A PORTABLE ENERGY ABSORBING SYSTEM

FOR HIGHWAY SERVICE VEHICLES

FINAL REPORT

John F. Carney III, Associate Professor

JHRAC Project 73-4 JHR 74-83

September 1974

This research was sponsored by the Joint Highway Research Council of the University of Connecticut and the Connecticut Department of Transportation, and was carried out in the Civil Engineering Department of the University of Connecticut.

TABLE OF CONTENTS

V_{tx}	x-component of truck velocity before impact
$V_{tx'}$	x-component of truck velocity after impact
V_{ty}	y-component of truck velocity before impact
$V_{ty'}$	y-component of truck velocity after impact
V^*	equivalent pre-impact automobile velocity
W	depth of ring
W_c	weight of automobile
W_t	weight of truck
α	virtual angle change
θ	angle of attack
μ	coefficient of friction
σ_o	static yield stress

A PORTABLE ENERGY ABSORBING SYSTEM
FOR HIGHWAY SERVICE VEHICLES

Abstract

This investigation is concerned with the design of a portable energy absorbing system to be attached to the rear of highway service vehicles to furnish protection of personnel and equipment when performing maintenance operations. A comparative study of the various energy absorbing mechanisms is conducted, leading to the adoption of thick-walled steel rings as the prime energy absorbing devices in the design. The resulting ring type energy absorbing apparatus appears to offer effective protection to people and equipment involved in the collision for a wide range of impacting velocities.

Summary of Related Work

The design of traffic safety barriers to reduce damage to people and equipment often involves the employment of impact attenuation devices. In most cases, these energy absorbing systems have been used to reduce the severity of vehicular collisions with immovable obstacles such as bridge piers, light poles, guardrails, sign posts, and concrete walls and abutments.

A vehicle crash attenuation system made up of 55-gallon drums was developed and successfully tested in 1968 [1]*. Patterns were cut into the lids of the barrels to reduce the crushing strength of the system. More tests were performed on the system in 1969 [2], and a report on the in-service experience of these "modular crash cushions" in Houston, Texas was presented [3]. In the one year period since their installation in October 1968, there were thirteen accidents involving impact with the device. No serious injuries or fatalities were reported. Further research to improve the basic modular crash cushion design was conducted in 1970 and 1971 [4,5].

The successful implementation of the 55 gallon steel drum modular crash cushion prompted a study of the feasibility of using other energy absorbing units. Corrugated steel pipe was statically crush tested and found to have favorable characteristics. The availability of corrugated steel pipe for a wide range of thickness and diameter dimensions led to the development of a polymodular design [6] in which the physical characteristics of the crash cushion could be varied on a row to row basis.

A hi-dro cushion cell barrier impact attenuator system has also been successfully tested and installed. This system uses an array of water filled

*Numbers in square brackets denote references which are collected at the end of the report.

plastic cells. Upon impact, the liquid is ejected through orifices in the top of the cells at a controlled rate. Analytical and experimental studies have been conducted by Warner and Free [7] and at the Texas Transportation Institute [8].

Sand-filled frangible plastic barrels have been used as energy absorbing barriers. They are usually composed of an array of 15-17 barrels 36 in. in diameter and 30 to 36 in. high. The sand inertia barrier concept leads to a low cost, versatile installation [9] and has been successfully employed in Connecticut, Idaho, Minnesota and New York.

The TOR-SHOK energy absorbing barrier is a U-shaped tubular guardrail attenuator that absorbs energy by means of the motion of supporting telescopic tubes. Upon impact, the impact forces are transmitted axially to the TOR-SHOK arms which contain many stainless steel torus elements that are squeezed between two cylindrical tubes. This system and a related one called ROTO-SHOK have been tested [10,11] and proven marginally effective but these units are expensive to install and maintain.

The Dragnet vehicle arresting system is composed of a steel entrapping net positioned across the roadway. The steel cables are attached at each end to Metal Bender energy absorbing devices. The "Dragnet" system was successfully tested [12] and deemed of practical use at such locations as dead ends of roads, ferry landings and highway medians at bridge overpasses.

A lightweight cellular concrete crash cushion constructed of easily frangible vermiculite concrete with vertical voids has been studied [13,14]. The vertical voids contribute to the controlled crushing characteristics of the system. Problems associated with this system include the high cost of repairs and the poor weathering characteristics of the attenuator. An analysis of the field accident data on the various impact attenuators discussed to this point was conducted in 1973 [15].

Other Devices

The fragmenting tube concept was developed during the early stages of the space program at NASA by John R. McGehee [16,17]. The main components of the fragmenting tube energy absorbing system are the thick walled aluminum tube and a flaring die. Energy is absorbed by forcing the thick walled tube over the flared die, resulting in a shedding of the tube into small segments. The fragmenting tube system was originally developed for use in planned lunar landing modules. However, its potential use in energy absorbing guardrail systems was noted, and an analytical and experimental investigation of the feasibility of such a system was undertaken by the Southwest Research Institute for the Federal Highway Administration. The final report [18] concluded that the fragmenting tube type energy absorbing system did function properly and recommended that a full scale installation of a fragmenting tube energy absorbing bridge rail be made at a bridge overpass. An installation of such a system is planned in Connecticut.

In 1972, the use of thick walled steel rings as energy-absorbing units was investigated [19] by Perrone. The obvious application, as with the fragmenting tube concept, was in energy absorbing guardrail systems. In [19], a quantitative study of the energy dissipated by the rings when loaded in-plane to complete collapse was presented.

Following Perrone's work, which included a feasibility study of the use of thick walled steel rings in bridge rail systems, the Southwest Research Institute was contracted to run a series of full scale tests on such a system

[20,21,22] and the results to date have been encouraging.

This review of related work has so far dealt with stationary energy absorbing systems. In many movable highway maintenance operations, however, personnel and equipment are inadequately protected from collision by an errant vehicle. To provide this needed protection, two types of portable units have been designed. The first unit is composed of hi-dro cell components [7,8] attached to the rear of the follower truck in the maintenance operation. The second design employs "modular crash cushion" elements (55-gallon drums) [23]. This system is presently being used in the states of Washington and Texas and consists of a trailer carrying 30 crushable barrels (10 rows of 3 barrels).

A Comparison of Portable Energy Absorbing Systems

The two operational portable units (hi-dro cell and modular crash cushion) are a study in contrasts. They both dissipate energy upon impact, but the portable hi-dro cell unit now being used is approximately 3 feet long while the modular crash cushion unit is 19-1/2 feet in length.

The hi-dro cell system consists of 5 rows of 13 polyvinyl chloride plastic cells enveloped in a corset like membrane. The entire unit rests on a metal platform which is attached to the rear of the truck. Each cell contains approximately 3-1/2 gallons of a water-calcium chloride solution.

The modular crash cushion portable system is composed of thirty steel drums (10 rows with 3 barrels per row) constructed of 20 gage steel resting on a trailer. The trailer is attached at five points to the truck to provide horizontal and vertical stability during impact.

The hi-dro cell unit is portable and relatively easy to install on the rear of a highway truck. Its usefulness as an energy absorbing system is of primary concern, however, and the present design offers satisfactory protection only for relatively low speeds (less than 30 mph). For higher speeds, the present design cannot simultaneously satisfy energy absorption and minimum stopping distance (deceleration) requirements.

The modular crash cushion possesses the required energy absorption capability for speeds of up to 60 mph. Furthermore, the 19-1/2 foot length of the barrel system, coupled with the energy absorbing characteristics of the individual barrels, results in acceptable deceleration levels for even 60 mph impacts. The modular crash cushion clearly performs its energy absorbing function admirably.

Day to day usage problems, however, have developed with the system. Tires wear much faster than expected when compared with other trailers. This problem is caused by the rigid connection required for stability between the trailer and the towing vehicle. Major difficulties associated with weld fatigue have developed. Three of the four units in operation (1973) have experienced this problem which is caused in part by the necessity of welding the light gage steel drum rims to #3 rod spacers, a connection subject to fatigue failure.

Problems, then, are clearly associated with both systems. A portable energy absorbing system could be constructed of aluminum frangible tubes, operating on the same principle employed in its guardrail application. The design becomes complicated, however, because of the significantly higher energy dissipation requirements of the portable system over the guardrail one due to the probability of normal impacts (see Fig. 1 with $\theta = 0$).

An energy absorbing system composed of thick walled steel rings is another possibility. The rings can be designed to behave dynamically in a manner similar to the frangible tube system but with several advantages:

1) The ring system exhibits less rebound than the frangible tube system, cutting down on the slingshot effect on the colliding vehicle and decreasing decelerations.
2) The ring system is more economical than the frangible tube system.
3) The rings can be reused after minor impacts by merely jacking them back to their original configuration.
4) The "shrapnel" problem associated with the fragmenting process in the frangible tube is eliminated.
5) The guide system and associated support devices needed in the frangible tube system are not necessary.

Because of these points and others to be discussed later, it was decided to concentrate on the design of a portable energy absorbing system made up of thick walled steel rings. The goal is to develop a system possessing the favorable characteristics of the modular crash cushion while avoiding the day to day usage problems associated with that design.

Formulation of the Problem

The basic problem involves the collision of two bodies: a passenger car travelling at a high rate of speed and a highway service vehicle moving at, say, 10 mph. A probable collision situation is depicted in Fig. 1. The car impacts with the truck with an angle of attack, θ. Given the values of the velocity vectors of the car and truck, before impact, the appropriate equations of dynamics may be applied to determine the velocities of the two vehicles after the collision and the amount of energy dissipated during the impact. In this development, any rotations about the mass centers of the vehicles will be neglected.

Applying the principle of the conservation of momentum in the x-direction yields

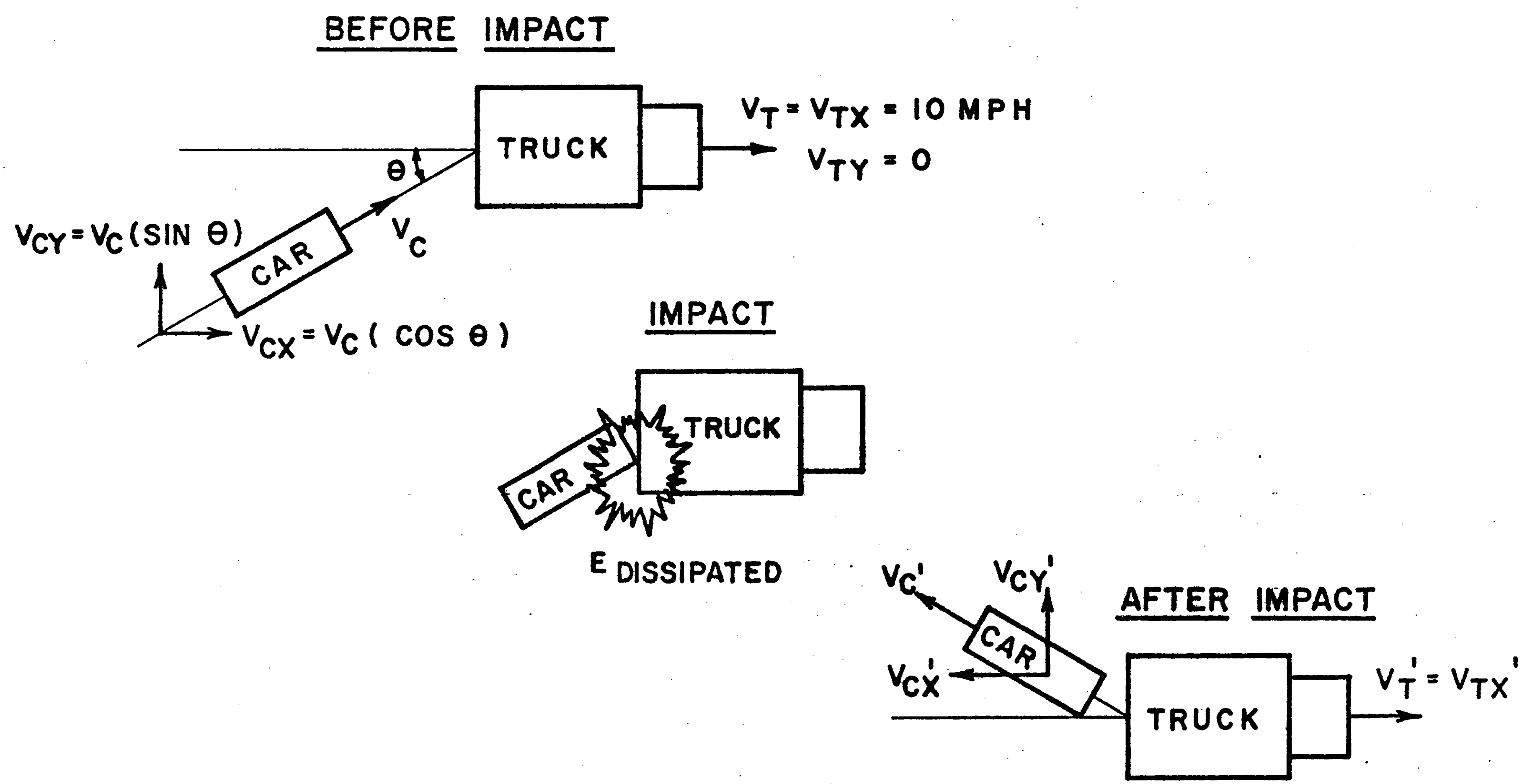

FIG. 1 Case A - Probable Collision Situation

$$\frac{W_c}{g} V_{cx} + \frac{W_t}{g} V_{tx} = \frac{W_c}{g} V_{cx'} + \frac{W_t}{g} V_{tx'} \tag{1}$$

where

W_c = weight of car

W_t = weight of truck

g = acceleration due to gravity

V_{cx} = x-component of velocity of car before impact

$V_{cx'}$ = x-component of velocity of car after impact

V_{tx} = x-component of velocity of truck before impact

$V_{tx'}$ = x-component of velocity of truck after impact

The principle of impulse and momentum will now be applied to each vehicle separately in the y-direction. Assuming that the y-component of the impulsive force is negligible compared to its x-component, it follows that the momentum of each vehicle in the y-direction is conserved. Since the masses of the car and truck are unchanged,

$$V_{cy'} = V_{cy} \tag{2}$$

$$V_{ty'} = V_{ty} \tag{3}$$

where

V_{cy} = y-component of velocity of car before impact

$V_{cy'}$ = y-component of velocity of car after impact

V_{ty} = y-component of velocity of truck before impact

$V_{ty'}$ = y-component of velocity of truck after impact

Assuming that the coefficient of restitution is known, the difference in velocities of the truck and car after impact can be obtained from the equation

$$V_{tx'} - V_{cx'} = e\,(V_{cx} - V_{tx}) \qquad (4)$$

where

e = coefficient of restitution ($0 \leq e \leq 1$)

Equations 1 and 4 may now be solved simultaneously for the x-components of both vehicles after impact. The resultant velocities after impact may then be expressed in the form

$$V_{c'} = [(V_{cx'})^2 + (V_{cy'})^2]^{1/2} \qquad (5)$$

$$V_{t'} = [(V_{tx'})^2 + (V_{ty'})^2]^{1/2} \qquad (6)$$

where

$V_{c'}$ = velocity of car after impact

$V_{t'}$ = velocity of truck after impact

The energy dissipated during the collision is equal to the energy in the system before impact minus the energy in the system after impact. The kinetic energy before impact is

$$E_{\text{initial}} = \frac{W_c V_c^2}{2g} + \frac{W_t V_t^2}{2g} \qquad (7)$$

where

V_c = velocity of car before impact

V_t = velocity of truck before impact

The kinetic energy after impact is

$$E_{\text{final}} = \frac{W_c (V_{c'})^2}{2g} + \frac{W_t (V_{t'})^2}{2g} \qquad (8)$$

and the energy dissipated is

$$E_{\text{dissipated}} = E_{\text{initial}} - E_{\text{final}} \qquad (9)$$

The portable energy absorbing system must be designed to handle the energy to be dissipated as given by Eq. 9.

An important design tool employed in this investigation is the Barrier VII computer program developed for the Federal Highway Administration by G. H. Powell [24,25]. The Barrier VII program analyzes the behavior of an automobile or other vehicle striking a deformable protective barrier. The program is extremely general in scope, incorporating dynamic effects, large displacements, and the inelastic, nonlinear behavior of the system's components. Because of the program's very general nature it is somewhat complicated to use, and no attempt will be made here to describe in detail its makeup. The interested reader is referred to references 24 and 25. One important restriction of the program is, however, that the barrier itself remain stationary. Since the barrier in the proposed system is to be attached to a service vehicle moving at approximately 10 mph, the problem at hand must be modified before using the Barrier VII package. The modified problem to be inputed into the Barrier VII program will be characterized by the requirement that the total energies absorbed by the two systems (actual and modified) be equal.

In the modified system, the truck will be assumed to remain stationary before and after impact in order to simulate a fixed barrier. The proposed collision situation is shown in Fig. 2. The constraint imposed on the impact shown in Fig. 2 is that the energy absorbed is equal to that of the real system depicted by Fig. 1. The unknown to be computed, therefore, is the revised velocity of the car before impact. This is the velocity which will be used as input into Barrier VII.

As in the real case, the vertical components of velocities of the car and truck remain unchanged, so that

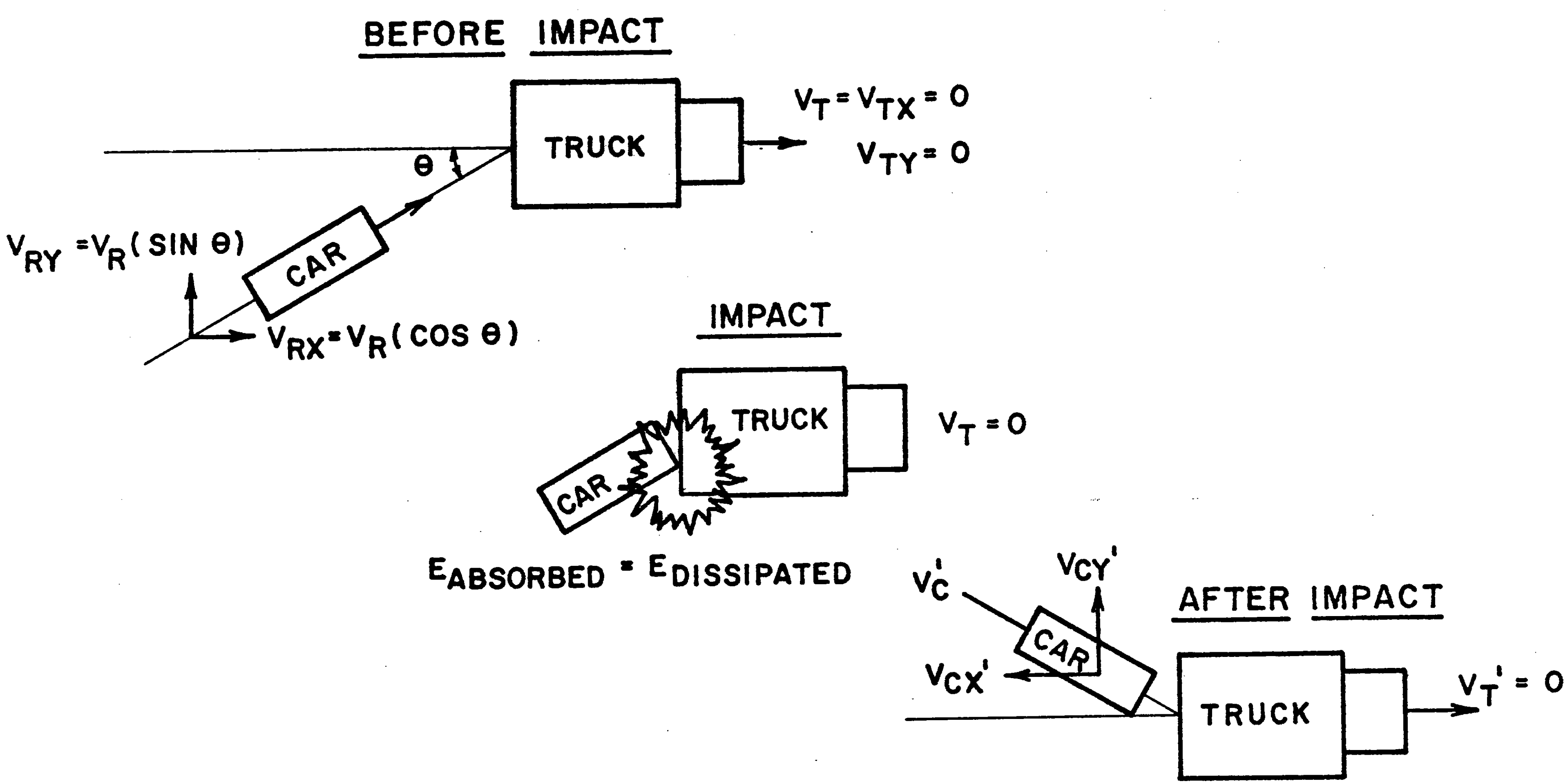

FIG. 2 Case B - Theoretical Collision Situation

$$V_{cy'} = V_{Ry} \tag{10}$$

$$V_{ty'} = V_{ty} = 0 \tag{11}$$

where

V_{Ry} = y-component of revised velocity of car before impact

From the definition of the coefficient of restitution, it follows that

$$V_{cx'} = - e V_{Rx} \tag{12}$$

where

V_{Rx} = x-component of revised velocity of car before impact

The velocity of the car after impact is therefore

$$V_{c'} = [(e V_{Rx})^2 + (V_{cy'})^2]^{1/2} \tag{13}$$

The energy absorbed in the revised system is

$$E_{absorbed} = \frac{W_c (V_R)^2}{2g} - \frac{W_c (V_{c'})^2}{2g} \tag{14}$$

Substituting Eqs. 10 and 13 into Eq. 14 leads to

$$V_R = \left[\frac{2g \, E_{absorbed}}{W_c \, (1 - \sin^2\theta - e^2 \cos^2\theta)} \right]^{1/2} \tag{15}$$

where the energy absorbed in the modified system is equal to the energy dissipated in the real system, which is given by Eq. 9.

The revised velocity is a function of the energy absorbed, the weight of the automobile, the coefficient of restitution, and the angle of attack. A computer program developed to calculate V_R for a wide range of these parameters shows, however, that the revised velocity is quite insensitive to small changes in small values of the coefficient of restitution. Since in this problem e is probably quite close to zero, the term in Eq. 15 containing e^2 can for practical purposes be ignored. Furthermore, a change in the angle of attack from 0° to 15° causes changes of less than 3% in the revised velocities, and the amount of energy absorbed will be a maximum when the coefficient of restitution and the attack angle are zero. In view of these considerations Eq. 15 can be simplified and written as

$$V_R \cong \left[\frac{2g\, E_{absorbed}}{W_c}\right]^{1/2} \tag{16}$$

when e and θ are small.

A plot of actual car velocity versus revised car velocity is presented in Fig. 3 for a normal impact (θ = 0°) under plastic conditions (e = 0). Two cases are considered. One case depicted is that of an equal weight collision in which both the automobile and the truck weigh 5000 pounds. In the second case, the impact is between a 4000 pound automobile and a 14,000# truck. In both cases the speed of the service vehicle (truck) before impact is taken as 10 mph in the same direction as the auto. Note that the revised car velocity before impact for a given actual car velocity is higher for the 4000-14,000 pound case than in the equal weight collision. This is caused because more energy is dissipated in the 4000-14,000 case than in the equal weight case, as shown in Fig. 4.

A specific example will serve to illustrate the different characters of the two collisions. Assume that, in both cases, the automobile is traveling at 55 mph when it impacts the truck which is moving at 10 mph. In the 5000-5000 pound case, the service vehicle and car attain a post-impact velocity of 32.5 mph. The collision therefore causes a decrease of 22.5 mph in the automobile speed accompanied by an increase in velocity of 22.5 mph in the truck. In this case, a significant amount of energy is transferred to the service

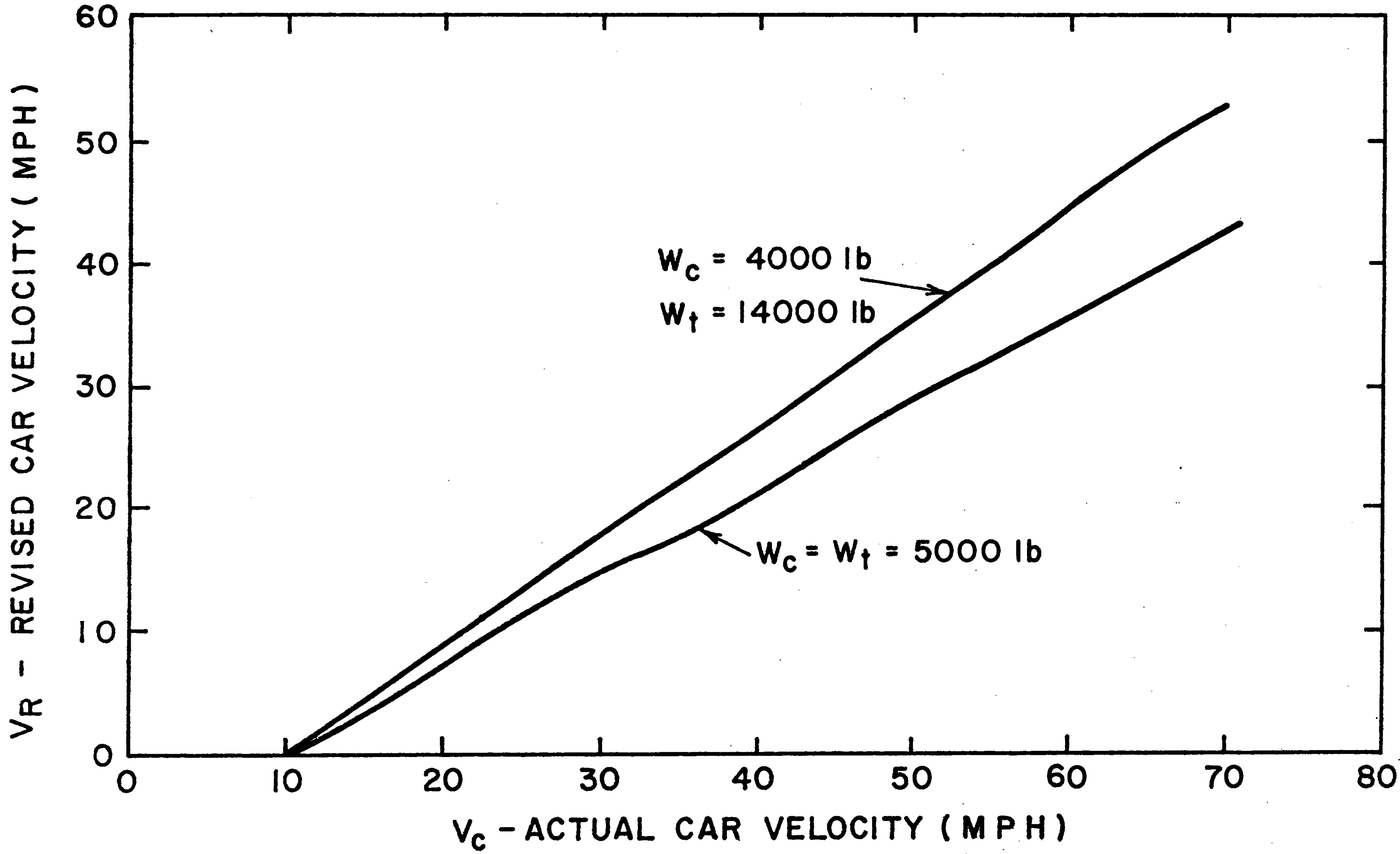

FIG. 3 Actual Car Velocity vs. Revised Car Velocity

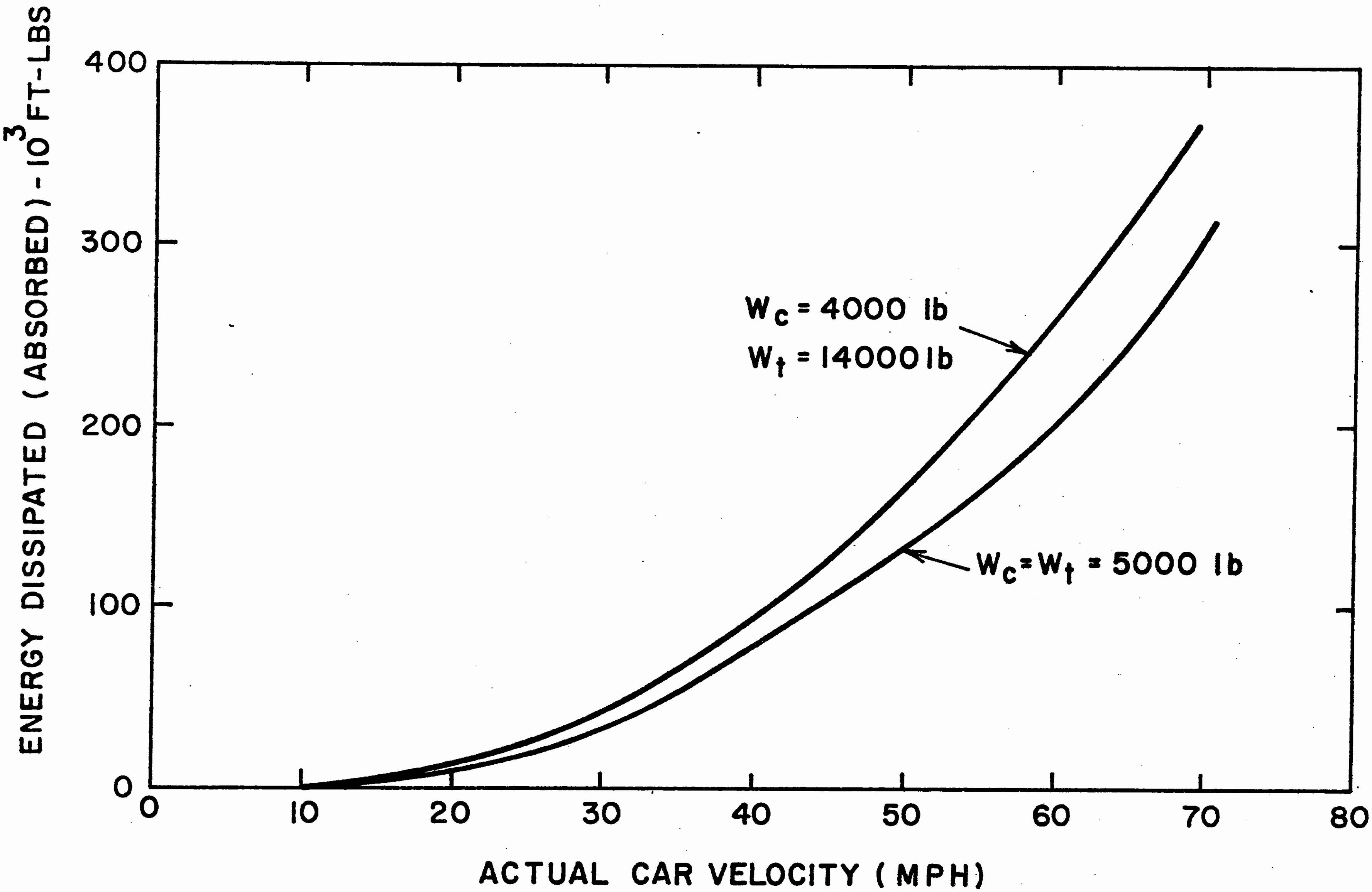

FIG. 4 Actual Car Velocity vs. Energy Dissipated (Absorbed)

vehicle with associated undesirable accelerations.

A more satisfactory situation occurs in the 4000-14,000 pound case. The automobile will again be assumed to be traveling at 55 mph when it impacts the 14,000 service vehicle moving at 10 mph. The post-impact velocity of the system in this case is 20 mph. This collision therefore causes a decrease of 35 mph in the automobile accompanied by an increase in velocity of only 10 mph in the truck. More energy will be dissipated during the impact in this system as compared with the equal weight case, resulting in lower levels of accelerations being felt by personnel occupying the service vehicle. Because of this fact, it is recommended that the 14,000 service vehicle be used with the portable energy absorbing system, and the additional energy absorption potential needed will be built into the system.

Energy Absorption Characteristics of Thick-Walled Rings

Research into the feasibility of employing thick-walled rings as energy absorbing units when loaded to complete collapse in the plane of the ring has been conducted by Perrone [19]. This excellent piece of work involves both experimental and analytical studies relating the dissipated energy in the ring to its geometry and material characteristics.

Rings with 18 inch diameters and 1/2 inch thicknesses made of A53A, A53B, and X52 steel were tested. Uniaxial tensile specimen coupons of these steels exhibited the stress-strain characteristics depicted in Fig. 5 [19]. It is clear from Fig. 5 that the constitutive properties of the three steels are almost identical. Next, load-deformation tests were conducted on the three rings and these results are shown in Fig. 6.

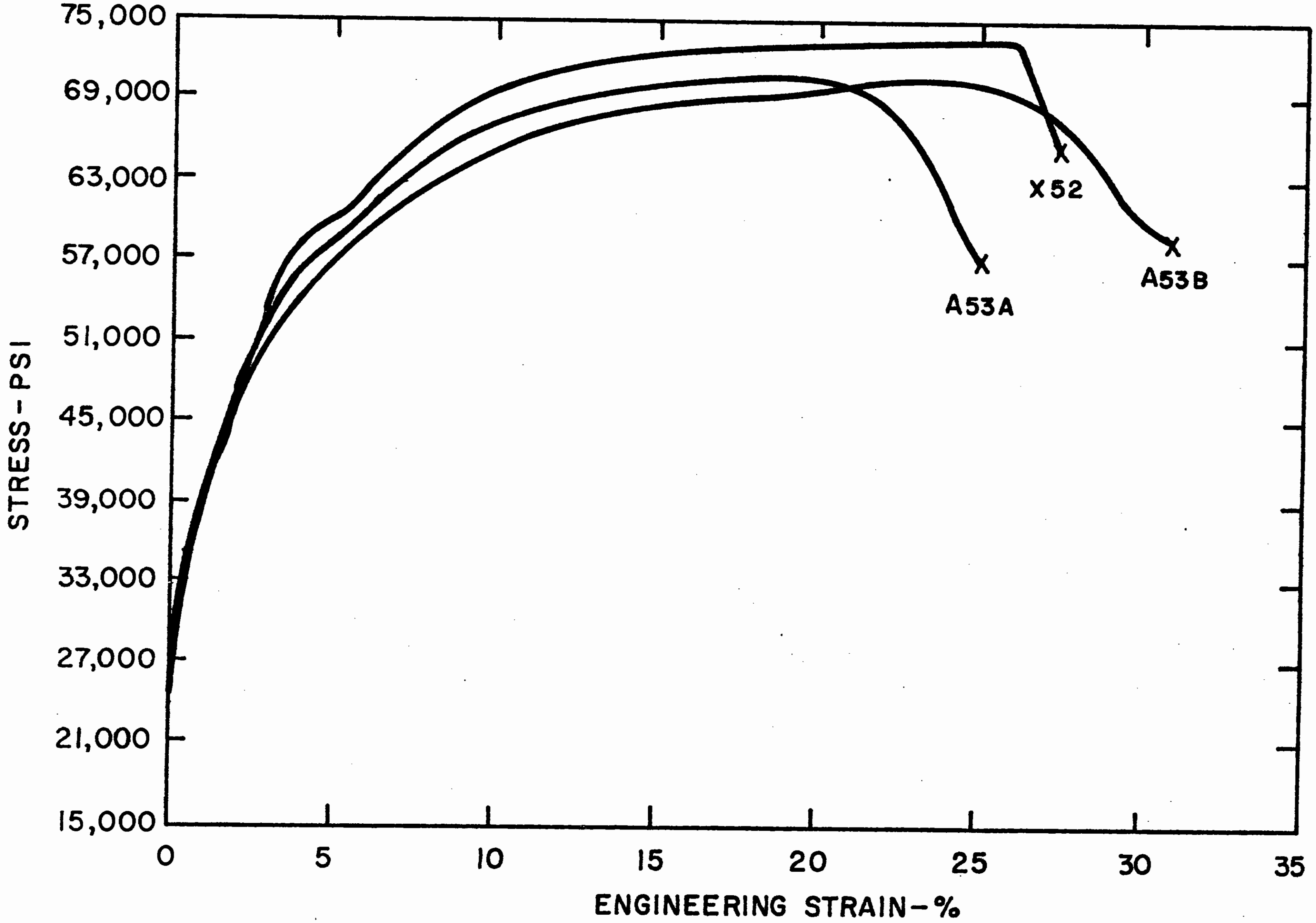

FIG. 5 - STRESS - STRAIN CURVES FOR A53A, A53B AND X 52 STEELS

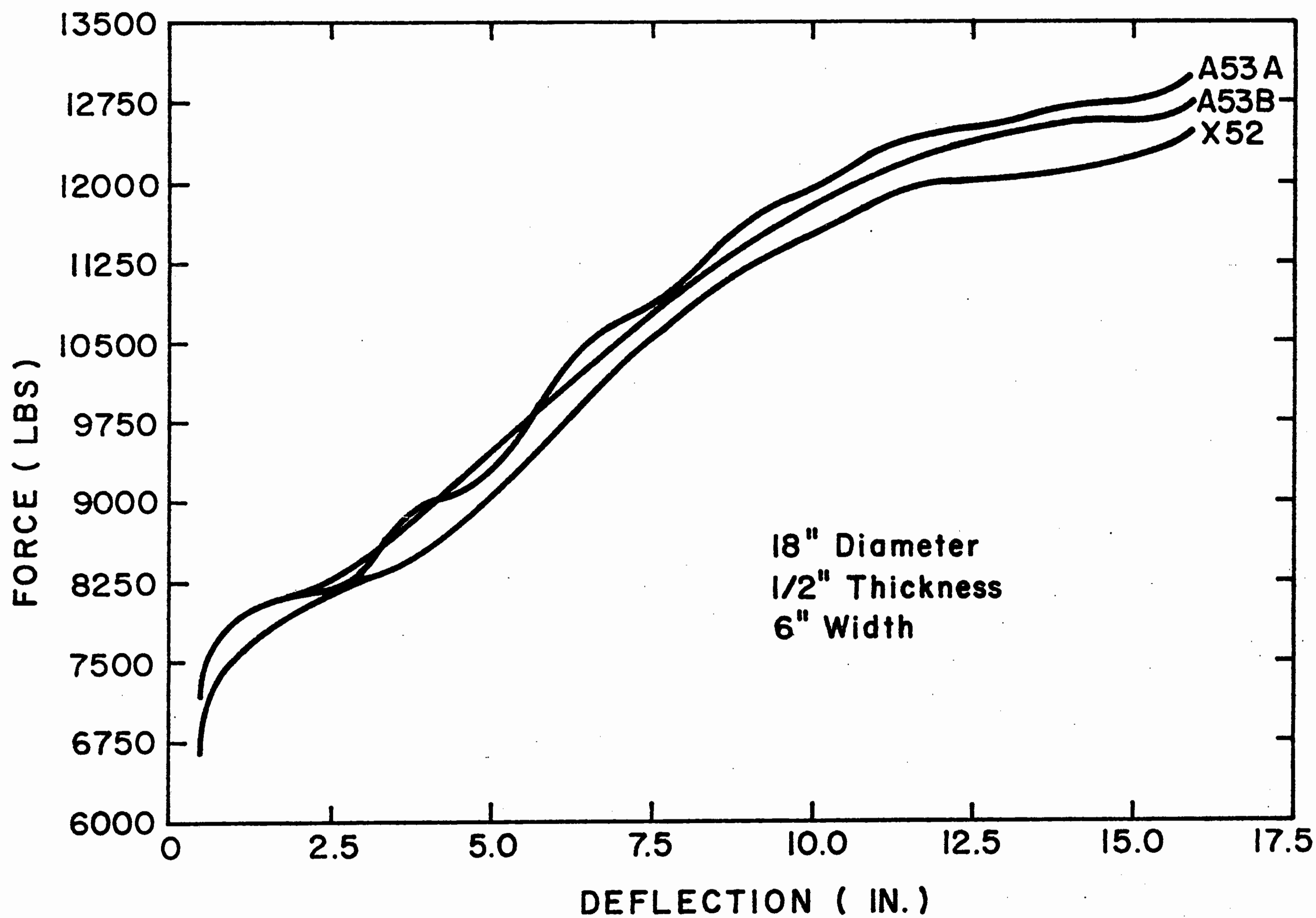

FIG. 6 Force - Deflection Curves for 3 Steel Type Rings

The Collapsing Mode

The individual rings in the energy absorbing system will be loaded as shown in Fig. 7, resulting in a collapsing mode which will be assumed to be associated with the formation of four plastic hinges 90° removed from one another. Applying the theorem of virtual work and equating the internal and external work done yields

$$2\, P_c \left(\frac{R\alpha}{2}\right) = 4\, M_o\, \alpha \tag{17}$$

where

P_c = small deflection static collapse load

R = radius of ring

α = virtual angle change

M_o = yield moment

It follows that the collapse load is

$$P_c = \frac{4\, M_o}{R} \tag{18}$$

But the yield moment at plastic collapse may be written as

$$M_o = \frac{\sigma_o\, Wt^2}{4} \tag{19}$$

where

σ_o = static yield stress

W = depth of ring

t = thickness of ring

which leads to

$$P_c = \frac{\sigma_o\, Wt^2}{R} \tag{20}$$

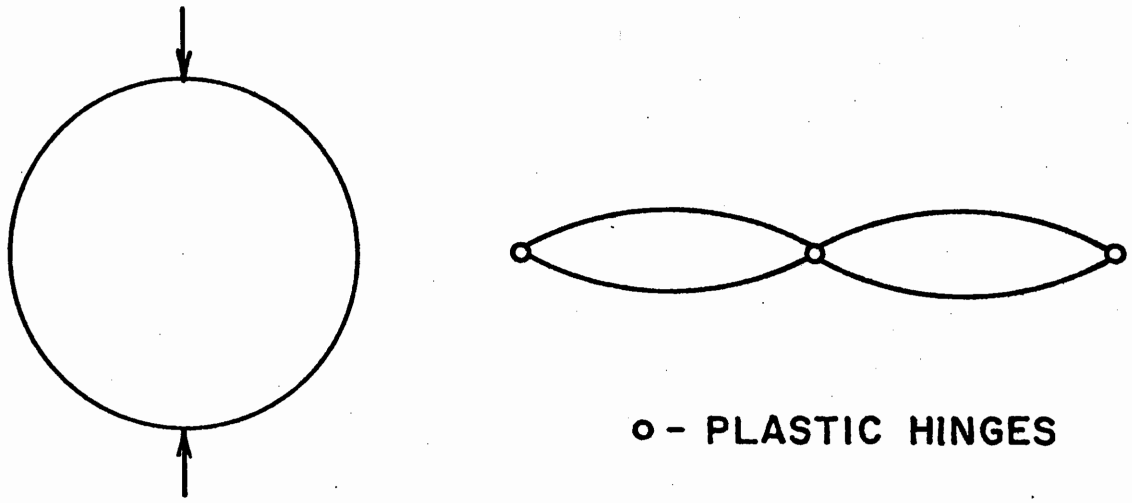

FIG. 7 Loaded Ring and Collapse Pattern for Perfectly Plastic Material

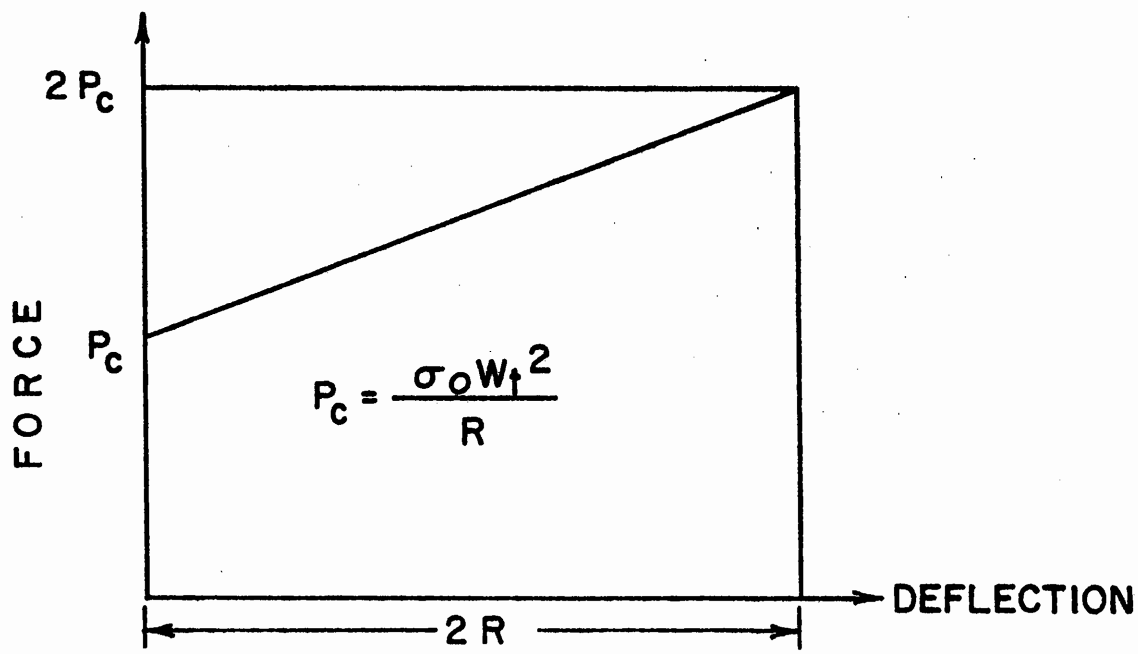

FIG. 8 Approximate Force Deflection Curve For Rings

Based on the uniaxial data shown in Fig. 5, Perrone suggests a value of static yield stress, σ_o, of 39,000 psi. For W = 6", t = 1/2", and R = 9", Eq. 20 then yields P_c = 6500 pounds, which agrees reasonably well with the results of Fig. 6. Noting that the collapsing force increases to approximately 2 P_c as the deformation of the ring increases, Perrone has proposed the force-deflection curve shown in Fig. 8. The energy absorbed in the ring would then be

$$\text{Energy} = 1.5\ P_c\ (2\ R) = 3\ \sigma_o\ Wt^2 \qquad (21)$$

It appears that Perrone's approximation is overly conservative, and a more accurate expression for the area under the load-deflection curve of Fig. 6, equal to the energy absorbed, will be taken in the form

$$\text{Energy} - 1.14\ (3\ \sigma_o\ Wt^2) = 3.42\ \sigma_o\ Wt^2 \qquad (22)$$

Figures 6 and 7 and, therefore, Eq. 22 are valid only under static loading conditions. Structural steel is a rate sensitive material, however, and its properties can change by as much as 100% depending on the strain rates during the deformation process. Much experimental and analytical research has been conducted in this area [26-32]. For the range of strains and strain rates to be encountered in this application, Perrone [19] suggests an overall rate sensitivity factor of 1.6. In Eq. 22, therefore, if the σ_o term is replaced by 1.6 σ_o the equivalent dynamic energy absorbed can be written as

$$\text{Dynamic Energy Absorbed} = 5.47\ \sigma_o\ Wt^2 \qquad (23)$$

For σ_o = 39,000 psi, W = 6 inches, and t = 0.5 inches, Eq. 23 yields a dynamic energy of 26,666 ft-pounds. In work associated with the frangible tube energy absorbing system [18] it was determined that an average force of 10,000 pounds in the tubes while fragmenting over an 18 inch stroke length operated quite successfully. This results in an energy absorption capacity of 15,000 ft-pounds.

If we adopt this 10,000 pound average force level for the thick-walled ring with $\sigma_o = 39,000$ psi, Eq. 23 becomes

$$\frac{Wt^2}{R} = 0.094 \text{ in}^2 \tag{24}$$

A plot of W versus t from Eq. 24 for 12", 18", and 24" diameter rings is shown in Fig. 9. The dynamic energy absorbed of course depends on the radius of the ring. For reasons which will become clear later, the 18 inch diameter thick-walled ring has been selected for use in the portable energy absorption system. The rings can be formed from electric fusion welded pipe manufactured by Bethlehem and U.S. Steel. The pipes come in 20 to 80 feet long sections which can be sliced to the desired lengths.

The Design of the System

The system is to be designed to attach to the rear of the 14,000 pound highway service vehicle shown in Fig. 10. The thick-walled ring system as used in guardrail applications consists of one row of rings cantilevered from a back up post and connected to a box beam. If adopted for use on the trucks such a system might consist of a row of three such rings as shown in Fig. 11a. This configuration is totally inadequate, however, for two reasons:

1) It is incapable of absorbing the tremendous amount of energy generated in a high speed collision (see Fig. 4).
2) It violates the Federal Highway Administration's guidelines on acceptable decelerations and deceleration rates, which are [32]
    a) Permissible average vehicle deceleration $\leq$ 12 g's.
    b) Permissible deceleration onset rate $\leq$ 500 g's per second

With regard to item #2 above, it is possible to calculate the minimum required length of the energy absorbing unit needed to slow a speeding vehicle,

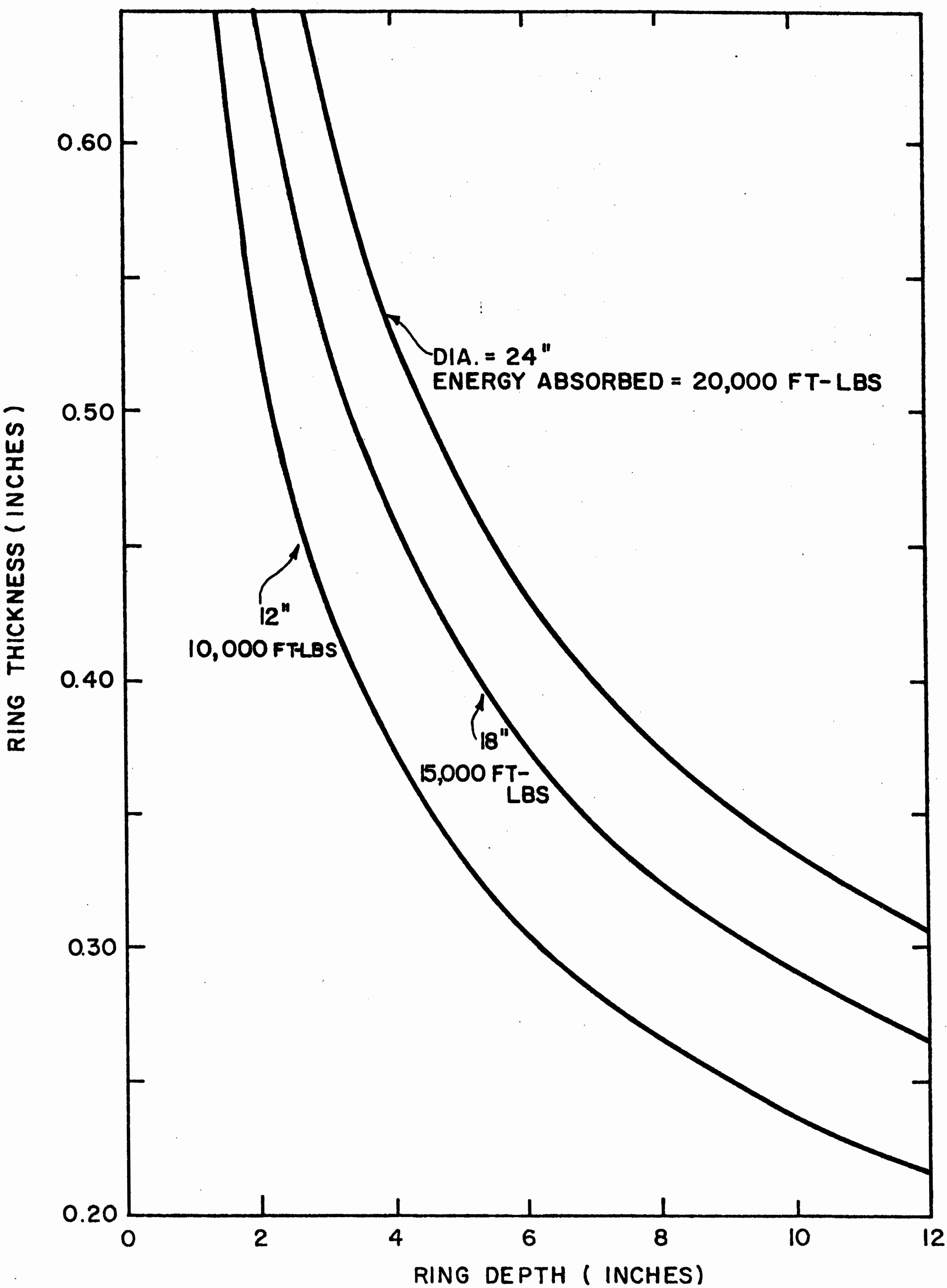

FIG. 9 - RING THICKNESS VS. DEPTH FOR AVERAGE RESISTING FORCE OF 10,000 LBS.

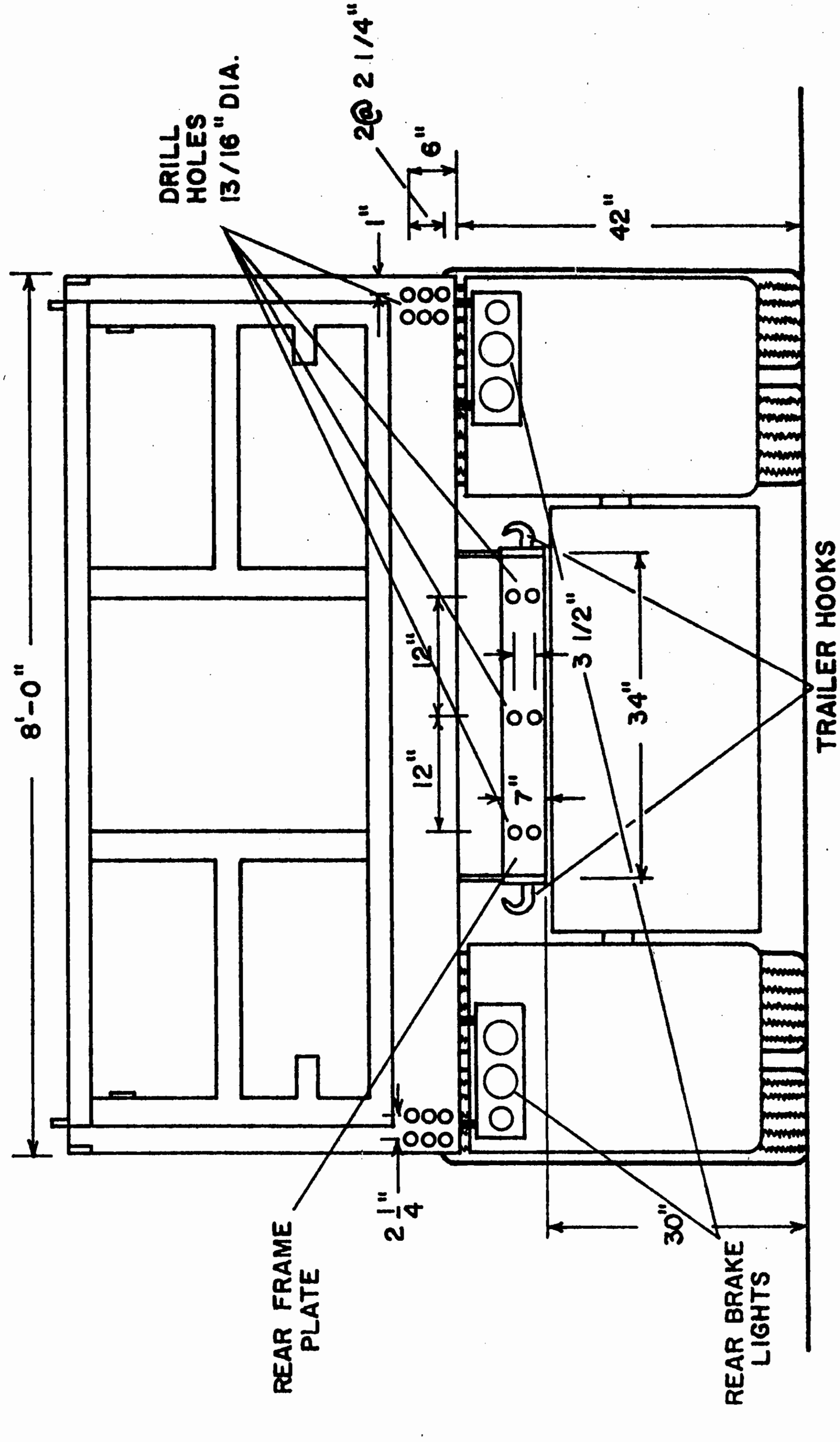

FIG.-10 STANDARD SERVICE VEHICLE with CONNECTION RECOMMENDATIONS as a function of velocity, in order not to exceed 12 g's average deceleration. From dynamics, one can write

$$V_c^2 = V_o^2 + 2as \tag{25}$$

where

V_c = automobile speed before impact

V_o = automobile speed when auto and truck move as a unit

a = average vehicle deceleration

s = required length of energy absorbing unit

Table I lists the required minimum lengths of the energy absorption systems as a function of automobile impact speed for the case when the pre-impact velocity of the service vehicle is 10 mph.

TABLE I

Minimum Required Length of Energy Absorption System

Automobile Impact Velocity (mph)	Required Length (feet)
20	0.70
30	1.92
40	3.68
50	5.97
55	7.31

The above distances are conservative because, in their determination, consideration has not been given to the fact that the service vehicle will be moving during the deformation period of the impact. If a 10 mph service vehicular speed is taken into account, the required lengths for the energy absorbing systems for various automobile impact velocities will be less than those listed in Table I.

While on the subject of minimum length provisions, there are requirements to be met with respect to the minimum spacing distance which should exist between the service vehicle and the vehicle it is following. If the brakes of the truck are locked after the impact, it follows from the conservation of energy that the distance required to stop the system is

$$d = \frac{\text{Kinetic Energy after Impact}}{\mu W_t} \tag{26}$$

where

μ = coefficient of friction

Equation 26 assumes that all of the wheels of the service vehicle are locked. The most critical case occurs when the 4000 pound automobile impacts the service vehicle at 55 mph. The kinetic energy remaining in the system after impact is 240,498 ft-lb and the required stopping distance is, assuming $\mu = 0.7$,

$$d = \frac{240,498}{14,000\ (0.7)} = 24.54 \text{ ft.}$$

If an additional factor of safety of 4 is applied, the recommended safe spacing between the service vehicle and the equipment preceding it would be approximately 100 feet.

With a truck width of 8 feet to work with, 18 inch diameter rings are chosen for the system. Three such rings may then be placed in each row of the unit, and there is adequate space available for the rings to collapse completely without interfering with one another. Assuming an average force of 10,000 pounds in the rings during collapse, each thick-walled ring could supply 15,000 ft-lb of energy dissipation. A row of 3 rings would dissipate

45,000 ft-lb of energy.

The Barrier VII Computer Program

As an independent check on the theory developed in this report, the Barrier VII computer program was applied to an example problem. Consider an energy absorbing system composed of one row of three rings as shown in Fig. 11a. The actual automobile velocity at impact was taken as 35 mph. Figure 3 then yields the appropriate revised automobile velocity to be used as input in the Barrier VII program, namely 22 mph for the 4000-14,000 pound case. The results of the Barrier VII run show complete collapse of the rings. This predicted result agrees with the data contained in Fig. 4. In Fig. 4, corresponding to an actual automobile velocity of 35 mph, the amount of required energy dissipation is in excess of 60,000 ft-lb. Under a 35 mph impact, therefore, the system depicted in Fig. 11a is inadequate. According to the results of Fig. 4, the unit will collapse completely when impacted by a 4000 pound automobile traveling at approximately 30 mph, corresponding to a 45,000 ft-lb energy dissipation level.

In view of the above results, it was decided to develop a design utilizing rows of rings, as shown in Fig. 11b. The system is to be designed to adequately absorb the energy imparted to it by a 4000 pound automobile traveling at 55 mph. It is seen from Fig. 4 that the required energy absorption capacity of the system is approximately 210,000 ft-lb. If the system were composed of rings with individual energy dissipation potentials of 15,000 ft-lb, then 14 rings would be required.

The Federal Highway Administration's guidelines on acceptable decelerations must also be considered in the design. According to Table I, a 12g average deceleration level under a 55 mph impact requires a unit 7.31 feet long. Five

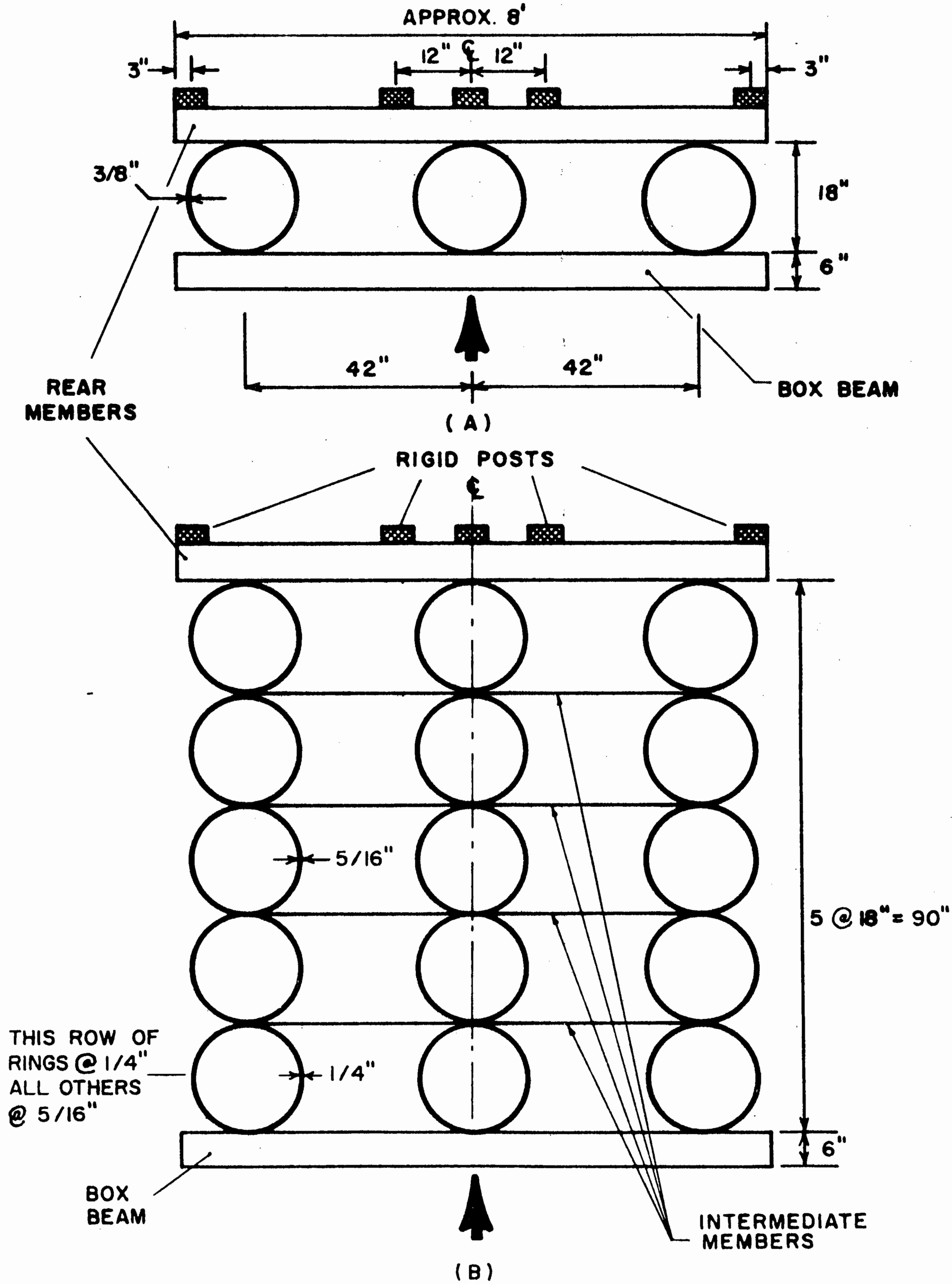

FIG.11 - PRELIMINARY RING CONFIGURATIONS

rows of 3 rings with 18 inch diameters furnish 7.5 feet.

The energy absorbing system should also be functional when impacted by a small automobile weighing, say, 2500 pounds. The energy dissipation versus actual car velocity curve for this 2500-14,000 pound case is not presented in Fig. 4, but the locus of its points lies below the two curves shown. In particular, for the 55 mph collision case, 143,445 ft-lb of energy must be dissipated. This energy dissipation level calls for 9.56 rings of 15,000 ft-lb capacity. This would result in only 4.75 feet of the energy absorbing system being deformed, however, and according to Table I, 7.31 feet is needed.

To handle this problem, a polymodular design will be used. The first row of rings (that row nearest the impact end of the unit) will be designed such that each ring will furnish a smaller average force during collapse than the 10,000 pound rings to provide an adequate collapse length for the system. The rest of the unit will be composed of rings having a 10,000 pound average force capacity.

Dimensions of the Rings

The size of the 10,000 pound capacity rings is determined from Fig. 9. Because of design constraints and lateral stability requirements, the 9 inch deep ring has been selected. From Fig. 9, it follows that a corresponding ring thickness of 0.31 inches is required. A ring thickness of 5/16 inches (0.3125 in) will therefore be used. The energy absorbing unit is to be designed for a 210,000 ft-lb capacity. Four rows of 3 - 15,000 ft-lb rings per row supply 180,000 ft-lb. The row of rings nearest to the impact end of the unit must therefore furnish 30,000 ft-lb of energy, or 10,000 ft-lb per ring. Using this 10,000 ft-lb figure in Eq. 23 along with σ_o = 39,000 psi and W = 9 inches yields a required ring thickness of 0.25 in. for the first row of rings.

The 2500 pound impact case should now be checked. The 143,445 ft-lb of energy will be absorbed by the system by a total collapse of the first three rows of rings (accounting for 120,000 ft-lb of energy dissipation) and a partial collapse of the fourth row of rings, resulting in a total deformation of approximately 5-1/2 feet. In view of the fact that the 7.31 foot requirement from Table I is conservative as previously noted, this 5-1/2 foot deformation length is considered adequate.

Several attempts were made to simulate the proposed design in the Barrier VII program. However, numerical instability problems developed. To quote from the Barrier VII report [25]: "numerical instability in the solution procedure has been a constant plague, and may still persist in some cases. For barriers which have fairly simple geometric configurations and are made up of simple numbers, the program should work well. For some complex barriers, however, experimentation in the selection of time steps, damping values and other parameters may be necessary to obtain a stable solution." After several attempts to solve the numerical instability problems encountered, and because of the expense involved in running the program, it was decided to discontinue using the Barrier VII package. However, in view of the excellent correlation between the theory developed in this report and the Barrier VII program output for the 35 mph impact with the system composed of one row of three rings, this 5-row system appears to be on solid ground.

The Energy Absorbing System

The design is shown in Figs. 12-15. Five rows of rings are cantilevered from the rear of the service vehicle through a W12 x 19 section which is

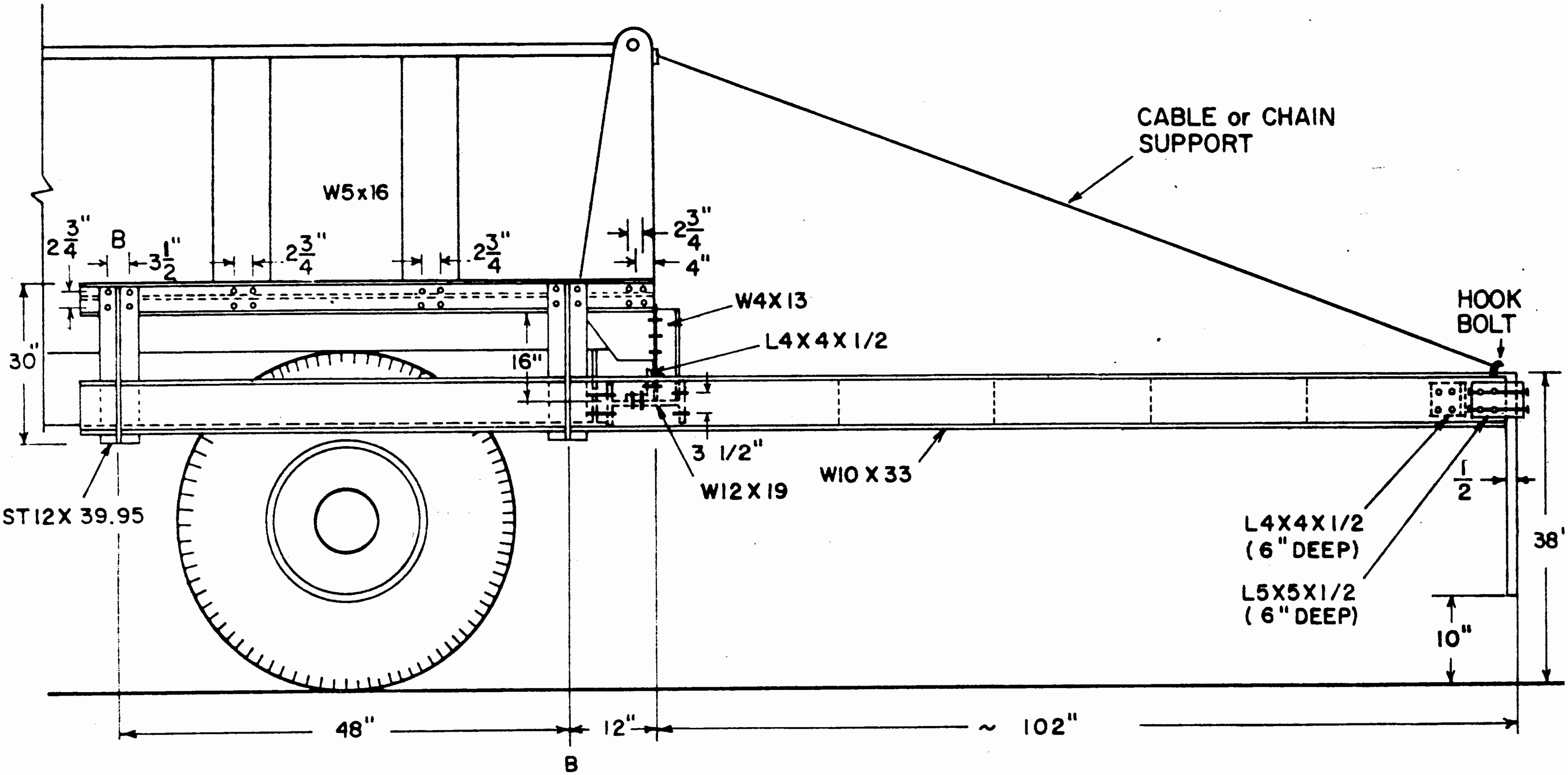

FIG. 12 SIDE VIEW (3/4" HSB FOR FIXTURE CONNECTIONS)

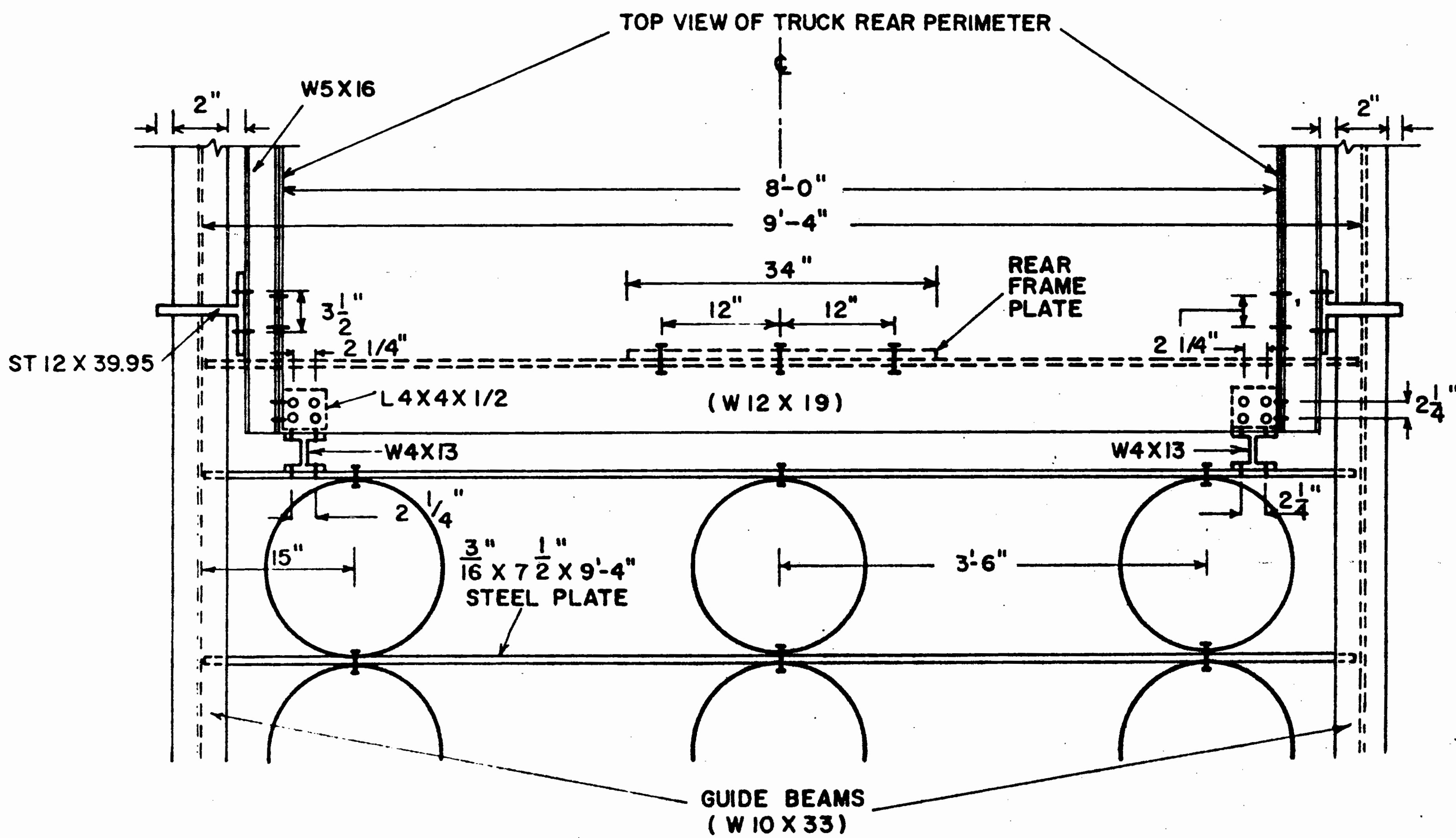

FIG. 13 - TOP VIEW (ALL CONNECTIONS 3/4" DIA. HIGH STRENGTH BOLTS WITH 13/16" DIA. HOLES)

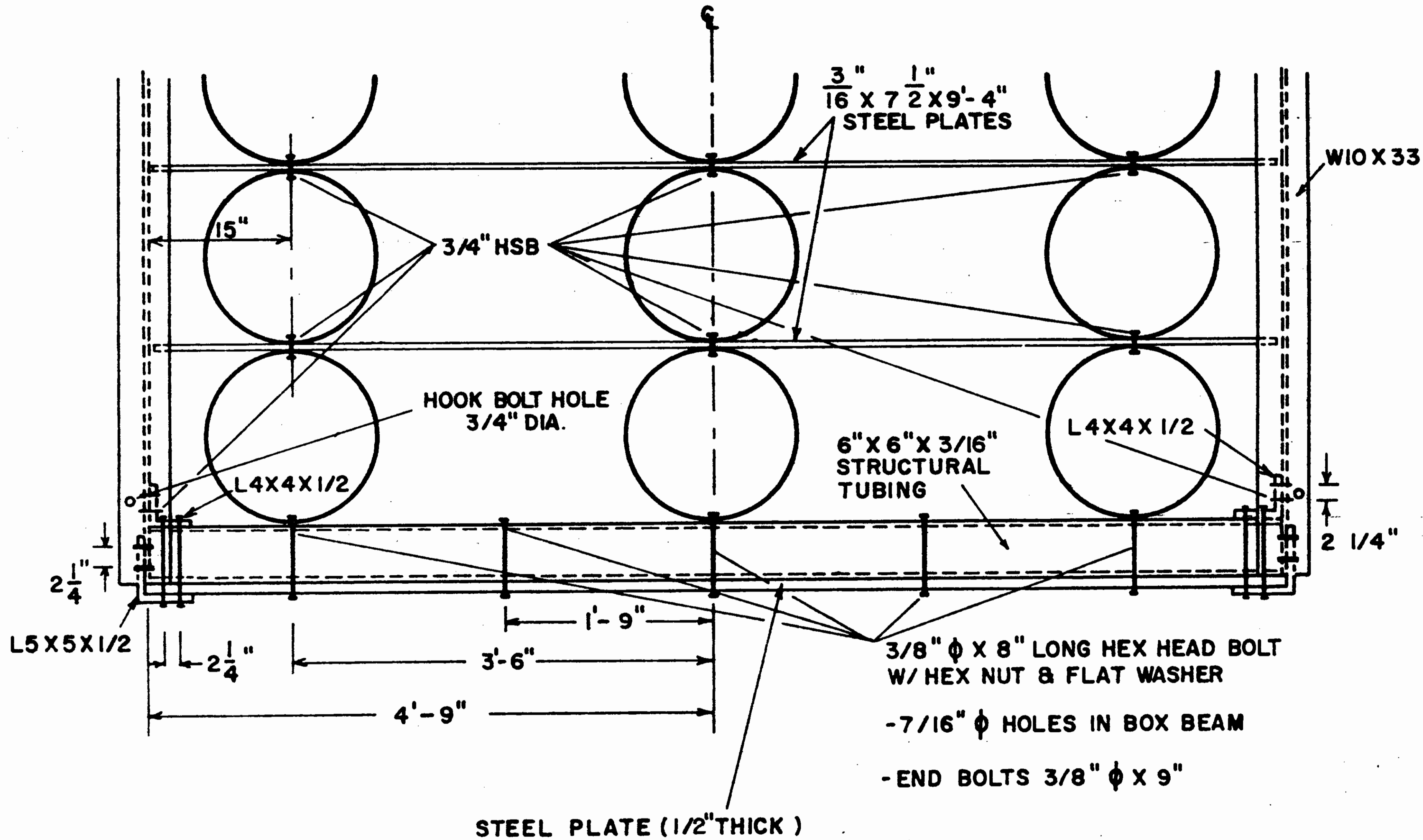

FIG. 14 - TOP VIEW OF REAR SEGMENT (HSB HOLES DRILLED TO 13/16" DIA.)

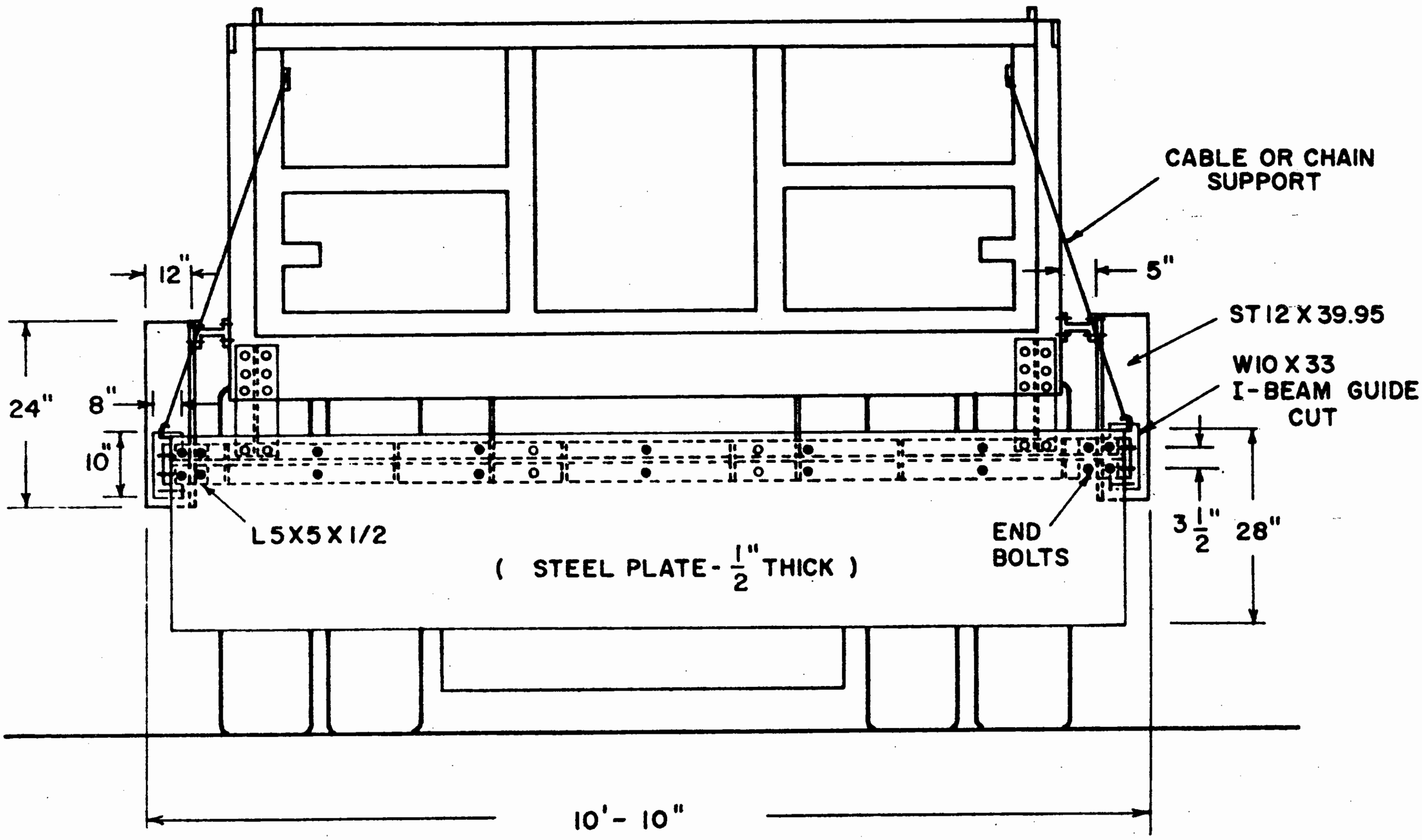

FIG. - 15 FULL ASSEMBLY - REAR VIEW (•-3/8" ϕ HEX HEAD BOLTS) bolted to the rear frame plate of the truck in three places and connected at the ends to the truck by W4 x 13 sections. Narrow steel shim plates may be required between the W4 x 13 sections and the rear truck bed to provide for a proper connection between the W4 x 13 section and the W12 x 19 section. The rear brake lights and trailer hooks on the service vehicle will have to be relocated or adjusted to accommodate the W12 x 19 rear frame section. The rings in the four rows closest to the truck body are 9 in. deep, 18 in. in diameter, and 5/16 in. in thickness. The rings of the row nearest the impact are 9 in. deep, 18 in. in diameter, but only 1/4 in. thick. This row of rings frames onto a 6 x 6 x 3/16 structural tubing (box beam).

The entire unit is centered approximately 33-1/2 in. above the road surface. The location of the center of gravity for most passenger vehicles, however, is approximately 24 in. above the roadway. A 1/2 in. thick plate is connected to the outside edge of the box beam extending down to a height of 10 in. above the road surface to act as a load transfer member. With the exception of the 8 in. long, 3/8 in. diameter hex head bolts passing through the box beam, all bolted connections in the system are to be made with 3/4 in. diameter high strength bolts. To assist in the transfer of the eccentric impact load to the rings, a 9 x 4 x 1/2 angle section should be welded along the length of the plate and the underside of the box beam (not shown in Fig. 12) with 7/16 in. fillet welds. The 9 inch leg should run along the impacting plate.

To furnish lateral stability to the system, side members in the form of W10 x 33's are rigidly connected to the ends of the box beam. To cut down on stress concentrations at the connection, it is recommended that two 4 x 3-1/2 x 1/4 angles, 12 in. long (not shown on figures) be inserted between the web of the W10 x 33 section and the outside 5 x 5 x 1/2 angle. The 4 in. legs should run along the web of the W10 x 33 while the 3-1/2 in. legs are to be connected to the flanges of the W section. The W10 x 33 side members pass through two guide members spaced 4 feet apart on each side of the truck. These guide members are ST 12 x 39.95's. Cutouts are specified in the web of the structural tees to permit the W10 x 33 members to slide through during an impact situation. To facilitate the sliding action, teflon pads should be attached to both flanges of the W10 x 33 members and along the cutouts in the structural tees. These structural tees are connected to a W5 x 16 member which is attached to the side of the truck. This offset is necessary to provide adequate clearance for the system.

In the horizontal plane, then, the system forms a rigid frame to provide the stiffness necessary to resist a force parallel to the box beam associated with an off-normal or off center impact. In fact, the potential of an off-normal or off center impact loading was the controlling factor in the design of many of the system's components.

Between the rows of rings, 7-1/2 in. deep, 3/16 in. thick steel plates are inserted. These members provide vertical stability to the system since they are guided top and bottom at their extremities by the flanges of the W10 x 33 guide members. These 7-1/2 in. steel plates are not attached to the side members. The bolted connection between the rings and the steel plates consists of 2-3/4 in. diameter high strength bolts symmetrically located 5 in. apart.

To cut down on vibrations and more evenly distribute the dead weight of the energy absorption system, a cable or chain support runs from the top of the rear truck panel to a hook bolt in the side member near the box beam. This support will go slack upon impact and will not affect the dynamic behavior of the system.

Recommendations

An extensive full scale testing program should be conducted on the portable energy absorbing system designed herein. In particular the following specific tests are recommended:

1) W_c = 4000 lbs | W_t = 14,000 lbs
V_c = 35 mph (27.5) | V_t = 10 mph (0)
θ = 0° | central impact

2) W_c = 4000 lbs | W_t = 14,000 lbs
V_c = 55 mph (47.4) | V_t = 10 mph (0)
θ = 0° | central impact

3) W_c = 2500 lbs | W_t = 14,000 lbs
V_c = 55 mph (46.7) | V_t = 10 mph (0)
θ = 0° | central impact

4) W_c = 4000 lbs | W_t = 14,000 lbs
V_c = 35 mph (27.5) | V_t = 10 mph (0)
θ = 0° | left rear edge impact

5) W_c = 4000 lbs | W_t = 14,000 lbs
V_c = 55 mph (47.4) | V_t = 10 mph (0)
θ = 0° | left rear edge impact

6) W_c = 4000 lbs | W_t = 14,000 lbs
V_c = 55 mph (47.4) | V_t = 10 mph (0)
θ = 10° | left rear edge impact

The fact that both vehicles are moving at impact increases the complexity of the proposed tests. Test No. 6, in particular, will be difficult to run because of the non-zero θ value. It is desirable, therefore, to determine the appropriate automobile velocities for the six tests corresponding to a pre-impact truck velocity of zero mph instead of 10 mph. In finding these "equivalent" velocities, V^*, it will be required that the total amount of energy dissipated in each case be equal to the amount expended in the original (V_t = 10 mph) six tests. Kinetic energy and conservation of momentum considerations lead to an expression for the "equivalent" pre-impact automobile velocity in the form

$$V^* = \left[V_c^{\,2} + \frac{W_t}{W_c} V_t^{\,2} - \frac{W_t V_t}{W_c} \left(\frac{2W_c V_c + W_t V_t}{W_c + W_t} \right) \right]^{1/2} \tag{27}$$

The specific values of V^* from Eq. 27 for the six proposed tests are given in parenthesis next to the "real" values of V_c for each case. The corresponding V_t value of zero mph is also written in parenthesis.

In summary, the portable thick-ring energy absorption system presented herein appears functional. A prototype design should be subjected to the full scale tests described above to assess the performance of the system.

During the proposed testing program the performance of the system should be constantly evaluated, and appropriate changes in the thickness of some of the rings should be made between tests if necessary. In addition, if the measured deceleration levels in test No. 3 involving the compact automobile should prove excessive, an additional row of rings with 1/4" thicknesses could be added to the system at the box beam. This additional row of thick-walled rings should only be installed if proven necessary since it will add to the weight, length, and expense of the system.

The fact that both vehicles are moving at impact increases the complexity of the proposed tests. Test No. 6, in particular, will be difficult to run because of the non-zero θ value. It is desirable, therefore, to determine the appropriate automobile velocities for the six tests corresponding to a pre-impact truck velocity of zero mph instead of 10 mph. In finding these "equivalent" velocities, V^*, it will be required that the total amount of energy dissipated in each case be equal to the amount expended in the original (V_t = 10 mph) six tests. Kinetic energy and conservation of momentum considerations lead to an expression for the "equivalent" pre-impact automobile velocity in the form

$$V^* = \left[V_c^{\,2} + \frac{W_t}{W_c} V_t^{\,2} - \frac{W_t V_t}{W_c} \left(\frac{2W_c V_c + W_t V_t}{W_c + W_t} \right) \right]^{1/2} \tag{27}$$

The specific values of V^* from Eq. 27 for the six proposed tests are given in parenthesis next to the "real" values of V_c for each case. The corresponding V_t value of zero mph is also written in parenthesis.

In summary, the portable thick-ring energy absorption system presented herein appears functional. A prototype design should be subjected to the full scale tests described above to assess the performance of the system.

During the proposed testing program the performance of the system should be constantly evaluated, and appropriate changes in the thickness of some of the rings should be made between tests if necessary. In addition, if the measured deceleration levels in test No. 3 involving the compact automobile should prove excessive, an additional row of rings with 1/4" thicknesses could be added to the system at the box beam. This additional row of thick-walled rings should only be installed if proven necessary since it will add to the weight, length, and expense of the system.

It should be emphasized that the proposed system has been designed to handle the energy absorption requirements associated with a high speed collision of an <u>automobile</u> with a 14,000 lb. service vehicle. The system will not <u>totally</u> absorb the energy generated in a high speed collision with a heavy truck.

References

1. Hirsch, T. J., Barrel Protective Barrier, Technical Memorandum 505-1, Texas Transportation Institute, July 1968.

2. Hirsch, T. J., and Ivey, D. L., Vehicle Impact Attenuation By Modular Crash Cushion, Research Report 146-1, Texas Transportation Institute, June 1969.

3. White, M. C., Ivey, D. L., and Hirsch, T. J., In-Service Experience on Installations of Texas Modular Crash Cushions, Research Report 146-2, Texas Transportation Institute, December 1969.

4. Hirsch, T. J., Hayes, G. G., and Ivey, D. L., The Modular Crash Cushion, Technical Memorandum 505-1S, Texas Transportation Institute, August 1970.

5. Nordlin, E. F., Woodstrom, J. H., and Doty, R. N., Dynamic Tests of an Energy-Absorbing Barrier Employing Steel Drums, Highway Research Record 343, 1971, pp. 123-141.

6. White, M. C., Hayes, G. G., and Hirsch, T. J., A Feasibility Study of Using Corrugated Steel Pipes in Modular Crash Cushions, Technical Memorandum 505-18, Texas Transportation Institute, August 1971.

7. Warner, C. Y., and Free, J. C., Water-Plastic Crash Attenuation System: Test Performance and Model Prediction, Highway Research Record 343, 1971, pp. 83-92.

8. Hayes, G. G., Ivey, D. L., and Hirsch, T. J., Performance of the Hi-Dro Cushion Cell Barrier Vehicle-Impact Attenuator, Highway Research Record 343, 1971, pp. 93-99.

9. Nordlin, E. F., Stoker, J. R., and Doty, R. N., Dynamic Tests of an Energy-Absorbing Barrier Employing Sand-Filled Plastic Barrels, Highway Research Record 386, 1972, pp. 28-51.

10. Hirsch, T. J., Tor-Shok Energy Absorbing Protective Barrier, Technical Memorandum 505-2, Texas Transportation Institute, July 1968.

11. Hirsch, T. J., Smith H. L., and Ivey, D. L., Tor-Shok and Roto-Shok Energy Absorbing Protective Barriers, Technical Memorandum 505-2S, Texas Transportation Institute, January 1969.

12. Hayes, G. G., Hirsch, T. J., and Ivey, D. L., Dragnet Vehicle Arresting System, Highway Research Record 306, 1970, pp. 39-49.

13. Ivey, D. L., Buth, E., and Hirsch, T. J., Feasibility of Lightweight Cellular Concrete for Vehicle Crash Cushions, Highway Research Record 306, 1970, pp. 50-57.

14. Ivey, D. L., Buth, E., Hirsch, T. J., and Viner, J. G., Evaluation of Crash Cushions Constructed of Lightweight Cellular Concrete, Highway Research Record 386, 1972, pp. 10-18.

15. Viner, J. G., and Boyer, C. M., Accident Experience With Impact Attenuation Devices, Report No. FHWA-RD-73-71, Federal Highway Administration, April 1973.

16. McGehee, J. R., A Preliminary Experimental Investigation of an Energy-Absorption Process Employing Frangible Metal Tubing, NASA TN D-1477, NASA, October 1962.

17. McGehee, J. R., Experimental Investigation of Parameters and Materials for Fragmenting-Tube Energy-Absorption Process, NASA TN D-3268, NASA, February 1966.

18. Woolam, W. E., An Impact-Energy Attenuating Device Combined with Guardrail-like Structures, SWRI Project 02-2372, FHWA Project FH-11-6863, January 1970.

19. Perrone, N., Thick-Walled Rings for Energy-Absorbing Bridge Rail Systems, Report No. FHWA-RD-73-49, December 1972.

20. Bronstad, M., Development of a New Energy-Absorbing Bridge Railing, Quarterly Report No. 1, DOT Project FH-11-7985, April 1973.

21. Michie, J. D., Development of a New Energy-Absorbing Bridge Railing, Quarterly Report No. 2, DOT Project FH-11-7985, July 1973.

22. Kimball, C. E., Development of a New Energy-Absorbing Bridge Railing, Quarterly Report No. 3, DOT Project FH-11-7985, October 1973.

23. Marquis, E. L., Hirsch, T. J., and Nixon, J. F., Texas Crash Cushion Trailer for Highway Maintenance Operations, Presented at 53rd Annual Meeting Highway Research Board, January 1973.

24. Powell, G. H., Computer Evaluation of Automobile Barrier Systems, Report to U. S. Department of Transportation, Federal Highway Administration, Report No. UC SESM70-17, Department of Civil Engineering, University of California, August 1970.

25. Powell, G. H., Barrier VII: A Computer Program For Evaluation of Automobile Barrier Systems, Report No. FHWA-RD-73-51, April 1973.

26. DeRuntz, J. A., and Hodge, P. G., Crushing of a Tube Between Rigid Plates, Journal of Applied Mechanics, American Society of Mechanical Engineers, Vol. 30, September 1963, pp. 391-395.

27. Perrone, N., On a Simplified Method for Solving Impulsively Loaded Structures of Rate-Sensitive Materials, Journal of Applied Mechanics, American Society of Mechanical Engineers, Vol. 32, September 1965, pp. 489-492.

28. Perrone, N., A Mathematically Tractable Model of Strain-Hardening, Rate-Sensitive Plastic Flow, Journal of Applied Mechanics, American Society of Mechanical Engineers, Vol. 33, March 1966, pp. 210-211.