TEXAS CRASH-CUSHION TRAILER TO PROTECT HIGHWAY MAINTENANCE VEHICLES

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The Texas crash-cushion trailer, which now has wheels, is a workable and easily used implement for the protection of personnel and equipment, especially during maintenance operations on highways and streets. One crash test to verify the design theory showed that the equations of mechanics predicted results that were very close to the test results. The Texas crash-cushion trailer differs from other crash cushions in that the object supporting the crash cushion is itself movable. This means that fewer steel drums are required but also that the trailer and backup maintenance truck will travel some distance if impacted by an errant vehicle. The distance traveled after impact and the number of steel drums required are determined by equations of momentum and friction.

•THE EFFECTIVENESS of the Texas crash cushion in contributing to highway safety is well documented (5, 7, 8). Previous research and field experience with this device have focused on protecting an errant motorist from a high-speed collision with a rigid obstacle. Common examples are elevated gores and bridge piers in median areas.

The purpose of this research was to use this energy-absorbing device on a trailer to protect slowly moving or stopped maintenance vehicles working on highways. The Texas crash-cushion trailer (TCCT) is to be used to protect highway maintenance equipment and personnel as well as motorists. An important requirement of the TCCT is that it be portable or mobile, easily constructed by highway maintenance personnel, and adaptable to dump trucks and other highway department vehicles.

DESIGN OF TEXAS CRASH-CUSHION TRAILER

The design of the TCCT is based on the law of conservation of momentum and on the dissipation of kinetic energy by plastic deformation of steel drums and through friction. This is somewhat different than the design of fixed crash cushions, which absorb energy by plastic deformation of steel drums only. The critical energy-absorbing condition for the design of the crash-cushion trailer will occur for an impact in which the automobile, crash cushion, and restraining mechanism (usually a truck) are in line at the time of impact (Fig. 1).

For this condition the momentum of the automobile (or striking vehicle) before impact will be equal to the total momentum of the system immediately after impact. Based on plastic impact,

$$\frac{W_c}{g} V_c = \frac{W_c + W_b + W_t}{g} V \tag{1}$$

where

- W_c = total weight of automobile, lb;
- W_b = total weight of portable crash cushion, lb;
- W_t = total weight of truck, lb;
- V_c = velocity of automobile at impact, ft/sec;
- V = velocity of entire system immediately after impact, ft/sec; and
- g = acceleration due to gravity, ft/sec^2.

Solving for the velocity of the entire system after impact yields

$$V = \frac{W_a}{W_a + W_b + W_t} V_c \tag{2}$$

For a truck weighing 9,500 lb, a portable crash cushion weighing 2,000 lb, an automobile weighing 4,500 lb, and an impact speed of 60 mph (88 ft/sec), we have

$$V = \frac{4,500}{4,500 + 2,000 + 9,500} 88 = 24.75 \text{ ft/sec or } 16.88 \text{ mph}$$

The kinetic energy (KE) of the automobile before impact is computed by the formula

$$\begin{aligned} KE &= \frac{MV^2}{2} \\ KE &= \frac{4,500 \times (88)^2}{2 \times 32.2} = 541,000 \text{ ft-lb} \end{aligned} \tag{3}$$

The kinetic energy of the automobile, crash cushion, and truck after impact is

$$KE = \frac{(4,500 + 2,000 + 9,500)(24.75)^2}{2 \times 32.2} = 152,000 \text{ ft-lb}$$

Consequently, 389,000 ft-lb of energy would be absorbed in the impact by plastic deformation of the steel drums in the crash-cushion trailer.

According to White and Hirsch (1), a single 20-gauge tight-head steel drum with 8-in. diameter holes in the top and bottom will absorb 9,000 ft-lb of energy under slowly applied loads. The dynamic factor has been shown to be 1.5 (7). Therefore, each barrel will absorb 1.5 × 9,000 or 13,500 ft-lb of dynamic energy. This would mean that the portable crash cushion would require 28.8 steel drums, but would have 30 barrels to achieve a rectangular configuration.

Figure 2 was developed as a design aid from the foregoing theory. The number of barrels required is plotted against the weight of the resisting truck for impacting vehicles of 2,000, 4,000, and 45,000 lb. The design impact speed is 60 mph in each case. A crash-cushion trailer generally weighs within 15 percent of the values shown. The design vehicle weight range is that recommended by the Federal Highway Administration (2). Figure 2 can serve as a tool for designing portable crash cushions and for comparing the limiting conditions.

After the barrels have deformed plastically and absorbed 389,000 ft-lb of energy, there still remain 152,000 ft-lb of energy because of the entire system moving at 24.75 ft/sec. If all of the wheels of the truck are locked, then the distance required to stop the vehicle is

$$d = \frac{KE \text{ (after impact)}}{W_t \mu} \tag{4}$$

where μ is the coefficient of friction, say, 0.7 for tires to concrete. Then,

$$d = \frac{152,000}{9,500 \times 0.7} = 22.9 \text{ ft}$$

Portable Crash Cushion in Motion

Although the critical design for the energy absorption of the crash cushion itself is for the stationary condition, the critical condition for the distance traveled after impact occurs when the crash cushion and towing vehicle are in motion. Such a condition, for example, occurs during the protection of a paint-stripping machine. In that instance, both the impacting vehicle and the impacted assembly have initial momentum and kinetic energy. Based on plastic impact and conservation of momentum,

$$\frac{W_c}{g} V_c + \frac{W_b + W_t}{g} V_t = \frac{W_c + W_b + W_t}{g} V \tag{5}$$

or

$$V = \frac{W_c V_c + (W_b + W_t)V_t}{W_c + W_b + W_t} \tag{6}$$

If V_c = 60 mph and V_t = 10 mph for a 4,500-lb vehicle and 9,500-lb truck,

$$V = \frac{4,500 \times 60 + (2,000 + 9,500)\ 10}{4,500 + 2,000 + 9,500} = 24.06 \text{ mph or } 35.29 \text{ ft/sec}$$

The kinetic energy before impact is

$$KE = \frac{4,500 \times 88^2}{2 \times 32.2} + \frac{11,500 \times 14.667^2}{2 \times 32.2} = 580,000 \text{ ft-lb}$$

The kinetic energy remaining after impact is

$$KE = \frac{(4,500 + 2,000 + 9,500)\ 35.29^2}{2 \times 32.2} = 309,000 \text{ ft-lb}$$

The change in kinetic energy is 271,000 ft-lb.

Because 271,000 is less than 389,000, the stationary condition governs for plastic energy absorption. However, the stopping distance with all truck wheels locked is

$$d = \frac{KE \text{ (after impact)}}{W_t \mu} = \frac{271,000}{9,500 \times 0.7} = 40.75 \text{ ft}$$

Figure 3 was developed by using the above theory and a series of initial speeds of the truck and portable crash-cushion unit.

Angle Impact

The above calculations consider only the effects of a head-on impact. Angle impacts are possible and, in fact, probable and should be considered. The most probable use for a crash-cusion trailer is to protect maintenance crews on Interstate highways where the usual maneuver is a 1-lane crossover. The impact angle can be determined from the formula (9)

$$\theta = \cos^{-1}\left(1 - \frac{\mu g d}{V^2}\right) \tag{7}$$

The maximum angle from this formula would be about 8 deg for a vehicle speed of 60 mph, a 12-ft lane width, and a coefficient of friction, μ, of 0.7.

Techniques developed by Emori (11) can be used to divide the velocity into longitudinal and tangential components, and energy assumed absorbed along these lines by adding to the analysis the factor

$$KE = \frac{I_o W^2}{2} \tag{8}$$

Figure 1. Crash-cushion trailer before test.

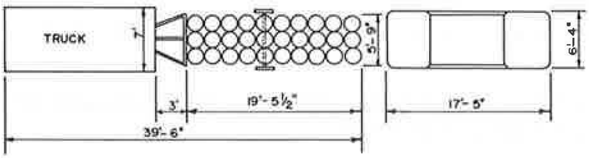


Figure 2. Number of barrels in crash-cushion trailer versus truck weight for several automobile weights.

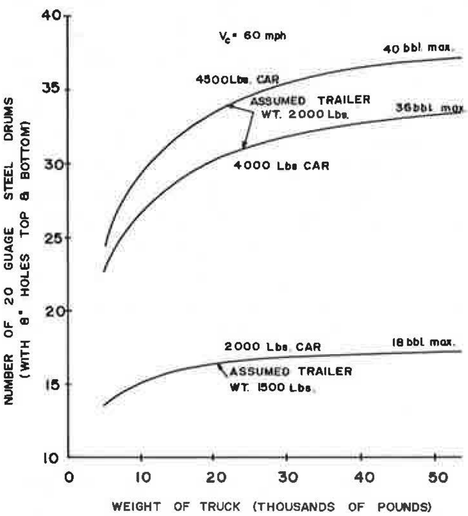
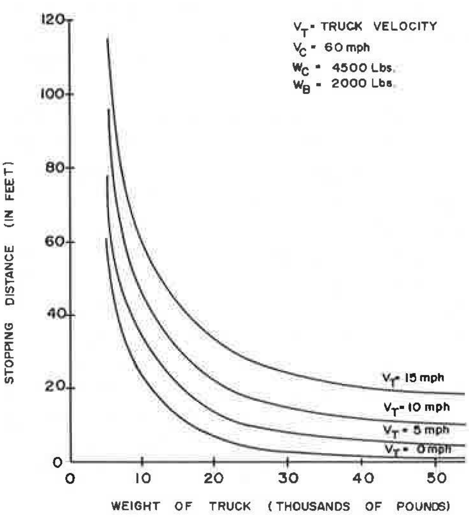


Figure 3. Stopping distance versus truck weight for various initial truck speeds.



where

W = the angular velocity, and

I_o = mass movement of inertia of the truck and crash-cushion combination about the combined mass center.

This energy is then being absorbed by friction of the tires. The maximum angular displacement of the truck is less than 20 deg for these conditions.

Test Crash-Cushion Trailer Design

The test design was based on an impacting vehicle weighing 4,500 lb, a portable crash cushion weighing 2,000 lb, and a truck weighing 9,500 lb. This design is the same as that of the sample calculated above and required 30 steel drums. The crash-cushion trailer was designed to be attached to a standard maintenance dump truck of 5 yd^3 capacity. The truck used was a Dodge D-600 dump truck manufactured in 1963, weighing 9,315 lb. The estimated weight of the crash-cushion trailer was 2,010 lb.

The design of the test portable crash cushion is shown in Figure 4. The drawbar on the truck required some minor modifications to accommodate the 5-point hookup, and the attachments were hand-fitted to the truck. Five points were considered necessary to stabilize the trailer and make it act more nearly as a unit with the towing truck when towing at low speeds or stationary. These additional points were located to produce horizontal and vertical stability of the portable crash cushion. That is, they would prevent the trailer from jackknifing during impact and the impacting vehicle from submarining. Pictures of the completed crash-cushion trailer are shown in Figure 5.

With the exception of the removable arms, all connections were welded. The 4 removable arms were bolted to the face of the portable crash cushion and to the truck. Two technicians pulled the portable crash cushion to the test site and made the complete hookup in less than 5 minutes.

VEHICLE CRASH TEST

The crash-cushion trailer was hooked to the truck (only the trailer hitch was used) and towed around the TTI safety proving grounds at speeds as high as 50 mph for qualitative observation. After this exercise, the steel drums connected to the trailer axle and the row directly behind the axle had slightly deformed tops. This indicated the desirability of moving the axle farther to the rear of the trailer to reduce the cantilever effect of the rear steel drums. The auxilliary connections were made, and the trailer was towed at speeds as high as 25 mph around curves as great as 20 deg. The trailer tracked the truck to a remarkable degree in view of the rigid attachment. There was, however, an abnormal amount of wear to the tires because of side slippage. At lower speeds this wear was insignificant, especially when compared to the life-saving potential.

The primary test was the dynamic or crash test on the stationary truck and crash-cushion unit. For this test, the unit was placed near the north end of the apron of the TTI safety proving grounds. Ample distance to the end of the pavement was allowed for the unit to slide after impact. The arms were bolted in place, the truck was placed in gear (ignition turned off), and the parking brake was set.

The impacting vehicle was a 1964 Chevrolet weighing 4,060 lb. Lateral and longitudinal accelerometers were located on the right and left frame members of the vehicle chassis. The vehicle was towed toward the target by a reverse tow-guidance system (3). The initial impact speed was 63.3 mph and the impact angle was 0 deg (head on) in the center of the rear end of the crash-cushion trailer. Figure 6, a series of pictures from the moving-picture cameras, shows the sequence of events of the test starting at impact. The truck is virtually stationary until the barrels have been crushed to nearly the maximum that occurred during the test. This is important to the use of the system in the field because it shows that most of the energy is absorbed by the crash cushion before the energy wave reaches the truck. In turn, this shows that the instantaneous peak or jolt is at a minimum to anyone seated in the truck. It follows then that there would be little or no damage to the truck. In fact, personnel of TTI and the highway department examined the truck and could find no damage.

Figure 4. Truck crash-cushion trailer assembly.

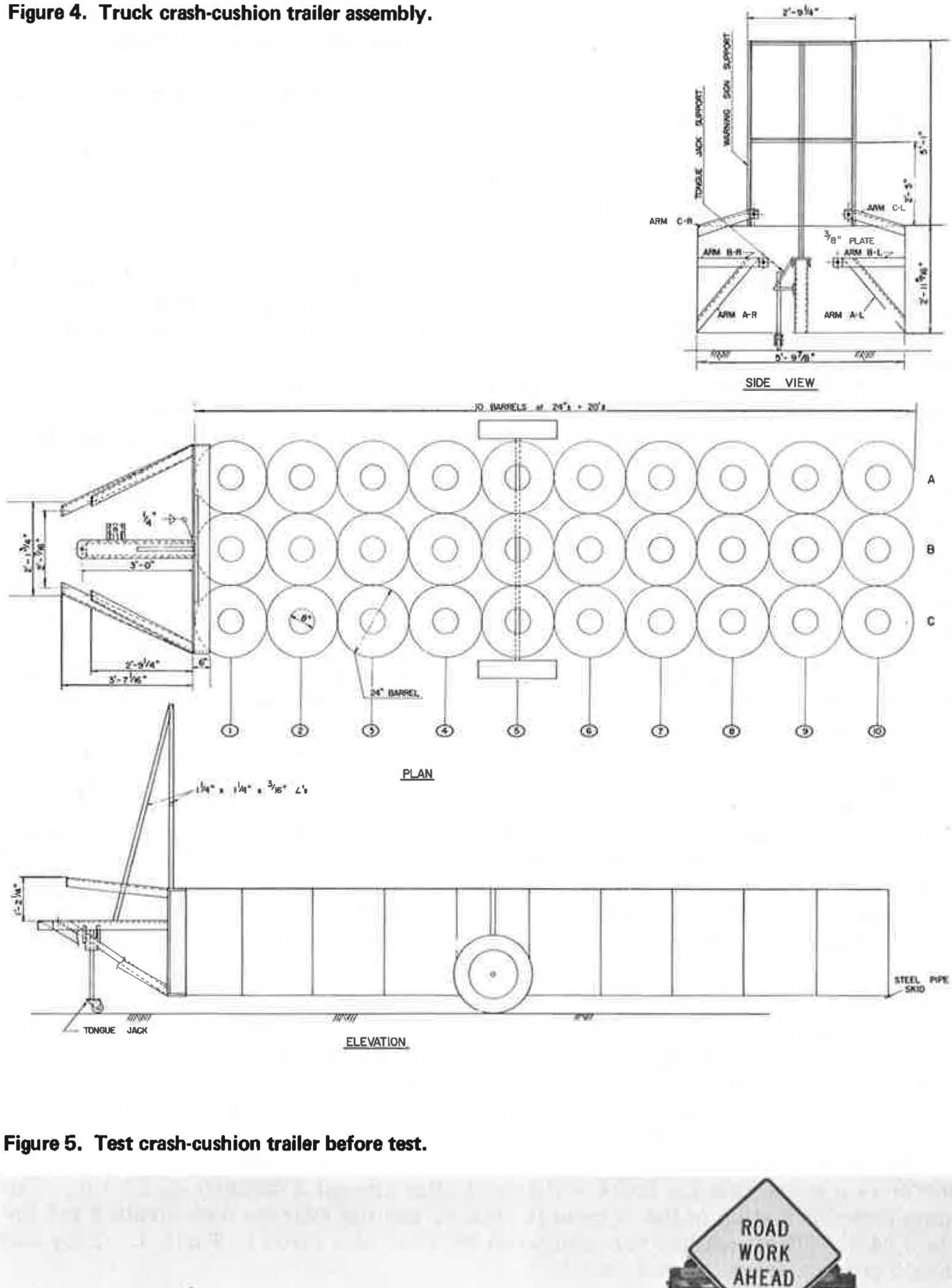


Figure 5. Test crash-cushion trailer before test.

Figure 7 shows the impacting automobile before and after the impact. A minimum amount of damage occurred to the vehicle. In fact, only the 2 inside headlights were broken.

Figure 8 shows the crash cushion after the test and after the vehicle had been pried loose and driven away. Quite obviously, most of the available energy of the steel drums had been used.

DISCUSSION OF RESULTS

Test Data

The data from the tests were collected from 3 different sources: field measurements, electronic instrumentation, and photographic instrumentation. The electronic instrumentation included an Inter-Range Instrumentation Group (IRIG) with 8 available channels (3). Two channels each were used for longitudinal acceleration, lateral acceleration, speed, and spares. The data were transmitted to a central receiver, put on a magnetic tape, and stored. The acceleration data were then filtered through an 80-Hz filter and transferred with the speeds to paper tape in the visicorder. An Impact-O-Graph was used for backup data in the event of a malfunction of the IRIG.

Three high-speed data cameras and 2 documentary cameras were used to record the test and to obtain additional data. A complete description of data reduction techniques using the Vanguard motion analyzer is given in another report (3).

Table 1 gives a summary of the more important test data. There are several comparisons to the theory shown and described in detail below. There is a 1.7-ft difference between the maximum forward motion of the vehicle and the final position of the vehicle. Of this, the truck rebounded approximately 1 ft, which was probably due to the movement of the truck acting against the compression of the engine. Also, the barrier and vehicle rebounded an additional 0.7 ft, indicating that there was some elasticity remaining in the barrier and the front end of the vehicle.

Figure 9 shows the longitudinal and acceleration trace from the visicorder of the IRIG system. The peak g occurs during the period when the first row of steel drums is crushed. The entire crash-cushion trailer started moving forward at 211 msec, the point where the vehicle, cushion, and truck move as a unit. The vehicle deceleration zeros out the first time at 366 msec, about the time the truck achieves maximum speed. After that time, the acceleration trace (not shown) ranges from less than 1 to 0 negative g. Hence, we see that most of the energy is absorbed in plastic deformation of the barrels and elastic and plastic deformation of the vehicle, as the analysis predicted.

Correlation of Theory and Test Data

The theory based on conservation of momentum and kinetic energy described earlier produced results that are in excellent agreement with the test data when the test values are used. That is, from the conservation of momentum and substituting values given in Table 1 into Eq. 1, we compute V at the end of the crash = 24.5 ft/sec and the change in kinetic energy = 399,640 ft-lb.

The calculations indicated that the number of barrels used is 29.60 with a cushion distance of 14.8 ft and that the truck will travel after impact a distance of 22.0 ft. The maximum forward motion of the vehicle is 36.8 ft, and the average deceleration for the event is 3.64 g. (These values are compared with the data given in Table 1. They vary less than 5 percent from the test data.)

SUMMARY AND CONCLUSIONS

The crash-cushion trailer is a workable, easily used solution for protection against certain classes of accidents on highways and streets. Those accidents are the ones most likely to occur during maintenance operations where head-on or near head-on collisions are likely. Simple equations of mechanics are extremely accurate for the design and use of the crash-cushion trailer. Further, the curve shown in Figure 2 can be used for the design of specific crash-cushion trailers, and the curves shown in Figure 3 will assist in the safe location of the crash cushion.

Figure 6. Crash-cushion trailer during test.

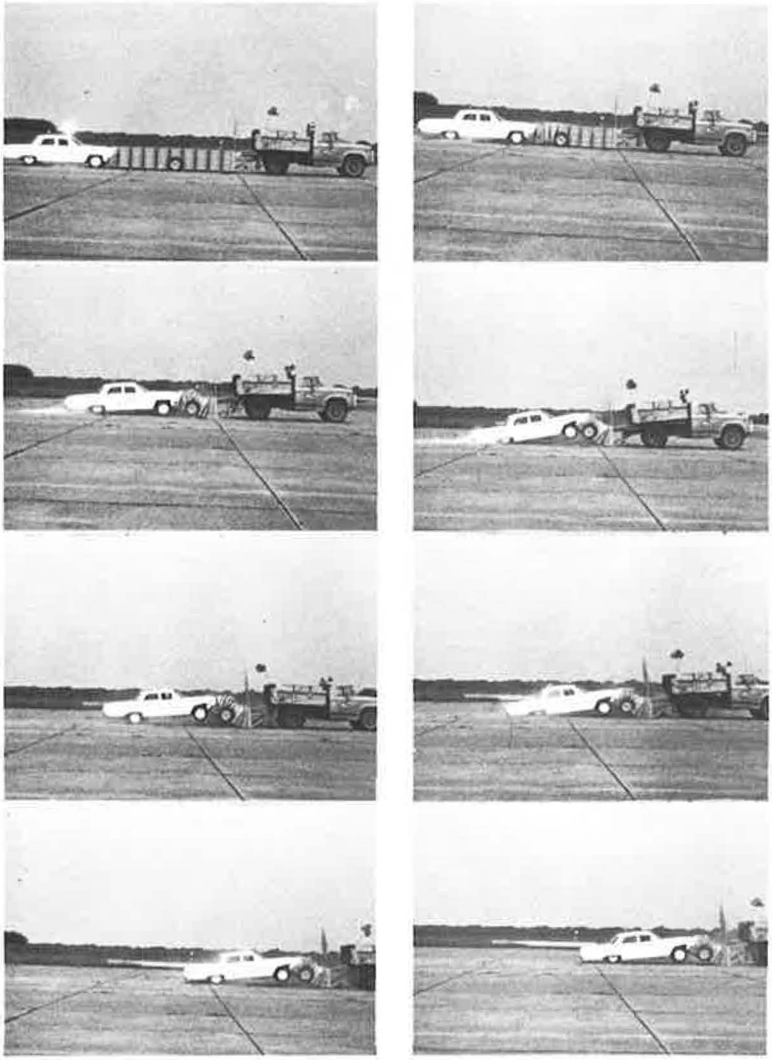


Figure 7. Test vehicle before and after test.

Figure 8. Crash-cushion trailer after test.

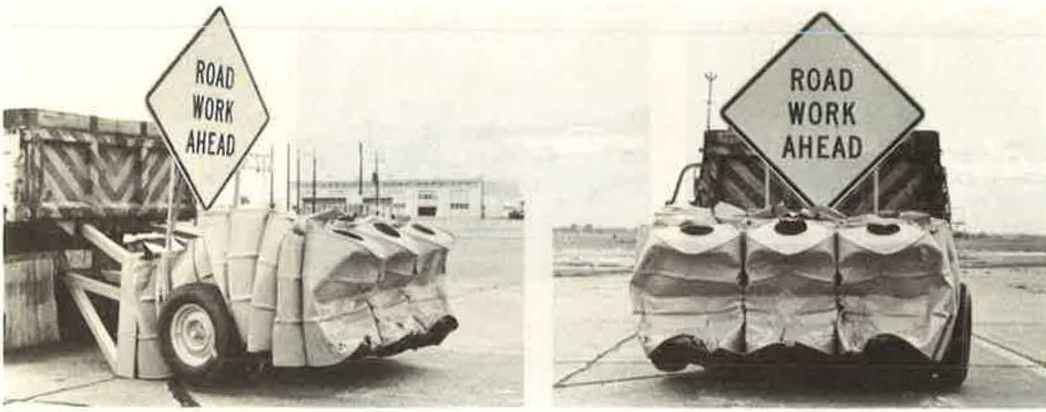
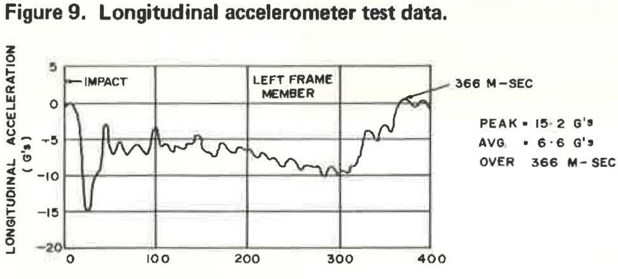


Table 1. Test data.

Item	Test Data	Computation
Initial speed		
ft/sec	92.8	
mph	63.3	
Maximum forward motion of vehicle, ft	36.4	36.8
Time to end of forward motion, sec	1.856	
Maximum forward motion of truck, ft	21.0	22.0
Final vehicle forward motion, ft	34.7	
Final truck forward motion, ft	20.0	
Final vehicle deformation, ft	0.2	
Final cushion deformation, ft	14.5	14.8
Average deceleration ($V^2/2gS$), g	3.67	3.64
Maximum longitudinal acceleration, g	-15.2	
Average longitudinal acceleration to end of significant peak, g	-6.6	
$\Delta V/\Delta t$ (to 0.366 sec), g		

Figure 9. Longitudinal accelerometer test data.



The structural connections between the crash-cushion trailer and the truck or other stabilizing vehicle should be adequate or even over-designed. The backup plate, also, should be stiff enough so that as uniform a restraining force as possible will be applied to the barrels during an accident.

The crash-cushion trailer can be used in 3 basic maintenance or construction operations. The first is in detour situations where a missed detour might result in injury to the vehicle occupants or to workers. In this situation, the crash-cushion trailer with its towing vehicle could be anchored on a temporary basis for the duration of the hazard and then moved to a new location as maintenance or construction progressed.

Another possible use is as a temporary stationary crash cushion to protect workers on travel lanes or on shoulders as they performed routine maintenance such as mowing, guardrail repair, chug-hole repair, trash collection. A driver could stay in the truck and move along with the task.

A third type of operation is a moving operation in which the progress of the operation proceeds at a much slower speed then that of the traffic. Such operations include striping of traffic lanes and placing traffic buttons.

A crash-cushion trailer for municipal streets could be much smaller than one used on high-speed expressways. The distance a crash-cushion trailer is placed from or follows an obstacle or worker to be protected should be governed by calculations for a safe distance or from curves shown in Figure 3. An adequate margin of safety should be used for the final distance.

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