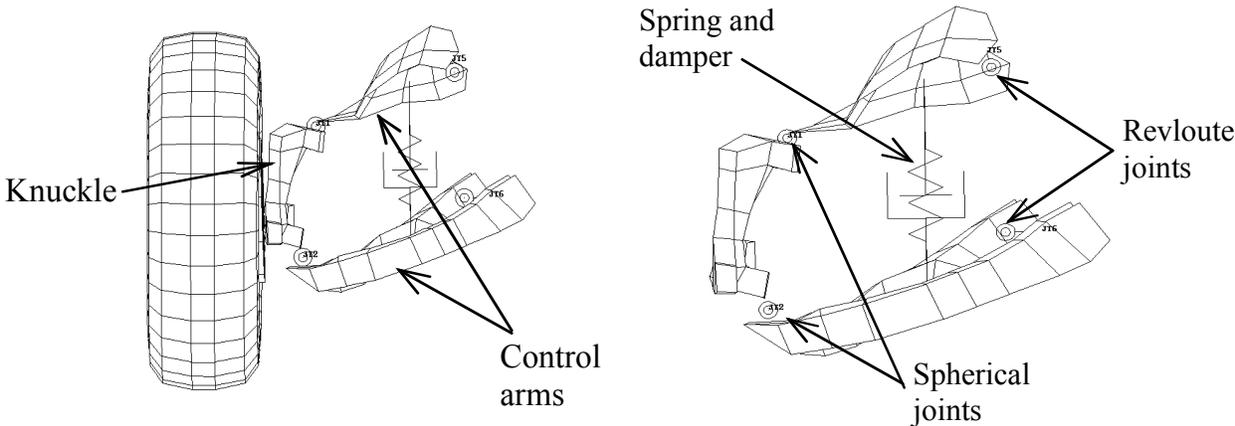
**FIGURE 7 Vehicle snagging due to relative sliding motion between barrier segments for connection system shown in Figure 6c**

**FIGURE 8 Final barrier design.**

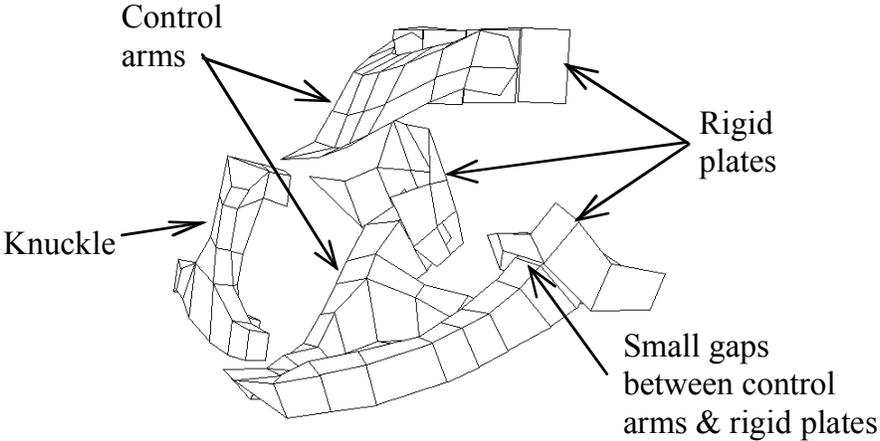
**FIGURE 9 Comparison of crash test results and simulation results for 2000P vehicle.**

**TABLE 1 Summary of crash test results and simulation results.**

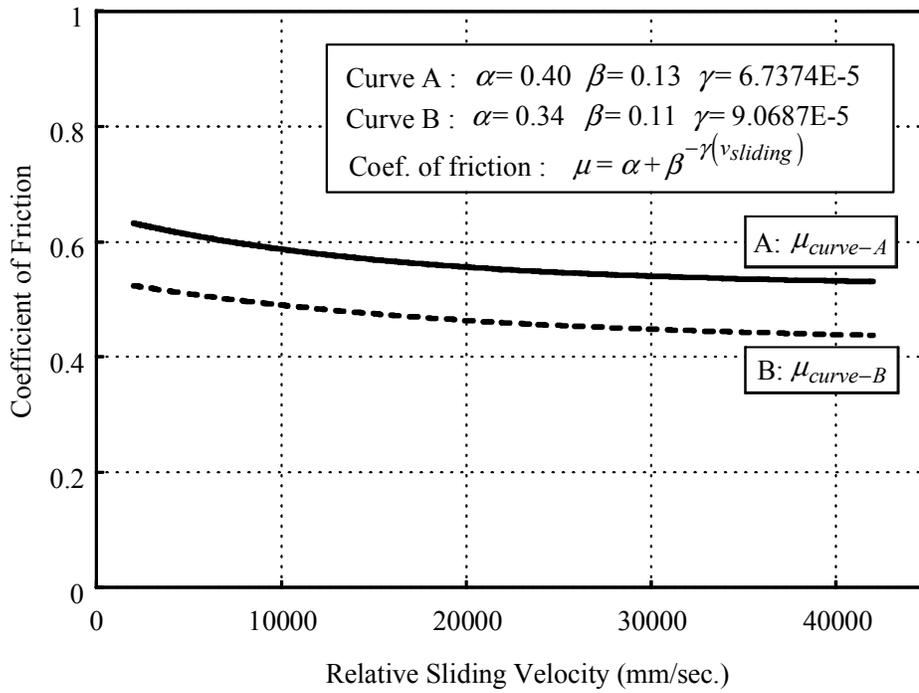
Test parameter	820C Test (Geo Metro car)	2000P Test (Chevy C2500 pickup)	2000P Simulation (upper bound friction)	2000P Simulation (lower bound friction)
Impact speed	70.6 km/h (44 mi/h)	68.0 km/h (42.3 mi/h)	68.0 km/h (42.3 mi/h)	68.0 km/h (42.3 mi/h)
Impact angle	20 deg	25 deg	25 deg	25 deg
Vehicle exit speed	48.6 km/h (30.2 mi/h)	43.6 km/h (27.1 mi/h)	42.3 km/h (26.3 mi/h)	44.3 km/h (27.5 mi/h)
Vehicle exit angle	5 deg.	0 deg.	3.0 deg	2.5 deg.
Dynamic deflection	64 mm (2.5 in.)	190 mm (7.5 in.)	174 mm (6.9 in.)	327 mm (12.9 in.)
Permanent deflection	51 mm (2.0in.)	165 mm (6.5 in.)	157 mm (6.2 in.)	280 mm (11.0 in.)
Max. roll angle	-4.3 deg.	29.7 deg.	7.0 deg.	8.4 deg.
Max. pitch angle	-6.2 deg.	-12.1 deg.	-1.1 deg.	-1.3 deg
Max. yaw angle	-58.6 deg.	-25.1 deg.	-27.5 deg.	-29.1 deg.
Occupant longitudinal ridedown acceleration	-3.6 g's	-3.7 g's	(Not computed for simulations)	
Occupant lateral ridedown acceleration	-10.6 g's	-6.3 g's		
Occupant longitudinal impact velocity	3.6 m/s (11.8 ft/s)	4.0 m/s (13.1 ft/s)		
Occupant lateral impact velocity	-4.3 m/s (-14.1 ft/s)	-3.7 m/s (-12.1 ft/s)		



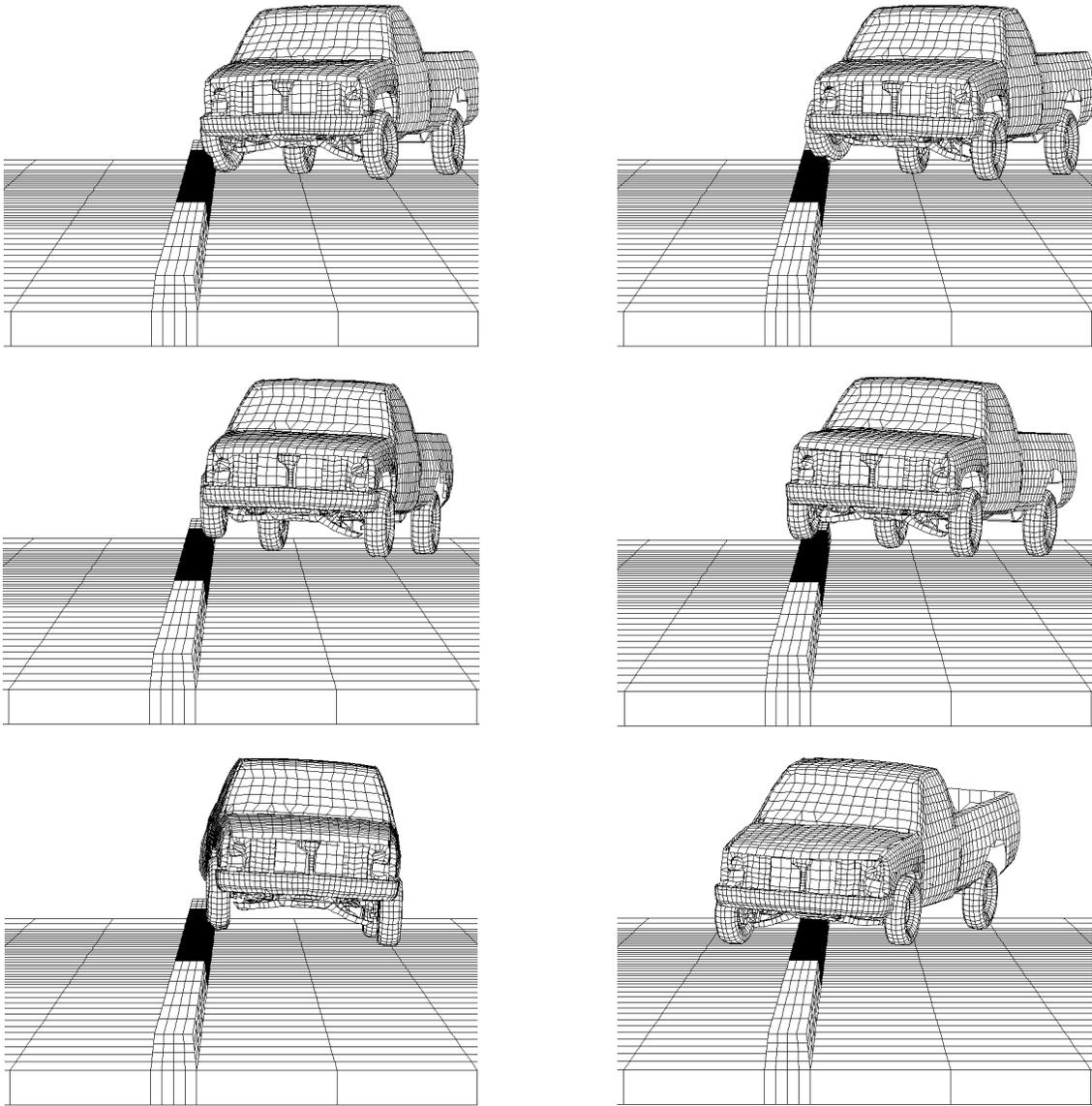
**FIGURE 1 Front suspension and wheel assembly of the NCAC reduced C2500 model.**



**FIGURE 2 Rigid plates that prevent rotation of front suspension control arms.**



**FIGURE 3 Upper and lower bound frictional relationships for tire-to-barrier contact.**



a) Friction model :  $\mu = 0.1$

b) Friction model :  $\mu = \mu_{curve-A}$

**FIGURE 4 Curb impact simulation results for differing frictional models.**

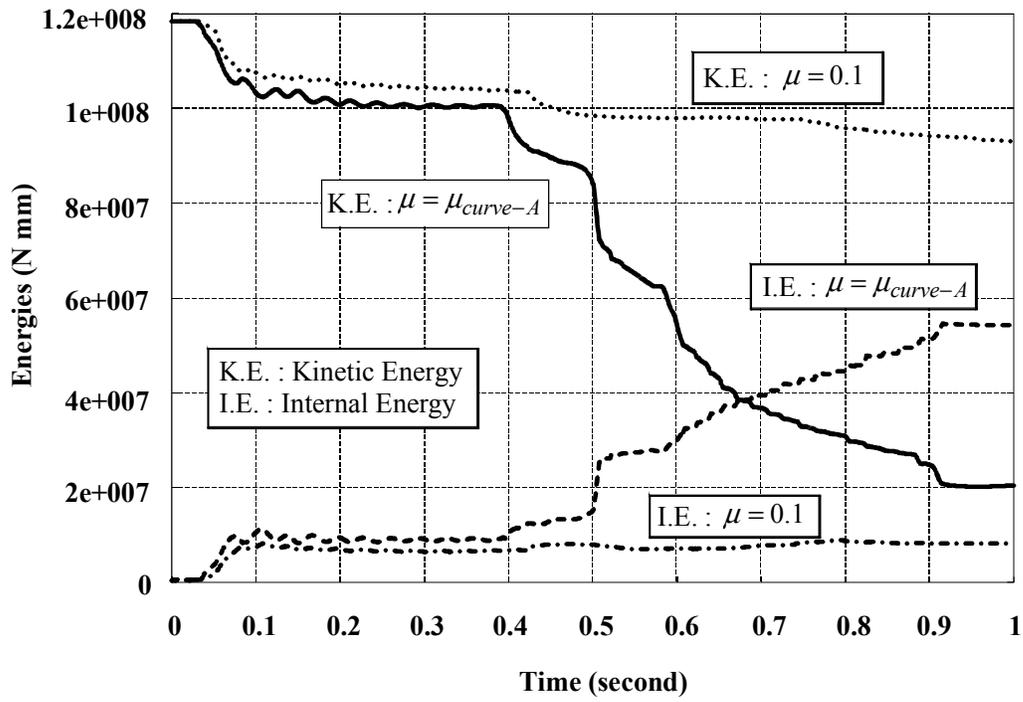
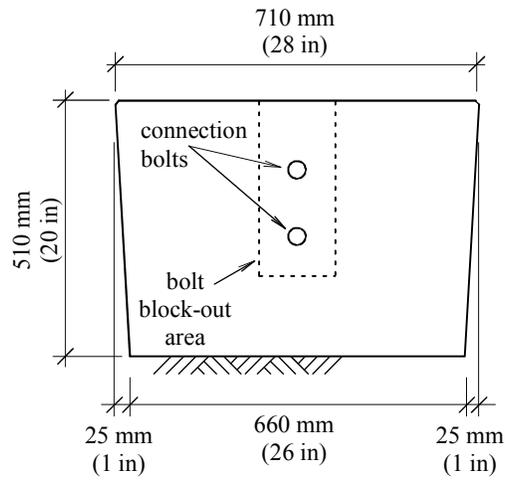
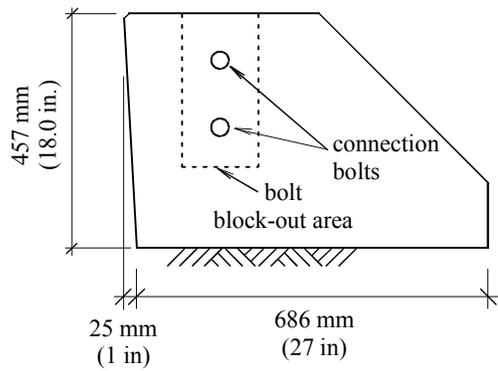


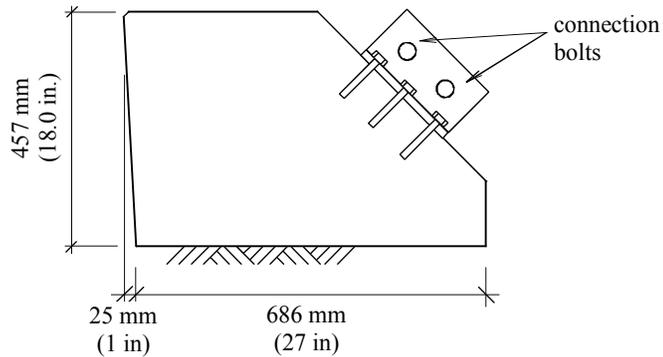
FIGURE 5 Kinetic and internal System energies for curb impact simulations.



a) Design of Guidry and Beason

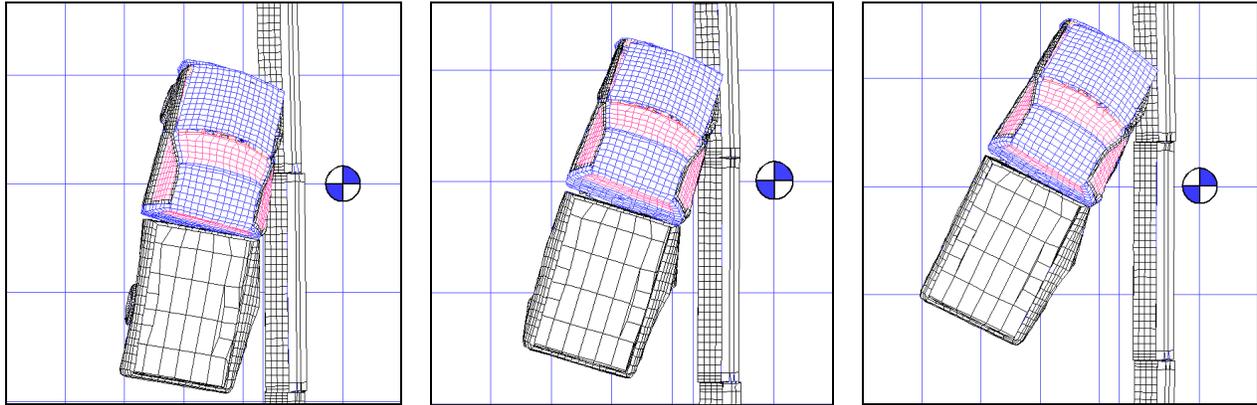


b) Design concept with internal bolts



c) Design concept with external brackets

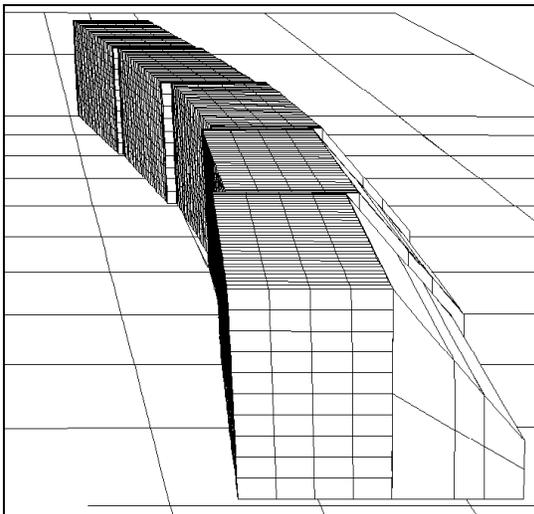
**FIGURE 6 Barrier design concepts considered.**



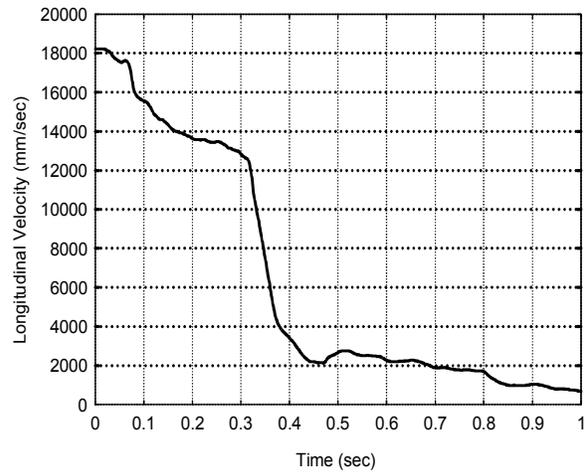
a)  $t = 0.35$  sec.

b)  $t = 0.45$  sec.

c)  $t = 0.65$  sec.

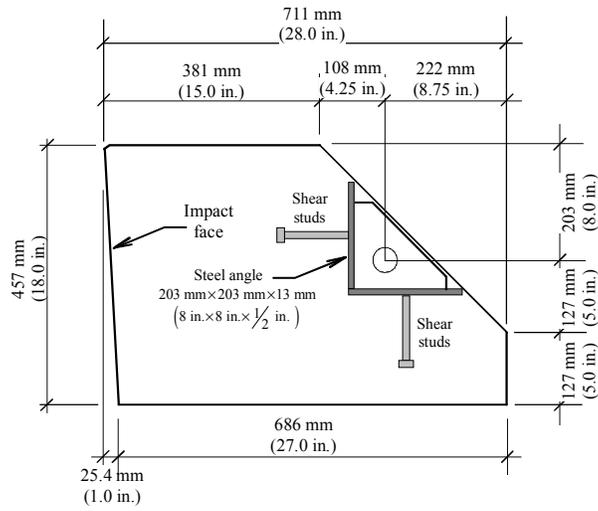


d) Protruding segments at  $t = 0.35$  sec  
(vehicle removed for clarity)

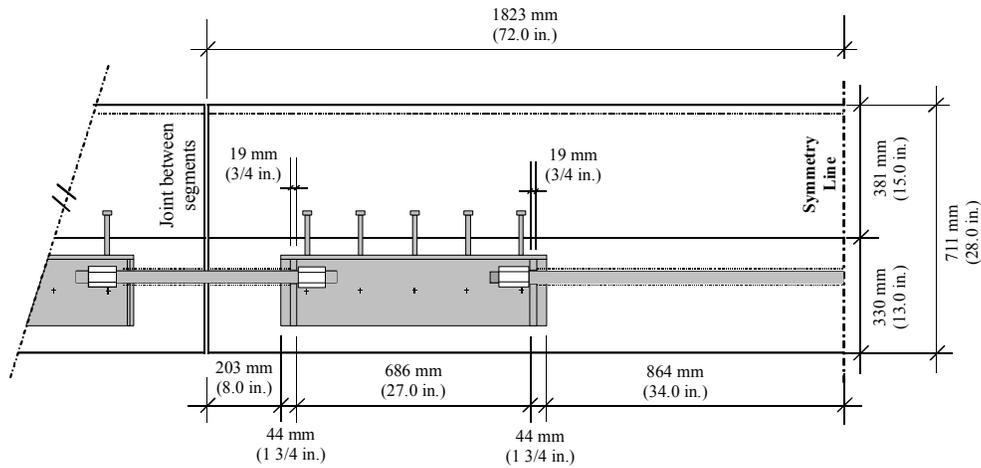


e) Longitudinal velocity of vehicle

**FIGURE 7 Vehicle snagging due to relative sliding motion between barrier segments for connection system shown in Figure 6c.**



a) Cross-sectional view



b) Plan view



c) End view



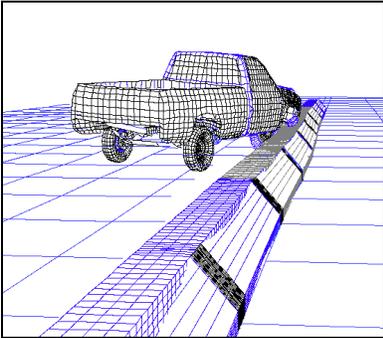
d) Joint between two segments



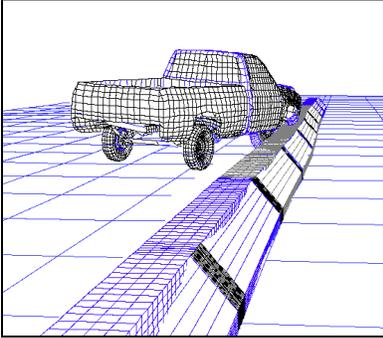
e) Field installation

**FIGURE 8 Final barrier design.**

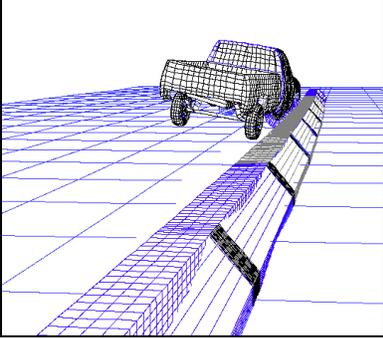
a)  $t = 0.1$  sec



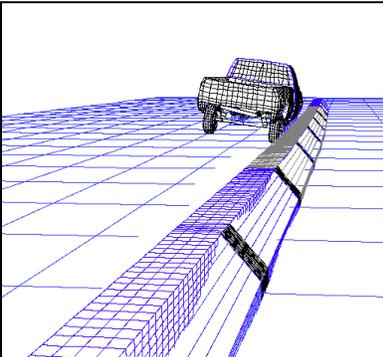
b)  $t = 0.2$  sec



c)  $t = 0.3$  sec



d)  $t = 0.4$  sec



**FIGURE 9 Comparison of crash test results and simulation results for 2000P vehicle.**