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PROGRAM FOR THE EXPLOITATION OF UNUSED NASA PATENTS

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PROGRAM FOR THE EXPLOITATION OF UNUSED NASA PATENTS

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I. INTRODUCTION

The purpose of this program is to exploit unused NASA patents through the use of a multidisciplinary approach involving faculty, students, and research staff. NASA patents are screened for their applicability outside the space program, specific applications are identified, the technical and commercial feasibility of these applications is established. Also application of this technology by governmental agencies outside the space program is sought.

The program is specifically interested in energy absorbing devices such as those developed for lunar soft landings. These energy absorbing devices absorb large amounts of mechanical energy but are, in general, not reusable. Some of these devices can also operate as structural elements until their structural load capacity is exceeded and they become activated as energy absorbers. The capability of these devices to operate as structural elements and as energy absorbing devices makes them candidates for many applications in the fields of transportation and materials handling safety where accidents take a large toll in human injury and property damage.

The ultimate objective is the commercial application of NASA technology with the university receiving a financial return with which to make the program self-supporting. A unique feature of this program is the interaction of the students with the faculty and the research staff in the solution of technology transfer problems. Students are assigned to groups which work directly with faculty and research staff members doing mathematical analysis, testing, design, market analysis, business management and financial analysis, library system development, and production analysis. The students derive great benefit from working on real problems while the program benefits from the ideas generated by the students and the work done by them.

During the first year the efforts on the program were directed toward the selection of the type of patents which would be considered, the screening of the patents to determine those having the greatest technical and commercial feasibility for a wide range of applications outside of the space program, and the definition of the general areas of applicability for these patents. Fifty-five patents on energy absorbing devices falling into 18 categories were evaluated. These included 12 NASA patents and 43 competing patents. In addition to this, literature concerning energy absorbing devices and applications was reviewed. Of the energy absorbing devices evaluated, two devices were considered to embody the combination of features required for a wide range of applications; these are the tube and mandrel (NASA Patent No. 3, 143, 321) and the folding tube (there are no patents on the tube, as considered here, without special end fixtures or preforming). The attractive features of these devices are their simplicity, high stroke to length ratio, low cost per unit of energy absorbed, and their applicability as structural elements.

During the second year the emphasis has been placed on activities which will promote the commercial applications of the technology and also secure support from other governmental agencies to study public safety applications for energy absorbing devices. These activities included the following:

1. The identification of companies and governmental agencies operating in the fields of possible application for energy absorbing devices.

2. Visits to companies and governmental agencies to discuss potential applications.

3. Analysis of energy absorber requirements based on information received concerning specific applications.

4. The development of energy absorbers based on the tube and mandrel but having unique features and characteristics required for specific applications.

5. The determination of the cost of producing specific energy absorbers.

6. Basic research on the tube and mandrel and folding tube energy absorbers.

7. The development of a library containing literature in the fields of potential applications.

8. The presentation of papers concerning the work on the project.

II. PROGRAM ORGANIZATION

The shift in emphasis from patent evaluation to patent application has necessitated some reorganization and the redefinition of areas of responsibility of the professional staff to maximize the effectiveness of the staff and students working on the program. The key activities during the second year have been the identification of specific candidate applications for the energy absorbing devices in commercial and public safety areas and the demonstration of the technical and commercial feasibility of these applications. The basic research on the energy absorbing devices has continued to support these activities.

The organizational structure of the project during the second year is illustrated by Figure 2.1. The role of Project Manager has been assumed by Richard J. Fay acting under the direction of the Principal Investigator, Dr. Arthur A. Ezra. The activities on the program have been subdivided into seven areas including promotion, design and development for specific applications, energy absorber research, highway safety applications, motor vehicle safety applications, commercial feasibility analysis, and educational activities. Most of the staff were involved in more than one of these activities. In Figure 2.1 the names are underlined to indicate the principal activity of each staff member.

There were a total of 20 students involved in the project during the year; three of these were graduate students. The students were divided into groups which were assigned to individual staff members and given specific tasks to accomplish. Working in this manner, students participated in a valuable educational experience while contributing significantly to accomplishment of the program's objectives.

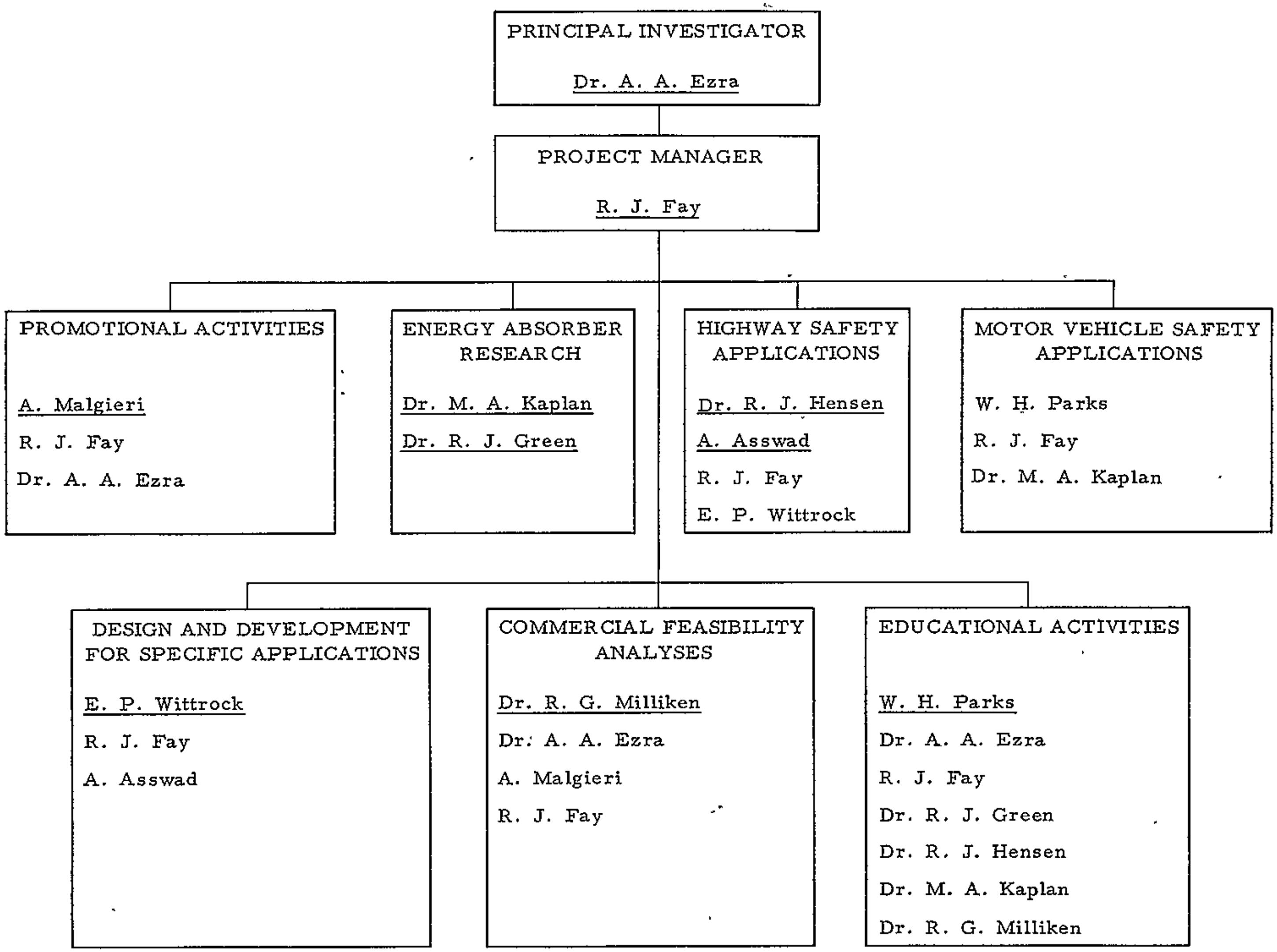


Figure 2.1 Organizational Chart

III. THE TECHNOLOGY TRANSFER PROCESS

1. Basic Considerations

In some cases technology transfer occurs more or less automatically with advances in technology in one area becoming known, accepted, and utilized for applications in areas other than those for which they were originally developed. However, much of the technology isn't automatically transferred even though good applications exist in other areas. The reasons for this are many. The technology may be unknown to those working in these other areas or may not be understood sufficiently by them to apply it or there may be resistance to change. Also the application of a new technology requires that initiative be taken; a company's resources must be expended in the determination of the technical and economic feasibility of the application and this expenditure can be somewhat speculative. After this, there may be a considerable development cost involved before the technology is actually incorporated in a product.

In the operation of companies competing in the market place, economic considerations are very important. For an application of new technology to be commercially feasible, it must be possible for a company to do the required development with reasonable assurance that the investment can be recovered and that a profit can be made. It is important, therefore, that some degree of protection be available such as an exclusive license on a patent. Also the product which incorporates this new technology must promise to have a strong position in the market place or, preferably, an unreasonable advantage for the period required to recover the initial investment.

In view of the preceeding considerations it can be seen that the technology transfer process has need of an agent to identify fields of possible application for areas of technology outside of those for which they were originally developed and to educate individuals working in these fields about the technology so that they can see possible applications for it in their company's product lines. The agent is also needed to find ways to overcome the barriers to technology transfer when he finds a good specific application. This may require that he seize the initiative and use his own resources to determine the technical and commercial feasibility of an application. He may even have to do some or all of the development work before the technology becomes attractive to a company.

The technology agent must be supported either by a subsidy or by participating in the commercialization of the technology. Several possible methods for doing this exist.

2. Approach Taken on This Program

At the beginning of the program energy absorbing devices were selected as the area of technology which would be exploited. This decision was made with the guidance of members of the National Inventors Council who considered energy absorbing devices to be very timely in light of the current national awareness of motor vehicle and highway safety needs.

During the initial phase of the program over 50 patents on energy absorbing devices including 12 NASA patents were reviewed. Screening criteria were developed to facilitate the evaluation of the patents. These included Mechanical efficiency, ratio of total length to maximum stroke, reliability, ability to withstand exposure to weather, and cost. The evaluation process included mathematical modeling and testing. Two energy absorbing devices were found to satisfy all of the screening criteria. These were the NASA tube and mandrel and the folding tube. Both of these devices also possess the ability to operate as structural elements prior to being loaded sufficiently to become activated as energy absorbers.

Possible applications for the energy absorbing devices were conceived using a combination of individual experience, intuition, analysis, and luck. The possible application of the devices in the fields of auto and highway safety were the first to be identified. Fields of possible application were also identified in other forms of transportation including transport trailers, cargo handling equipment, and elevators and in earthquake protection for buildings. After identifying the fields of possible application, companies and governmental agencies operating in these fields were identified and contacted. Individuals in these organizations were visited and given presentations concerning the technical aspects of the energy absorbing devices and the methods and goals of the program.

In the course of these visits the individuals were encouraged to discuss possible applications for the technology in their own areas. They were also asked to define the requirements for possible applications so that we could make analyses to determine the feasibility of the applications. Table 3.1 summarizes the visits made and the applications studied during the year. In the cases where we have had continuing interest we have taken the initiative for demonstrating that the application was technically and commercially feasible. Discussions of these analyses are given in Section VI, VII, VIII and IX.

As stated before the technology transfer process, as we have practiced it, requires financial support either in the form of a subsidy or a percentage of the proceeds from the commercialization of the technology. While this program is presently supported by a grant from NASA it is our intention to generate support from the commercialization of the technology so that the program can be self supporting. The support for the project can come from royalties or profits from the sale of products incorporating the technology. It may be desirable in some cases to set up new companies to make and sell these products. In Section XI the requirements and financial considerations involved in the start of a new company to manufacture an energy absorbing device for application on the front support structure of semitrailers is discussed as an example of what might be done.

TABLE 3.2.1

Summary of Visits Made and Applications Studied During the Year

Company or Agency	Applications Discussed	Applications Given Further Study
Aeroquip Corp. Jackson, Michigan	Cargo Restraint Systems	
American Motors Corp. Detroit, Michigan	Auto Frontal and Rear Structure	
Allied Chemical Corp. Detroit, Michigan	Auto Passenger Restraints, Integral Seat and Restraint	Integral Seat and Restraint with Energy Absorber
Budd Co. Ft. Washington, Pa.	Auto Frames and Bumpers	
Crysler Corp. Detroit, Michigan	Auto Frontal and Rear Structures	
Ford Motor Co. Dearborn, Michigan	Auto Frontal and Rear Structures	
Freightmaster Corp. Ft. Worth, Texas	Railroad Car Coupling	
Fruehauf Corp. Detroit, Michigan	Side and Rear Under Ride Protection for Autos, Dock Bumpers and Front Support Structure	Front Support Structure Drop Protector
General Motors Detroit, Michigan and Saginaw Division Saginaw, Michigan	Frontal and Rear Structures for Autos, Safety Steering Column, and Occupant Restraints	Energy Absorber Under Study to Meet G.M. Requirements
Lord Mfg. Co. Erie, Pa.	Applications to Cargo Handling Equipment and Dock Bumpers	Discussions Continuing
Signode Corp. Chicago, Illinois	Cargo Handling, especially as load limiters for steel bands	
Sea-Land Service, Inc. Elizabeth, N.J.	Cargo Protecting Bulk Heads	
Otis Elevator Co. New York, New York	Elevator Safety System	Elevator Safety System
Dept. of Transportation National Highway Safety Bureau	Controlled Collapse Frontal Structures for Autos, Passenger Safety Seat	Proposal Written and Submitted on Collapsing Structures; Proposal in Progress on Safety Seat
Dept. of Transportation Bureau of Public Roads	Use of Energy Absorbers for Cushioning Fixed Highway Structures	Discussions Continuing

IV. BASIC RESEARCH ON THE TUBE AND MANDREL DEVICE

Figure 4.1 shows the tube and mandrel energy absorbing device developed by NASA for the space program. Such a device consists of a tube pressed upon a mandrel. The end of the tube on the mandrel fractures and deforms thus absorbing energy. Such a device has several excellent features. It can be designed to give a flat force deflection diagram as pictured in Figure 4.2. There may be an initial spike to initiate the cracks but if notches are cut into the end of the tube and the inside of the tube beveled this is essentially eliminated. It is simple and easy to manufacture consisting of only two parts, a tube, easily obtained commercially and a mandrel of simple conical design. As opposed to many energy absorbers it utilizes almost the total length of the device to absorb energy thus extracting the most energy out of the available space. Different force levels are easily accomplished by increasing the tube thickness or diameter or by changing the mandrel configuration.

The basic research on the tube and mandrel energy absorber during the year has been directed toward the development of a mandrel which could be produced at less cost than the curved mandrel used by the inventor, McGehee, and the development of analytical predictions of the operating force of the device. Two mandrel configurations have been developed, a truncated cone and a truncated cone with a base which give very good results while having configurations which are more adaptable to mass production. Equations have been derived which predict the force required to operate the device with these two mandrel configurations using hard tube materials such as 2024-T3 aluminum and 4130 steel. An equation has also been developed to predict the force required to operate a device consisting of a ductile tube with a conical mandrel without a base.

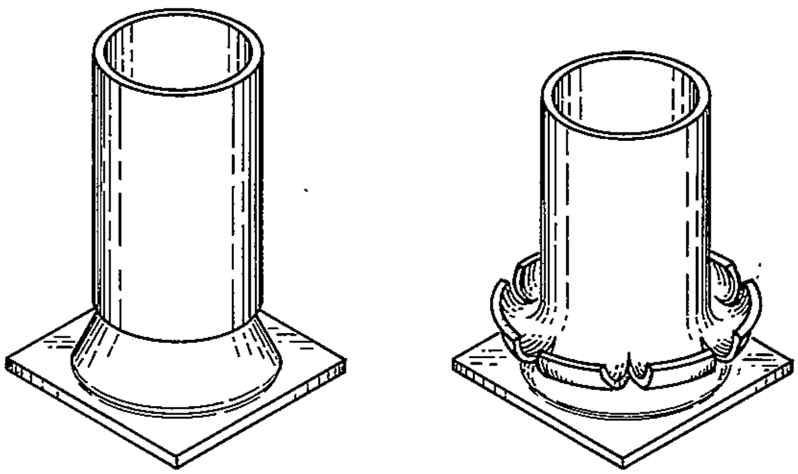


Figure 4. 1 Tube and Mandrel

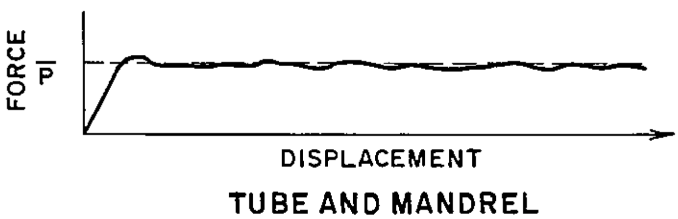


Figure 4. 2 Force-Displacement Curves

1. Analysis of Tube and Mandrel Device

What follows will be a very simple analysis of a tube and mandrel type energy absorbing device. It will be assumed that a brittle material is used so that the cracks in the tube are formed very near the top of the mandrel. It is further assumed that the strips roll up during crushing of the device.

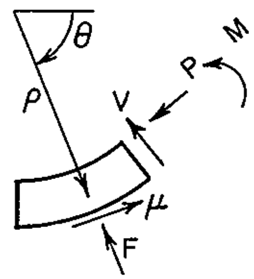


Figure 4.1.1

Equilibrium of an infinitesimal element results in two equations

$$\mu F ds = -dP + V d\theta \qquad (4.1.1)$$

$$F ds = dV + P d\theta \qquad (4.1.2)$$

The meaning of the variables is indicated by Figure 4.1.1.

Considering the independent variable θ, (4.1.1) and (4.1.2) can be combined to give

$$\frac{dP}{d\theta} + \mu P - \frac{1}{\rho}\frac{dM}{d\theta} + \frac{\mu}{\rho}\frac{d^2M}{d\theta^2} = 0 \qquad (4.1.3)$$

If M is assumed constant then the equation for P becomes the well known pulley equation whose solution is

$$P = P_0\, e^{-\mu\theta} \qquad (4.1.4)$$

The energy absorbed by friction can thus be determined from (4. 1. 1)

$$E_f = \int \mu F ds = -\int_0^{a} \frac{dP}{d\theta} \, d\theta \qquad (4.1.5)$$

Integrating

$$E_f = P_0 \left[1 - e^{-\mu a} \right]$$

The total energy absorbed per unit circumference is given by

$$E = P_0 \left(1 - e^{-\mu a}\right)/\mu + M_0/\rho + 2\sigma_0 t \, \epsilon_R/3 \qquad (4.1.7)$$

The second term in the expression represents bending energy and the third terms represents energy due to fracturing of the metal. The total force is thus given by

$$P_T = \pi D \, [M_0/\rho + \sigma_0 \, 2t \epsilon_R/3] \, e^{\mu a} \qquad (4.1.8)$$

2. Conical Mandrel

a. No Base

For this case as shown in Figure 4. 2. 1 there is no base and the curls are in a sense free to form an arbitrary radius. Thus unlike the other types of mandrels the radius cannot be determined merely from the geometry of the mandrel.

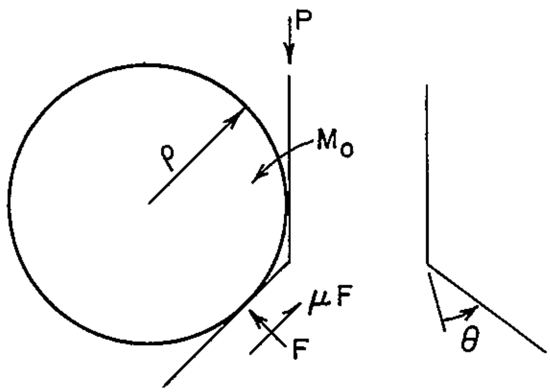


Figure 4. 2. 1'

Vertical equilibrium gives

$$P = \mu F \cos\theta + F \sin\theta \tag{4.2.1}$$

Moment equilibrium gives

$$M_0 = F\rho \cot\theta/2 \tag{4.2.2}$$

Combining (4. 2. 1) and (4. 2. 2) and including the energy due to fracturing of the metal results in an equation for the total load.

$$P_T = (\pi D\, \sigma_0\, t^2/4\rho) \tan\theta/2\ (\sin\theta + \mu\cos\theta) + 2/3\ \pi D\, \sigma_0\, \epsilon_R \tag{4.2.3}$$

It is still necessary however to determine the curl radius ρ. This will be a function of the cone angle and the material properties of the tube. It has been observed that stronger and thicker tubes produce larger curls for the same cone angle. Consider the tube in the material to follow a stress strain law given by a power expression of the form

$$\sigma = \sigma_0 \left(\frac{\epsilon E}{\sigma_0}\right)^{1/n} \tag{4.2.4}$$

n = positive integer

The moment therefore is given by

$$M = \int_{-t/2}^{t/2} \sigma_0 \left(\frac{\epsilon E}{\sigma_0}\right)^{1/n} y\, dy \tag{4.2.5}$$

But

$$\epsilon = y/\rho \tag{4.2.6}$$

thus

$$M = \sigma_0^{1 - 1/n} \left(\frac{E}{\rho}\right)^{1/n} \int y^{1/n + 1}\, dy \tag{4.2.7}$$

integrating

$$M = 2\sigma_0^{1 - 1/n} \left[\left(\frac{E}{\rho}\right)^{1/n} \Big/ (1/n + 2)\right] (t/2)^{1/n + 2} \tag{4.2.8}$$

From (4.2.3) the force per unit circumference is given by an equation of the form

$$P = M/\rho + 2/3\ \sigma_0\, \epsilon_R \tag{4.2.9}$$

Now the curl radius will be that one which minimizes the load. Therefore taking the derivative of (4.2.9) with respect to ρ and setting equal to zero produces

$$\frac{dP}{d\rho} = \frac{dM}{d\rho} - \frac{M}{\rho} \tag{4.2.10}$$

thus

$$\frac{dM}{d\rho} - \frac{M}{\dot{\rho}} = 0 \tag{4.2.11}$$

integrating

$$M = \rho/C(\theta) \tag{4.2.12}$$

where C (θ) is an arbitrary function of θ. Equating (4.2.12) and (4.2.8) produces the desired equation for ρ.

$$\rho^{1/n+1} = 2\sigma_0^{1-1/n}\left(E^{1/n}/(1/n+2)\right)(t/2)^{1/n+2}\,C(\theta) \tag{4.2.13}$$

The function C will have to be determined experimentally. As n approaches infinity the stress strain curve approaches the rigid plastic situation.

Thus

as $n \rightarrow \infty$

$$\rho = \sigma_0\, t^2\, C(\theta)/4 \tag{4.2.14}$$

A series of experiments were run to determine the function C(θ) giving a final equation for ρ.

$$\rho = \sigma_0\, t^2\, (.0368 - .029\,\theta)/4 \tag{4.2.15}$$

As expected the curl radius increases with the yield point and the thickness of the material. The total load equation is now calculated to be

$$P_T = \pi D \tan\theta/2\ (\sin\theta + \mu\cos\theta)/(.0368 - .029\,\theta) + 2\sigma_0\,\pi D\, t\, \epsilon_R/3 \tag{4.2.16}$$

b. <u>No Base and Ductile Tube Material</u>

For a ductile material the analysis will have to be changed somewhat since now the crack tip will form lower down on the tube.

In what follows it will be assumed that the point of rupture occurs at the point where the crack tip is located. If this be the case the curl radius is easily calculated to be

$$\rho = r_i \epsilon_R / \sin\theta \tan\theta/2 \tag{4.2.17}$$

where

r_i = initial tube radius

ϵ_R = rupture strain

From vertical equilibrium we find

$$P = (1 + \epsilon_R)\, F\, (\sin\theta + \mu\cos\theta) \tag{4.2.18}$$

F is the force normal to the deformed tube at the crack tip. To determine F we need to write an equilibrium equation for shear along the tube. Thus

$$\cos\theta \frac{dV}{d\theta} = \rho\, N_0 t/r_0 + \rho\,(1 - \cos\theta) \tag{4.2.19}$$

The tube has been assumed to behave rigid perfectly plastically. Since θ is, in general, small we will retain only second order terms in (4.2.19). The boundary conditions on (4.2.19) are

$$\begin{aligned} V(0) &= 0 \\ V(\theta) &= F \end{aligned} \tag{4.2.20}$$

Integrating (4.2.19) and using (4.2.20) leads to an equation for F.

$$F = \frac{\sqrt{2}}{2\sqrt{r_i (r_i - \rho)}} \log \left(\frac{r_i + \theta\sqrt{r_i (r_i - \rho)/2}}{r_i - \theta\sqrt{r_i (r_i - \rho)/2}} \right) N_0 t \rho \tag{4.2.21}$$

The total load equation becomes as a consequence

$$P_T = \frac{(1 + \epsilon_R)(\sin\theta + \mu\cos\theta)}{2\sqrt{r_i (r_i - \rho)/2}} \log \frac{r_i + \theta\sqrt{r_i (r_i - \rho)/2}}{r_i - \theta\sqrt{r_i (r_i - \rho)/2}} N_0 t \rho \pi D \tag{4.2.22}$$

where ρ is given by (4.2.17),

$N_0 = \sigma_y t$

The above equation is somewhat cumbersome to use. In addition the equation for the curl radius (4.2.17) indicates that the curl radius increases in proportion to the rupture strain. Experimental results indicate that this is not the case. Consequently an empirical relation is desired. To be dimensionally correct an acceptable form would be

$$P = \sigma_0 t D f(\theta) \tag{4.2.23}$$

where f is an unknown function of the cone angle θ. The results of a series of experiments indicate that an approximate equation for the total load may be given by

$$P = \sigma_0 t D \left[0.3 + \pi\theta \times 10^{-3}\right] \tag{4.2.24}$$

where θ is measured in degrees from the vertical.

3. Conical Mandrel With Base

This mandrel operates in the manner shown in Figure 4.3.1. The base plate now gives additional bending to the tube. Note that if the dimension 'ℓ' is large enough the curls don't touch the base and the device operates as the simple conical mandrel.

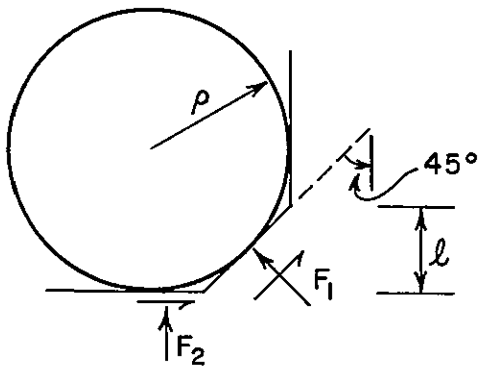


Figure 4.3.1

From purely geometrical consideration

$$\rho = \ell/\sqrt{2}\tan\pi/4 = 1.7\ell \tag{4.3.1}$$

Taking moments about 0 gives

$$P = \mu[F_1 + F_2] + M_0/\rho \tag{4.3.2}$$

Vertical equilibrium gives

$$F_1 = \frac{P - F_2}{\sqrt{2}\,(1+\mu)} \tag{4.3.3}$$

$$P = F_2\mu\left(\frac{\sqrt{2}\,(1+\mu) - 1}{\sqrt{2}\,(1+\mu) - \mu}\right) + \frac{M_0}{\rho}\,\frac{\sqrt{2}\,(1+\mu)}{\sqrt{2}\,(1+\mu) - \mu} \tag{4.3.4}$$

But

$$F_2 = \frac{M_1}{\rho}\left(\frac{\sqrt{2}}{1 + \mu(1 - \sqrt{2})}\right) \tag{4.3.5}$$

Therefore

$$P_T = \left[\frac{M_1}{\rho}\left(\frac{\sqrt{2}}{1 + \mu(1 - \sqrt{2})}\right)\left(\frac{\mu - a}{1 - a}\right) + \frac{M_0}{\rho(1 - a)}\right]\pi D \tag{4.3.6}$$

where

$$\rho = 1.7\ell$$

$$a = \mu/\sqrt{2}\,(1 + \mu)$$

Now M_1 is somewhat arbitrary but as an approximation we may take

$$M_1 \approx M_0$$

4. Experimental Results

a. Conical Mandrel No Base

A series of tests were performed on 2024-T3 aluminum of 1" diameter and .035" wall thickness. The cone angle was varied in increments of 15°. Figure 4.4.1 shows a comparison of these tests with equation (4.2.16). The correlation is seen to be good.

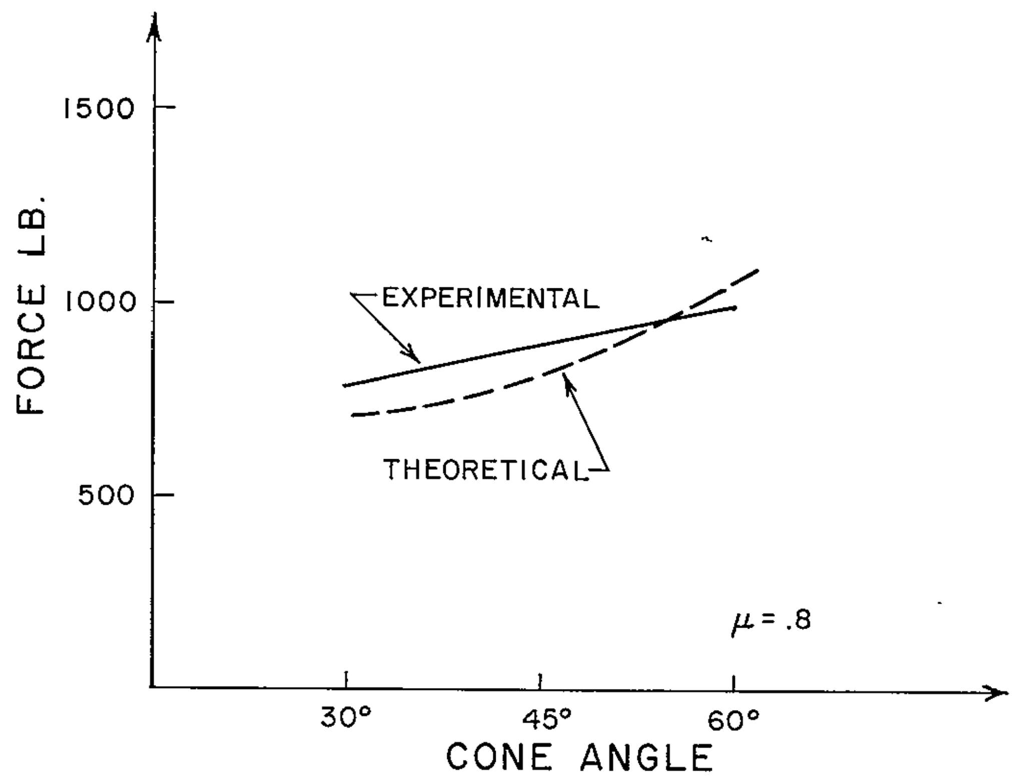


Figure 4.4.1

b. Conical Mandrel With Base

A series of tests were performed on a mandrel with a base where l was varied. The material used was 2024-T3 aluminum of 1" diameter and .065" wall thickness. Figure 4.4.2 shows comparison of equation (4.3.1) with experiments. Note that the slopes of the two curves are almost identical.

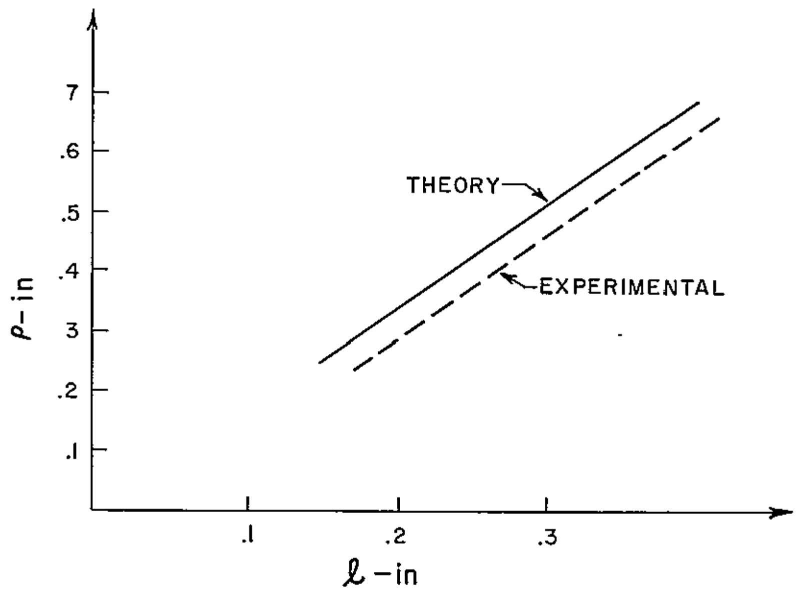


Figure 4.4.2

Figure 4.4.3 shows a comparison for the total load, experimental and theoretical.

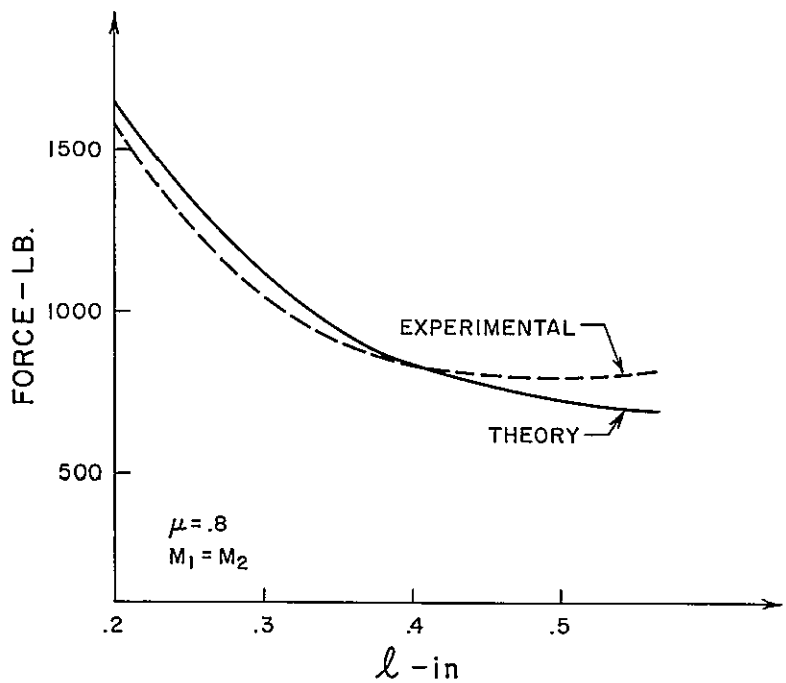


Figure 4.4.3

V. BASIC RESEARCH ON THE FOLDING TUBE DEVICE

A. Basic Research on the Folding Tube Device

The folding tube, Figure 5.1, is simply a straight thin walled tube. Under the application of a sufficiently large axial load the tube forms circular or polygonal folds as it shortens. Energy is absorbed by the plastic work alone in forming the folds.

A typical force-displacement curve is shown in Figure 5.2. There is an initial peak force P_{max} followed by a regular fluctuation in the force level about some average value $\overline{P}$. Each cycle corresponds to the formation of a single fold. The force level is not constant because of the change in geometry which occurs during the folding process.

Research to date has concentrated on the problem of determining $\overline{P}$ for a given tube geometry and material. The primary parameters governing the operating force level have been found to be the yield strength of the material, the diameter and wall thickness of the tube, and the work hardening characteristics of the material. A secondary parameter is the axial length of the tube.

To determine the effect of these parameters on $\overline{P}$ a basic experimental program was begun. The program involves the testing of aluminum and steel seamless tubing of different lengths, wall thickness, and tube diameter and the tensile testing of the alloys being used to determine the stress-strain curve of each alloy. The detailed stress-strain curves are necessary in order to predict the onset of plastic column instability and to determine the actual yield strength and work hardening characteristics of the material (only minimum values of yield strength, tensile strength, and elongation are generally available in the literature).

B. Stress-Strain Curves

The tensile stress-strain curve of a seamless tube differs from the stress-strain curve obtained from a standard cylindrical tensile test specimen of the same material. Therefore, special test methods were devised to enable us to conduct tensile tests using seamless tubing. The tubes were held at either end by the slightly oversize knurled cylinders shown in Figure 5.3. The tubes were installed by first warming them to 300-500° F and then slipping them over the knurls. Semi-circular

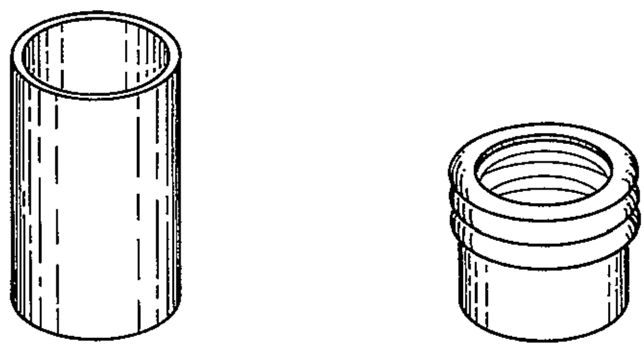


Figure 5. 1. Folding Tube

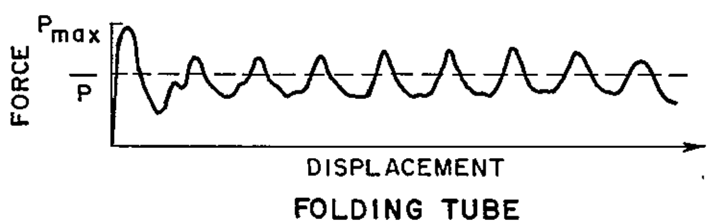


Figure 5. 2. Typical Force-Displacement Curves

clamps were then placed over the tube ends and tightened with bolts. As the tube was pulled in tension the reduction in diameter produced a sufficient increase in the friction force to prevent slippage. As a check on the procedure, one tube was strain gaged at its midpoint. The strains determined by measuring the change in total length of the tube and computing $\Delta L/L$ were in agreement with the strain gage readings. A typical force-extension curve obtained in this manner is shown in Figure 5.4 for 6061-T4 aluminum. The stress-strain curve is obtained by dividing the force by the cross-sectional area of the tube 0.11 inch^2 and the extension by the total free length 8.25 inch.

C. Folding Tube Experiments

The experimental program has been partially completed. Tests have been conducted with one inch diameter tubing of 0.035 inch, 0.049 inch, and 0.065 inch wall thicknesses and with two inch diameter tubing of 0.035 inch wall thickness. The tube materials were 2024-0, 3003-H14, 5052-0, 6061-0, 6061-T4, and 6061-T6 aluminums and 1015, 4130, and 5050 steels. 2024-T3 aluminum tubing was also used but the material is too brittle to allow the formation of complete folds. The alloy split, cracked, and chipped after the partial formation of a single fold.

The results of the experiments are summarized in Table 5.1. There are no results given for those tubes which experienced column instability.

An empirical relationship between the average folding force and the primary parameters has been developed from the experimental results. It is

$$\overline{P} = 8(\sigma_y + .25H)\, t^{3/2}\, D^{1/2} \tag{5.1}$$

where σ_y is the yield strength of the material, H is the average slope of the stress-strain curve in the plastic region, t is the wall thickness and D is the outer diameter of the tube. The relevant material properties are given in Table 5.2.

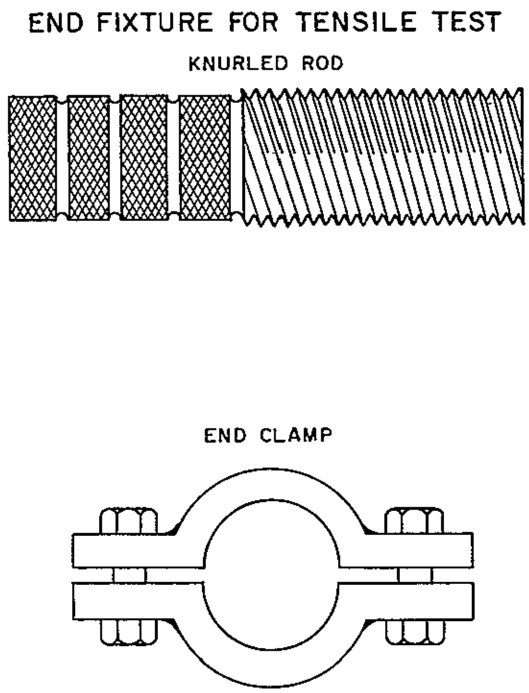


Figure 5.3

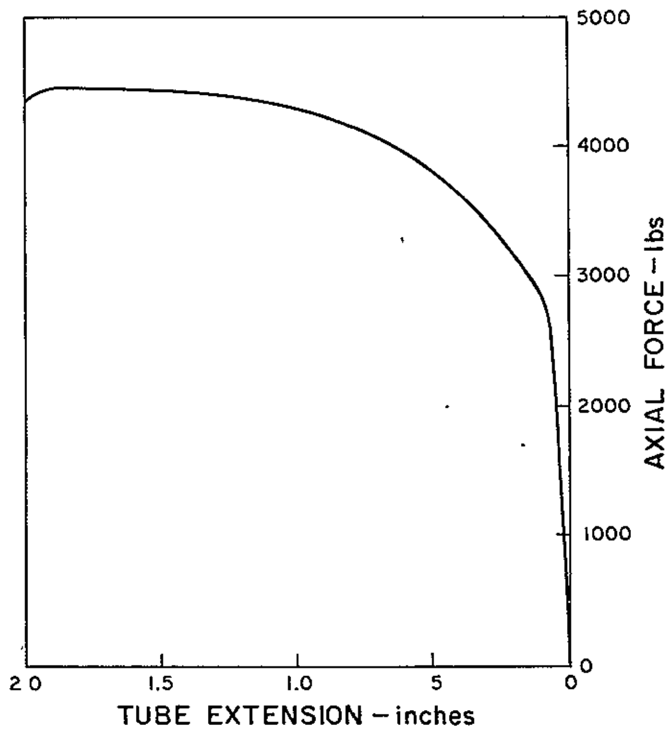


Figure 5.4. Typical Force-Extension Curve for 6064-T4 Aluminum Tubing

TABLE 5.1

Compilation of Data

Alloy	D	t	P_{max}	$\overline{P}$
4130	1	.035	12,240	6,100
	1	.035	12,350	6,200
	1	.035	12,325	6,000
	1	.035	12,060	6,500
	1	.035	12,060	6,400
	1	.065	23,000	17,000
	1	.049	16,300	11,500
1015	1	.035	7,800	4,500
	1	.035	8,000	4,500
	1	.035	8,000	4,400
	1	.065	15,900	11,500
	1	.049	12,300	7,500
	2	.035	17,000	5,100
5050	1	.035	5,200	3,400
	1	.065	10,700	7,000
	1	.049	9,000	5,500
5052-0	1	.035	2,430	1,460
	1	.035	2,420	1,660
	1	.035	2,450	1,500
	1	.049	3,810	2,700
	1	.065	5,140	3,400
	2	.035	3,880	1,700
3003-H14	1	.035	2,270	1,200
	1	.035	2,250	1,200
	1	.035	2,290	1,150
	1	.049	3,365	2,200
	1	.065	4,780	3,800
	2	.035	4,790	1,800
6061-T6	1	.035	4,640	2,400
	1	.035	4,625	2,400
	1	035	4,640	2,100
	1	.049	6,970	4,050
	1	.065	8,440	5,200
	2	.035	9,050	3,000
2024-0	1	.035	2,680	1,600
	1	.035	2,680	1,400
	1	.035	2,650	1,500
	1	.049	4,305	2,500
	1	.065	6,970	4,200
6061-0	1	035	1,380	900
	1	.035	1,345	900
	1	.065	3,480	2,400

TABLE 5.2

Material Properties

Material	σ_y	H
2024-0	11,000	70,000
3003-H14	21,000	7,000
5052-0	13,000	50,000
6061-0	8,000	33,000
6061-T4	21,000	55,000
6061-T6	40,000	30,000
1015	45,000	60,000
4130	75,000	80,000
5050	31,000	60,000

VI. TRANSPORT TRAILER SUPPORT APPLICATION

As a result of a visit to the Fruehauf Corporation, Detroit, Michigan, a potential application for the tube and mandrel energy absorber was found in the forward support structure on transport trailers. This support structure (Figure 6.1) with the wheels or pads lowered, is used to hold up the front of the trailer when it is not hooked up to a tractor. When hooked to the tractor, the front of the trailer is supported by a plate on the rear of the tractor known as the "fifth wheel" and the front support wheels or pads are retracted as shown in Figure 6.2. Occasionally, when disconnecting the trailer, the driver will drive off without having completely lowered the front support. This can result in severe damage to the support structure and sometimes to the trailer structure. It was believed that a commercially feasible tube and mandrel energy absorbing device could be designed for incorporation in the front support structure to absorb the energy of the falling trailer and protect the support and the trailer from damage.

1. Requirements

a. Mathematical Model

To design the energy absorber it was necessary to determine the kinetic energy of the trailer at the time the support wheels impact the pavement so that the energy absorber requirements could be established. The analysis was made using the following assumptions:

1. The clearance between the support wheels and the pavement at the time the tractor pulls out from the trailer is 6".
2. The fifth wheel remains horizontal or parallel with the underside of the trailer until the pivot moves to a position in front of the front corner of the trailer at which time the fifth wheel rotates downward to form a ramp on which the front corner of the trailer slides down.
3. The fifth wheel and the ramps on the tractor frame provide sufficient length for the trailer corner to slide down far enough to allow the support wheels to contact the pavement.
4. The underside of the trailer does not contact any of the tractor parts other than the fifth wheel and the ramps on the tractor frame.

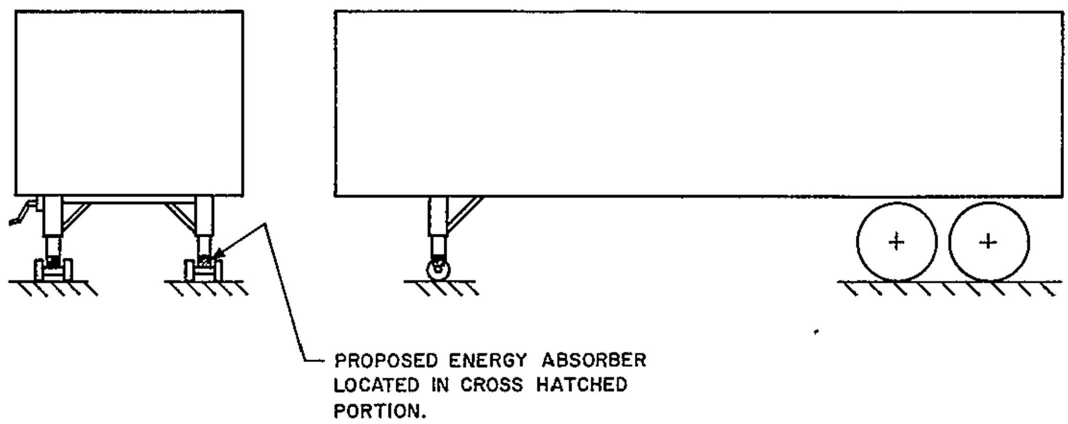


Figure 6.1 Sketch of Transport Trailer Showing a Typical Support Structure

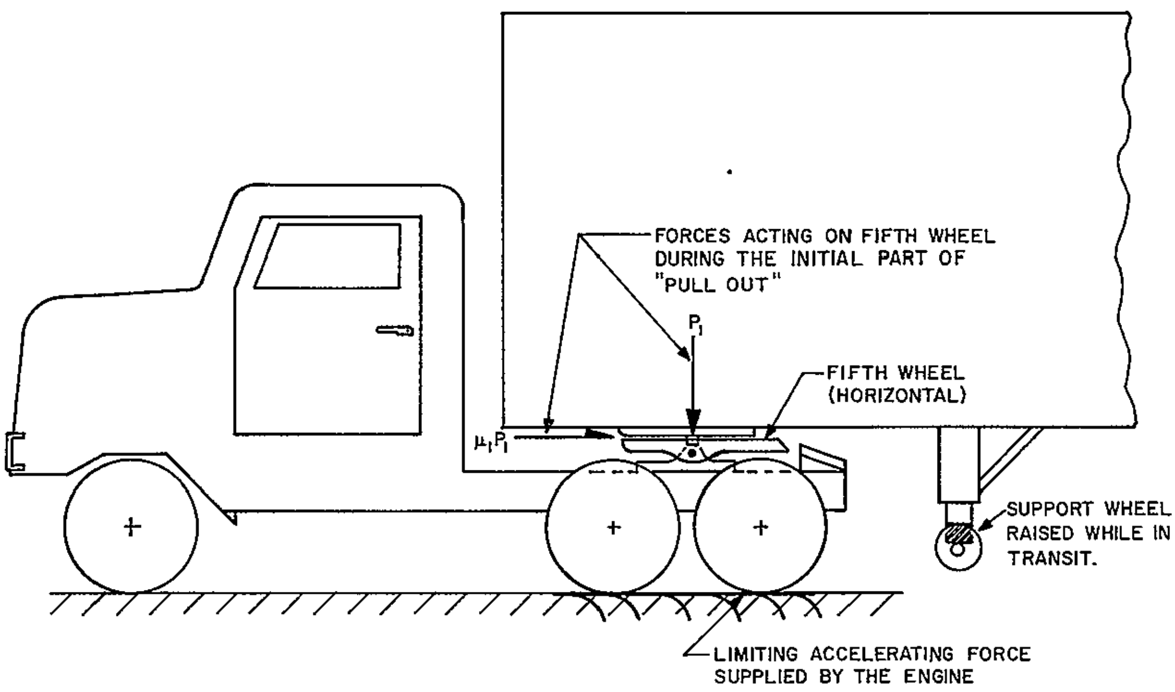


Figure 6.2 Sketch Showing the Tractor-Trailer Hook-up and Forces Between the Trailer and the Tractor Fifth Wheel During the Initial Phase of the Pull-Out

5. The tractor applies a constant torque to the rear wheels during the pull-out.

It was observed in an examination of several tractors that there are a wide variety of tractor frame and fifth wheel configurations and these could result in a different sequence of events in the drop process. For instance, some tractors are so constructed that the underside of the trailer would contact the top of the tractor tires at some point if the tractor was pulled out with the trailer support wheels retracted. In other configurations the front corner of the trailer would impact the rear part of the tractor frame. The mathematical analysis presented here is valid only for the case defined by the assumptions given above.

Initially the tractor of mass M_t is accelerated forward, away from the trailer, by the tractive force F_t resisted by the sum of the friction forces F_f. The acceleration, A_{t1}, of the tractor during this initial phase of the pull-out is given by

$$A_{t1} = \frac{F_t - F_f}{M_t} \tag{6.1}$$

The friction in the tractor wheel bearings and the rolling friction between the tires and the pavement are small compared to the friction between the trailer and the fifth wheel and are, therefore, neglected. The friction term in Equation 6.1 can then be approximated by

$$F_f = \mu_1 P_1 \tag{6.2}$$

where μ_1 is the coefficient of friction between the fifth wheel and the trailer and P_1 is the downward force of the trailer on the fifth wheel. F_t is assumed to be constant during the pull-out and is computed from the maximum engine torque T, the gear ratio of the power train G, the overall efficiency of the power train e, and the radius of the tractor wheels according to

$$F_t = \frac{T\,G\,e}{r} \tag{6.3}$$

We have assumed that the rear wheels of the tractor do not slip when this torque is applied. This is generally a valid assumption since the typical tractor is capable of producing only about one-half the torque at the rear wheels which would be necessary to slip the wheels under the weight of the fully loaded trailer. Using Equation 6.2 in equation 6.1 we have

$$A_{t1} = \frac{F_t - \mu_1 P_1}{M_t} \tag{6.4}$$

The fifth wheel is pivoted in the center to allow relative movement between the tractor and the trailer while operating over the road. This pivot will generally allow the fifth wheel to tilt downward at the rear approximately 18.5°. When the tractor pulls out from a trailer which has its support partially retracted the fifth wheel will remain parallel with the underside of the trailer until the tractor has advanced sufficiently to put the pivot point in front of the front corner of the trailer (this distance is typically 2.3 ft) then the fifth wheel will tilt downward at the rear to form a ramp for the front corner to slide down. The force of the trailer on the tilted fifth wheel produces a component parallel to the frame of the tractor which helps to accelerate it outward. The system of forces on the tilted fifth wheel is shown in Figure 6.3.

When the front corner slides down the tilted fifth wheel the body of the trailer rotates about a pivot point between the axles (see Figure 6.4). The downward force P is a function of the angular acceleration α of the trailer. The acceleration A_{t2} of the tractor during this second phase of the pull-out is given by

$$A_{t2} = \frac{F_t + P \tan \theta - \mu_1}{M_t} \tag{6.5}$$

where θ is the angle of tilt of the fifth wheel. The angular acceleration of the trailer about its pivot (see Figure 6.4) is given by

$$\alpha = \frac{Mg R_c - \mu_1 P R_L \tan\theta - P R_L - Pa \tan\theta + \mu_1 Pa}{M R_c^2} \tag{6.6}$$

This equation may be simplified somewhat by observing that the last two terms in the numerator are quite small compared with the others and can be neglected; we write

$$\alpha = \frac{Mg R_c - \mu_1 P R_L \tan\theta - P R_L}{M R_c^2} \tag{6.7}$$

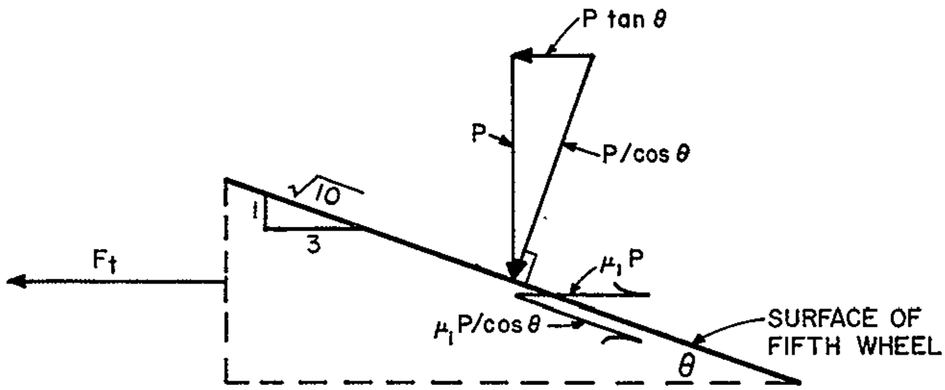


Figure 6.3 Free Body Diagram Showing Forces Acting on the Fifth Wheel After it has Rotated

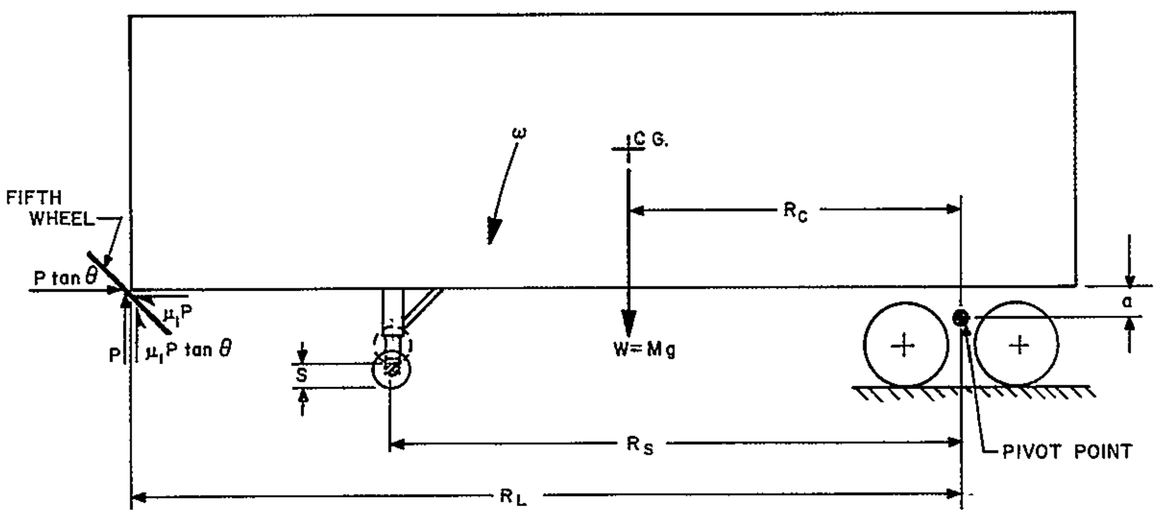


Figure 6.4 Sketch of Trailer Showing the Forces Acting at the Interface Between the Fifth Wheel and the Trailer and the Important Trailer Dimensions

Also since the angle through which the trailer rotates is small and since the ratio of the horizontal travel of the fifth wheel to the vertical travel of the front corner of the trailer is 3/1 the angular acceleration of the trailer is related to the acceleration of the tractor by

$$\alpha = \frac{A_{t2}}{3R_L} \qquad (6.8)$$

Equations 6.5, 6.7 and 6.8 may be solved simultaneously to eliminate P. The acceleration of the tractor may then be written as

$$A_{t2} = \frac{3R_L\,(W R_c\,\beta + R_L F_t \gamma)}{M R_c^2\,\beta + 3R_L^2 M_t\,\gamma} \qquad (6.9)$$

where

$$\beta = \operatorname{Tan}\theta - \mu_1$$

$$\gamma = \mu_1 \operatorname{Tan}\theta + 1$$

The velocity of the tractor, V_1, at the time the fifth wheel tilts is

$$V_1 = \sqrt{2A_{t1} S_1} \qquad (6.10)$$

where S_1 was given previously as 2.3 ft. The velocity of the tractor at the time the support wheels impact the pavement, V_2, is given by

$$V_2 = \sqrt{V_1^2 + 2A_{t2} S_2} \qquad (6.11)$$

where S_2 is the distance traveled by the tractor while the front corner of the trailer slides down the tilted fifth wheel (typically 2 ft). Since the slope of the fifth wheel is 1/3, the downward velocity V of the front corner of the trailer at the instant the support wheels impact the pavement is

$$V = V_2/3 \qquad (6.12)$$

The instantaneous angular velocity ω_1 which corresponds to V is

$$\omega_1 = V/R_L = V_2/3R_L \qquad (6.13)$$

After the support wheels impact the pavement, the energy absorbers apply a nearly constant stopping force over their stroke which has been taken as 2 inches. If we let α_s be the angular acceleration of the trailer while the energy absorbers are stopping its rotation and ϕ be the angle of rotation of the trailer about its pivot point, the acceleration may be expressed as

$$\alpha_s = \frac{d\omega}{dt} = \frac{d\omega}{d\phi}\frac{d\phi}{dt} = \omega\frac{d\omega}{d\phi} \tag{6.14}$$

but also,

$$\alpha_s = \frac{\Sigma T}{I} \tag{6.15}$$

where ΣT is the sum of the torques on the trailer and I is the moment of inertia about the pivot point. Thus the motion of the trailer during the operation of the energy absorbers can be described by

$$\omega\, d\omega = \left[\frac{F R_s - Mg R_c}{M R_c^2}\right] d\phi \tag{6.16}$$

Integration of this expression from $\omega = \omega_1$ (Equation 6. 13) to $\omega = 0$ on the left side and from $\phi = S/R_s$ to $\phi = 0$ on the right ($\phi = \tan\phi$ since the angle is small, and S is the stroke of the energy absorber) gives

$$\frac{\omega_1^2}{2} = \left[\frac{F R_s - Mg R_c}{M R_c^2}\right] \frac{S}{R_s} \tag{6.17}$$

where F is the average force required to operate the two energy absorbers. Equation 17 may be solved for F giving

$$F = \frac{\omega_1^2 M R_c^2}{2S} + \frac{Mg R_c}{R_s} \tag{6.18}$$

Using the data given in Table 1, F = 74, 700 lb. This would require an operating force of 37, 350 lb for the energy absorber on each leg of the support. Thus, in this example an energy absorber incorporated into each leg of the support structure should operate at a steady load of 37, 350 lb over a stroke of 2 inches in order to bring the trailer to rest without damaging the support structure of the structure of the trailer.

TABLE 6.1

Data for One Tractor-Trailer Combination

Engine Horsepower		275 bhp @ 2300 rpm
Max. Engine Torque	T_w	700 ft/lb @ 1600 rpm
Max. Torque Rating on Clutch	T_c	1200 ft/lb
Truck Weight	W_t	13,105·lb
Rear Wheel Radius	r	43.5 in (neglecting tire deflection)
Max. Gear Reduction	G	60:1
Assumed Coefficient of Friction for Trailer on Fifth Wheel, Typical	μ_1	.1
Assumed Gear Train Efficiency	e	65%
Loaded Trailer Weight	W	57,500 lb
Trailer Dimensions	R_c	218 in.
	R_s	360 in.
	R_L	487 in.
Pull-out Distances	S_1	2.3 ft
	S_2	2 ft

b. Static Load Requirements

In addition to the dynamic load requirements just discussed it is also necessary to consider the requirements, the tube and mandrel must meet while the trailer is in every day service. That is, the energy absorbing device must, for the greatest portion of the life of the trailer, function merely as a structural member supporting the dead weight of the loaded or unloaded trailer, up to 45,000 lbs, (see Figure 6.5 for specifications) while it is not being pulled. In this connection, it may also happen that the support wheels will become imbedded in the surface on which they are resting by freezing or sinking into soft ground or asphalt. When this happens, considerable force may be applied to the wheels and legs of the structure by the gear mechanism when the driver attempts to crank the wheels up. Consequently, the tube and mandrel must be able to operate in tension up to a load of approximately 10,000 lbs.

An additional load which the support structure is expected to withstand is a side load, generated when a load is put on the side of the trailer with the support wheels lowered. This creates a frictional resistance load at the interface of the support wheels and the pavement. This load has, in testing a standard trailer, reached values of 11,000 lbs per leg without damage. It is assumed that the tube and mandrel would be required to meet the same standards when incorporated into the support legs.

c. Operational Requirements

When a trailer is accidentally dropped it is desirable to be able to determine, easily, whether or not the energy absorber has been actuated. Thus, the energy absorbers must be located where they can be inspected visually both by the driver and by maintenance personnel.

d. Manufacturing Requirements

The device must be economical to make, install, service and replace. It was therefore decided that the tube and mandrel device should be made of materials which are relatively inexpensive and easy to obtain. Also the configurations of the attachments and the mandrel should lend themselves to low cost mass production.

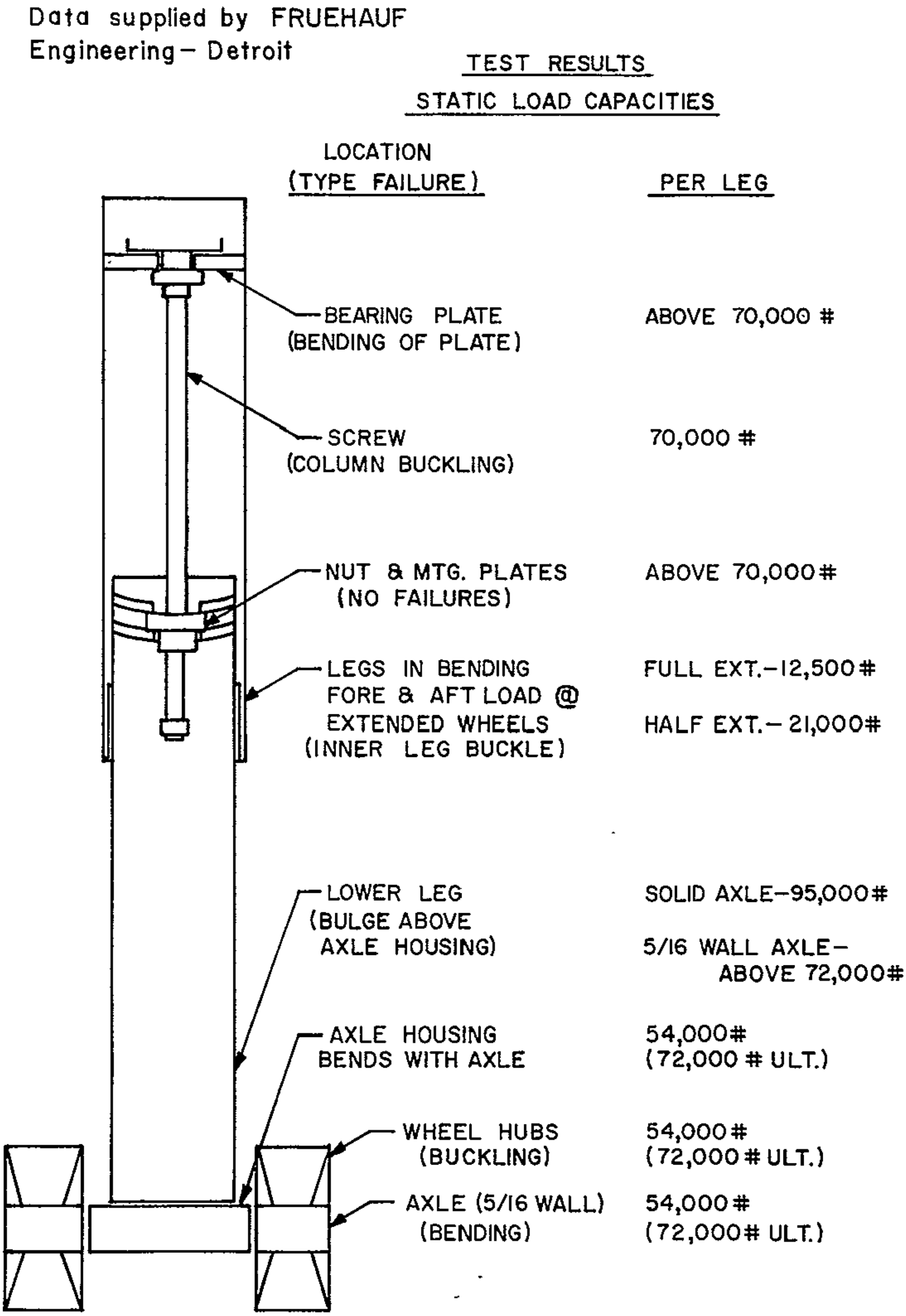


Figure 6.5 Test Specifications for Support Structure of the Fruehauf Trailer

2. Design

Bearing in mind the requirements outlined previously, a prototype tube and mandrel energy absorbing device was developed as shown in Figures 6.6 and 6.7. This device consists of a 3.25 in. I.D. mild steel tube with a 0.125 in. wall and a conical shaped mandrel made of mild steel. For a discussion of the use of conical mandrels see Section IV. The tube is beveled on the inside and notched in eight places on the end which is in contact with the mandrel. The bevel is required to control the shape of the load-deflection curve during the start and the notches are required to initiate splitting so that strip curls will form. The tube is welded to the mandrel on the inside so that the device will function as a structural element under normal loading. The quantity of weld used determines the force required to activate the energy absorber, since the weld is sheared when the device is activated. The load-displacement curve for this device is shown in Figure 6.8. The peak force is 50,000 lbs followed by a relatively steady load of 35,000 to 40,000 lbs.

The device operates as follows. When the support wheels strike the ground, the decelerating force, acting upward through the wheels, is transmitted to the trailer through the tube and mandrel. As long as the force does not exceed the maximum static design load of 50,000 lbs in each leg, the device will not operate and the trailer will be brought to rest as though the energy absorber were simply a rigid part of the support. This might occur, for example, if the trailer were dropped through a distance of only 1 or 2 inches. However, if the drop is from a sufficient height, the device will operate when the retarding force reaches 50,000 lbs. When this happens, the welds holding the tube to the mandrel shear off and the tube moves down over the conical mandrel. As the tube continues down, the steady state force of 35,000 to 40,000 lbs is reached. This retarding load continues until sufficient stroke of the device has been used to absorb the energy in the trailer.

The device is designed for a bolted connection to the bottom of the support leg as shown in Figure 6.9. This design makes the replacement of the device a simple matter. The retainer strap shown in the figure is required so that the wheels will not be lost from the trailer after the energy absorbers are activated since the operation of the device shears the welds between the tubes and the mandrels allowing the end of the support attached to the wheels to be free when the trailer is raised.

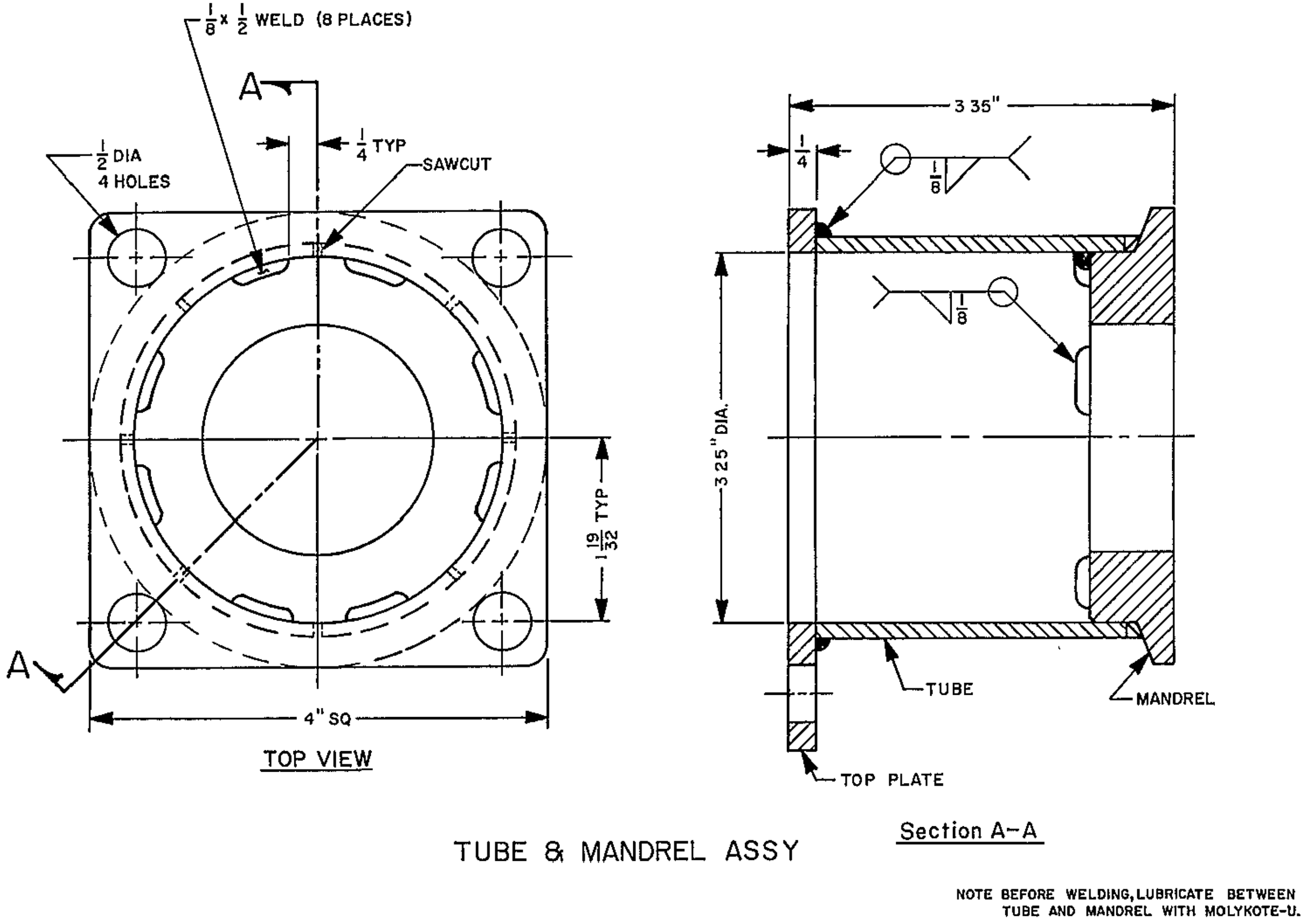


Figure 6.6 Prototype Tube and Mandrel Energy Absorbing Device Developed for use on Semi-Trailer Front Support Structure. See Figure 6.7 Also

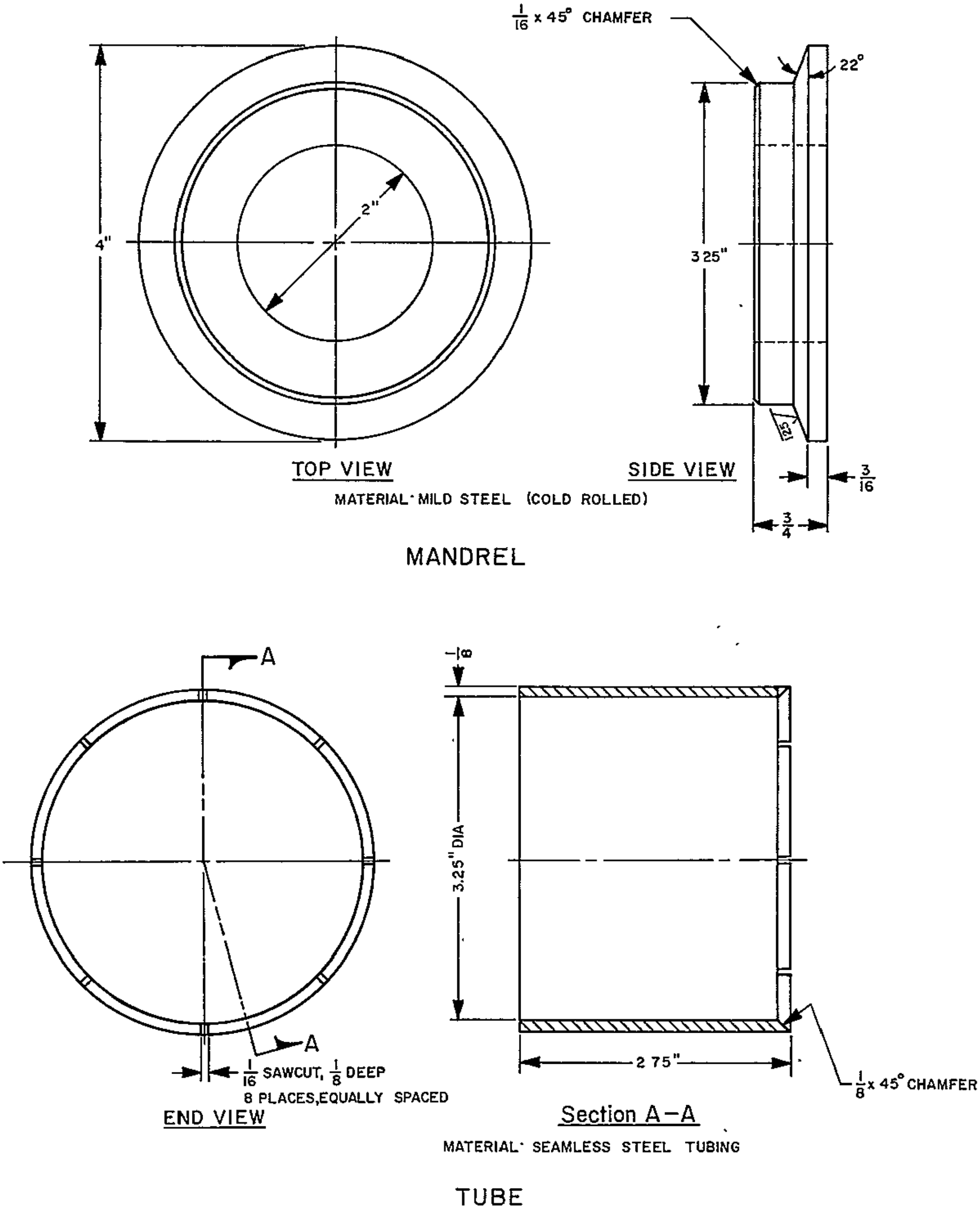


Figure 6.7 Details for Protype Energy Absorbing Device Shown in Figure 6.6

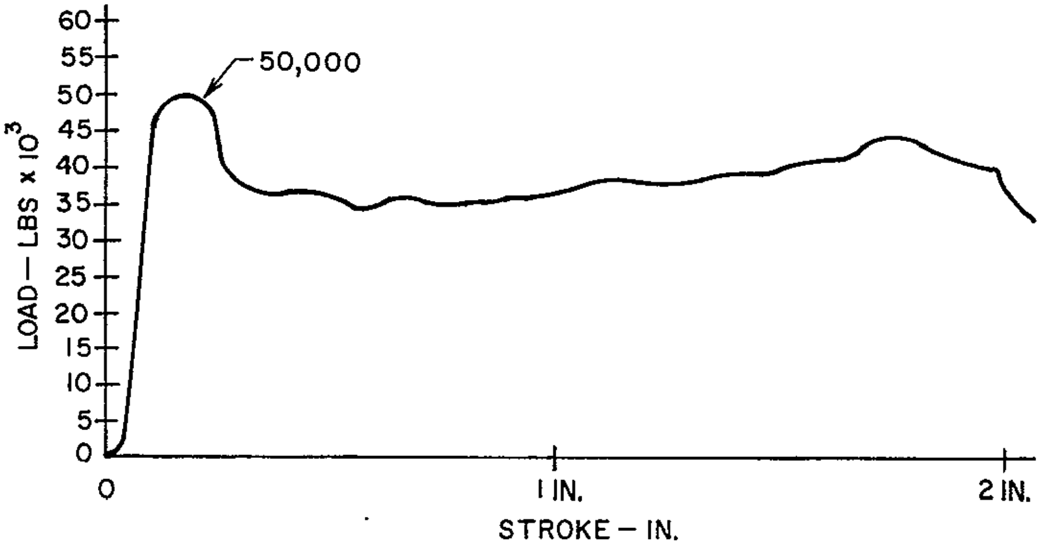


Figure 6.8 Load-Displacement Curve for the First Prototype Tube and Mandrel Energy Absorber Proposed for use on the Fruehauf Trailer.

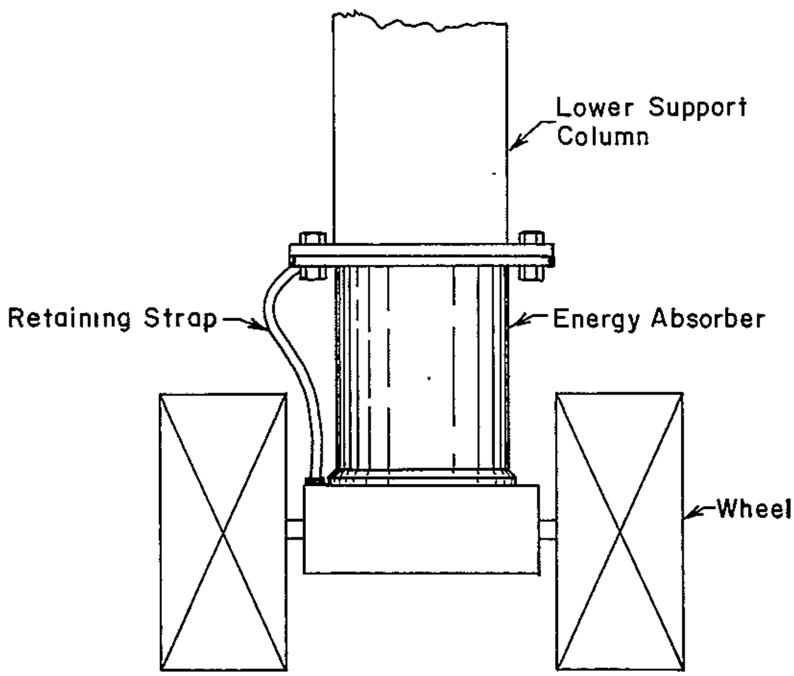


Figure 6.9 Sketch Showing Probable Mounting of Energy Absorber in Trailer Support Structure

3. Testing

The device has been subjected to many static tests and found to perform in a very repeatable fashion with a variation in peak and average loads on the order of ± 5%.

Drop testing and field testing of the device on Fruehauf trailers will be conducted to determine the performance of the device in actual drop conditions. The results of these tests will be evaluated before the design of the device is finalized.

4. Market Potential

The total number of trailers sold in the U.S. last year was 100,000. Fruehauf sold 30,000 of these and also sold 18,000 sets of support legs to other trailer manufacturers and 2500 sets of support legs as replacement parts.

VII. ELEVATOR SAFETY SYSTEM APPLICATION

A visit was made to the Otis Elevator Company, New York, to make a presentation of the work being done on energy absorbers, and to determine the possibility of using these devices to promote elevator safety. It was learned that elevators don't free-fall in the event of cable breakage as we had previously supposed, since they are equipped with safety brakes which operate against the elevator guide rails to prevent the elevator from moving faster than 15% over the design speed. However, an energy absorber is used to stop the elevator if it overshoots the bottom or top floor due to circuit malfunction, operator error, or cable breakage. The energy absorbers presently used are hydraulic, and are positioned under the elevator car for overshoot of the bottom floor and under the counterweight for overshoot of the top floor.

The cost for the present buffers used in the average elevator is $600.00 per buffer or $1,200.00 per elevator. A tube and mandrel device costing only a fraction of this amount can be designed for this use. Since these devices are only rarely activated, replacement of the tube and mandrel device does not present a serious problem. To avoid the possibility that the same device could be impacted more than once, an alarm can be designed to signal when the energy absorber has been activated and must be replaced.

1. Design Criteria

a. Operational Requirements

The specifications which the buffer must meet are defined in the "Safety Code for Elevators." The pertinent portion of this code used for oil buffers states:

> "Oil buffers shall develop an average retardation not in excess of 32.2 feet per second per second and shall develop no peak retardation greater than 80.5 feet per second per second having a duration exceeding one-twenty-fifth (1/25) of a second with any load in the car from rated load to a minimum load of one hundred and fifty (150) pounds when the buffer is struck with an initial speed of not more than one hundred fifteen (115) percent of rated speed for buffers conforming with Subdivision 1 of Rule 201.4a."

Rule 20.4a states,

"The minimum stroke of oil buffers shall be based on the following:

1. The stroke shall be such that the car or the counterweight on striking the buffer at one hundred and fifteen (115) percent of rated speed shall be brought to rest with an average retardation of not more than 32.2 feet per second."

Additional information from this code is contained in Table No. 201.4a and is reproduced below.

Table No. 201.4a from the "Safety Code for Elevators"

Rated Speed in Feet Per Minute	115% of Rated Speed in Feet Per Minute	Minimum Strokes of Oil Buffers in Inches
200	230	2 3/4
225	259	3 1/2
250	288	4 1/4
300	345	6 1/4
350	402	8 1/4
400	460	11
450	517	13 3/4
500	575	17
600	690	24 3/4
700	805	33 1/4
800	920	43 3/4
900	1035	55 1/2
1000	1150	68 1/2
1100	1265	83
1200	1380	98 1/2
1300	1495	115 1/2
1400	1610	134 1/2
1500	1725	154
1600	1840	175
1800	2070	222
2000	2300	274

Three commonly used elevators made by the Otis Elevator Company along with their operating characteristics are tabulated in Table 7.1. The strokes used by Otis in their production model oil buffers are identically the same as those given in the code except that each buffer is designed for a speed range and therefore uses the stroke which applies to the highest speed. Table 7.2 lists the strokes of the hydraulic buffers used by Otis for various speed ranges. It is assumed that the specifications given in the code for oil buffers will also be applicable to the tube and mandrel energy absorbing device.

b. Manufacturing Requirements

To make the tube and mandrel an attractive replacement for the presently used oil buffer it is necessary to make it as simple and

TABLE 7.1

Operating Data for Three Commonly Used Elevators
(Courtesy of Otis Elevator Co.)

Elevator Capacity (lbs)	Loaded Weight (lbs)	Empty Weight (lbs)	Rated Speed (ft/min)	Buffer Striking Speed (ft/min)	Max. Allow. Peak Retard. Force Loaded (lbs)	Max. Allow. Peak Retard. Force Empty Plus 150 lbs (lbs)	Energy Absorber Stroke (in.)
3500	10, 500	7, 000	350	402	26, 200	17, 500	11
3500	10, 500	7, 000	700	805	26, 200	17, 500	43-3/4
8000	24, 400	16, 400	600	690	61, 000	41, 000	33-1/2

TABLE 7.2

Strokes Used in Production of Hydraulic Buffers
(Courtesy of Otis Elevator Co.)

Buffer Stroke (in.)	Rated Speeds (ft/min)
8-1/4	Over 200 up to 350
11	Over 350 up to 400
17	Over 400 up to 500
27-3/4	Over 500 up to 600
33-1/2	Over 600 up to 700
43-3/4	Over 700 up to 800
68-1/2	Over 800 up to 1000
84	Over 1000 up to 1140

inexpensive as possible using materials which can be readily obtained. Since the proposed application is inherently suitable for the tube and mandrel these manufacturing requirements can be met using standard mild steel tubing both for the tube and the mandrel and using the conical mandrel shape as discussed in Section IV.

c. Load Requirements

1. General Considerations

A simple analysis of the energy absorber requirements for decelerating an elevator can be made neglecting wave mechanics. The kinetic energy of the elevator at impact may be equated to the work W required to split the tube and roll up the segments over a stroke length, S.

Thus,

$$W = FS \tag{7.1}$$

Where F is the force (assumed to be constant) required to operate the energy absorber.

Then

$$\frac{1}{2} MV^2 = FS \tag{7.2}$$

where M is the mass, and V is the impact velocity of the elevator. Restricting the deceleration of the elevator to 32.2 ft/sec or 1 G restricts the value of F to the weight of the unloaded elevator. For example, using the 3500 lb capacity elevator traveling at 115% of rated speed the stroke is computed to be S = 33-1/4 inches, which compares with the value given in the code. However, this is the minimum stroke for this elevator. The stroke required to satisfy the requirements of the code for both the unloaded elevator as well as the loaded elevator is found using the loaded weight while retaining the value of F = 7150 lbs. Thus, the required stroke is S = 51 inches. Note that the Otis Company uses a stroke of 68-1/2 inches for this specific elevator since its speed is in the range from 800-1000 fpm, and that the deceleration of the loaded elevator becomes 22 ft/sec^2, which is well below the maximum allowable of 32.2 ft/sec^2.

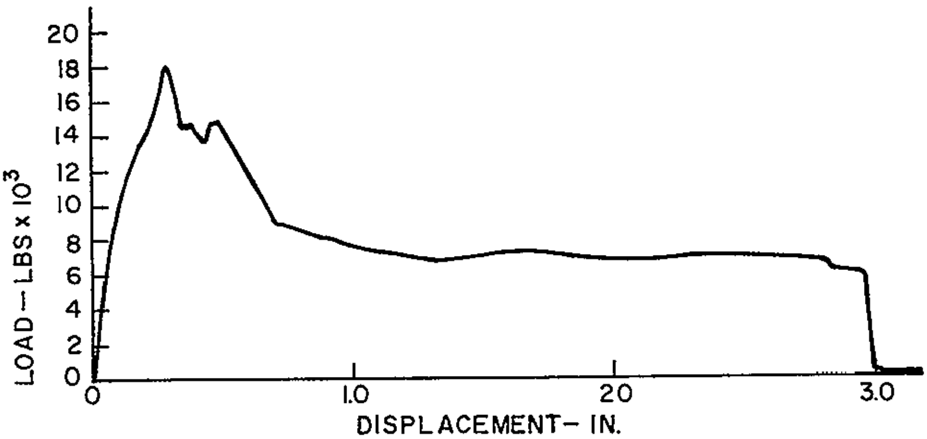
The resisting force, F, used above was assumed to be a constant value equal to the weight of the unloaded elevator. That this is a reasonable assumption for the tube and mandrel is shown in Figure 7.1. The initial spike is of short duration compared to the steady state force which can be made to go on for several feet at an essentially constant value. It is also possible, by modifying the end conditions of the tube and/or the taper angle of the mandrel, to change the shape of the force-displacement curves to satisfy a rather wide variety of requirements. In particular, the peak can be raised or lowered without affecting the steady force or the entire curve can be raised or lowered.

The above analysis, as stated, was based only on static load-deflection characteristics of the energy absorber which are independent of time. Consideration of the dynamic load-deflection characteristics (refer to part (B) following) leads to the conclusion that some modification of the impact end of the tube may be required to lower the magnitude and the duration of the initial spike, which results from accelerating the tube up to the initial velocity of the elevator. The presently used hydraulic buffer has a similar initial spike but of greater magnitude than that of the proposed tube and mandrel. This is apparent when the construction of the hydraulic buffer is considered. Basically these buffers consist of a steel plunger which is forced, by the impact, down into an oil filled reservoir. Thus, not only is it necessary to accelerate the plunger (a tube more massive than that used in the proposed tube and mandrel), but also it is necessary to accelerate the mass of the internal parts such as the return spring, oil seals and other parts as well as the oil which is forced out of the escape holes in the plunger. To attenuate the initial spike, in the case of the hydraulic buffer, a rubber contact block is used. It is assumed that a similar approach would be satisfactory in attenuating the initial peak load in the tube and mandrel.

2. Dynamic Analysis

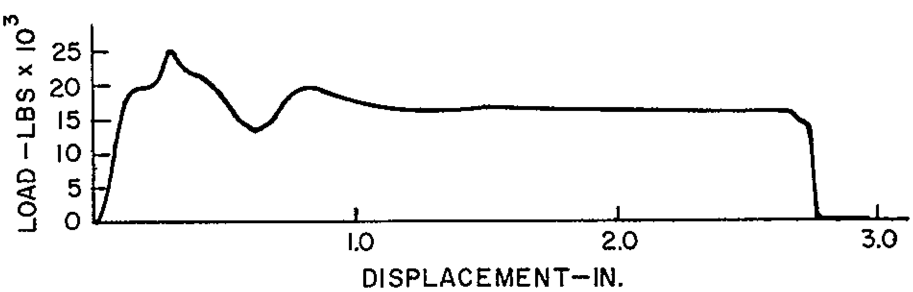
The fact that the tube must be accelerated up to the velocity of the elevator in a relatively short time means that an inertia load will be superimposed on the load required to simply split the tube into eight segments, and curl up the segments. The following analysis, involving wave mechanics, was performed to determine, for the worst case, what the dynamic load would be.

When a mass, M, (represented by the elevator) traveling at a velocity, V_0, impacts the end of a tube, such as that used in the tube



(a)

Static Load-Displacement Curve For The 3.31" I. D. × .094" Mild Steel Tube On A 37° Mandrel (see Figure 7.5 for mandrel configuration and dimensions)



(b)

Static Load-Displacement Curve For The 3.25" I. D. × .125" Mild Steel Tube On A 30° Mandrel (see Figure 7.5 for mandrel configuration and dimensions)

Figure 7.1 Static Load-Displacement Curve For the Tube and Mandrel Energy Absorbers Designed For Use As Elevator Buffering Devices and mandrel device, a longitudinal stress wave is generated which travels down the tube as indicated in Figure 7.2.

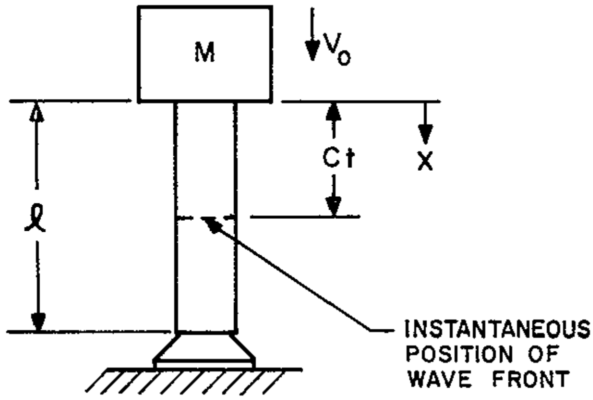


Figure 7.2

This stress wave creates a high initial resistance, which damps out to the average steady state value required to run the tube down over the mandrel. The analysis required to determine the magnitude of this stress wave neglecting the radial inertia of the tube follows. (It should be pointed out that this analysis is for a worst case condition, i.e., it assumes a plane wave oscillating in the tube. This condition actually exists only for the first half cycle since the maximum static load which the tube can carry is exceeded when the wave reaches the mandrel, causing it to split and curl.)

Let σ, ϵ, u, E and ρ represent the longitudinal stress, strain, displacement, elastic modulus and density respectively. The stress-strain law is then given by

$$\sigma = E\epsilon \qquad (7.3)$$

The strain displacement relation is

$$\epsilon = \frac{\partial u}{\partial x} \qquad (7.4)$$

Equilibrium yields

$$\frac{\partial\sigma}{\partial x} = \rho\frac{\partial^2 u}{\partial t^2} \tag{7.5}$$

Combining (7.3), (7.4), and (7.5) gives the wave equation for displacement.

$$\frac{\partial^2 u}{\partial t^2} - \frac{E}{\rho}\frac{\partial^2 u}{\partial x^2} = 0 \tag{7.6}$$

The solution of (7.6) is well-known.

$$u = f(x - ct) + g(x + ct) \tag{7.7}$$

Where C is the wave speed given by

$$c = \sqrt{\frac{E}{\rho}} \tag{7.8}$$

Let

$$m = M/\text{Area of tube} \tag{7.9}$$

Then Newton's law yields the boundary condition at $x = 0$.

$$E\,\frac{\partial u}{\partial x}(0,\ t) = m\,\frac{\partial^2 u}{\partial t^2}(0,\ t) - mg \tag{7.10}$$

The initial condition is

$$\frac{\partial u}{\partial t}(0,\ 0) = V_0 \tag{7.11}$$

Immediately after impact the function, g, does not appear in (7.7). In terms of f, the boundary condition (7.10) becomes

$$E\,f'(-ct) = m[c^2 f''(-ct) - g] \tag{7.12}$$

The initial condition (7.11) gives

$$-cf'(0) = V_0 \tag{7.13}$$

Solving (7. 12) gives

$$f'(-ct) = A\,e^{-\alpha ct} - \frac{mg}{E} \tag{7.14}$$

where

$$\alpha = \frac{E}{mc^2} \quad ; \quad A = -\frac{V_0}{C} + {}^{mg}/_{E} \tag{7.15}$$

Now from (7. 14)

$$f'(x - ct) = A\,e^{\alpha(x - ct)} - \frac{mg}{E} \tag{7.16}$$

Integrating

$$f(x - ct) = \frac{A}{\alpha}\,e^{\alpha(x - ct)} - \frac{mg}{E}(x - ct) + B \tag{7.17}$$

Now continuity at the wave front requires

$$u = f(0) = 0 \tag{7.18}$$

Then (7. 17) can be written

$$u = f(x - ct) = \frac{A}{\alpha}\left[e^{\alpha(x - ct)} - 1\right] - \frac{mg}{E}(x - ct) \tag{7.19}$$

The stress at x = 0 can be determined from (7. 16)

$$\sigma_0 = EA\,e^{-\alpha ct} - mg \tag{7.20}$$

This equation holds until the wave front strikes the upper end of the tube, that is to say when

$$t = \frac{2\ell}{c} \tag{7.21}$$

Thus at this time the stress has decayed to a value of

$$\sigma_0 = EA\,e^{-2\ell\alpha} - mg \tag{7.22}$$

The spike represented by (7.20) just before striking the upper end again has only decayed about 1% for a typical example.

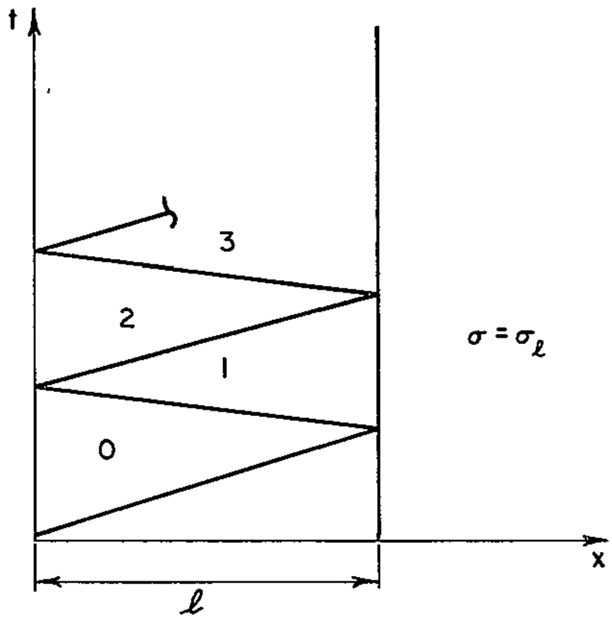


Figure 7.3 Approximate Solution for Long Times

For short bars; i.e., where the dissipation of the spike is caused mainly by conditions at the far end, we may assume a region of constant stress and velocity in the numbered regions in Figure 7.3. By applying the shock relations we may arrive at a solution for the decay of the spike. The shock relation is

$$\sigma^+ - \sigma^- = -\rho c[V^+ - V^-] \tag{7.23}$$

We may approximate the boundary condition by

$$\frac{V_{n+2} - V_n}{\Delta t} 2m = \sigma_{n+1} + \sigma_n \tag{7.24}$$

Δt is the reflection time.

Making use of (7.23) and (7.24) gives

$$V_{n+1} = V_n + \frac{1}{\rho c}[\sigma_n - \sigma_\ell]$$
$$\sigma_{n'+1} = \sigma_\ell \qquad (7.25)$$

for

$$n = 0, 2, 4, \ldots\ldots$$

For the even numbered regions we derive

$$V_{n+2} = \left(\frac{\Delta t}{\rho c \Delta t + 2m}\right)\left\{\sigma_e + \rho c\, V_{n+1} - 2mg + \frac{V_{n2m}}{\Delta t} + \sigma_n\right\} \qquad (7.26)$$

$$\sigma_{n+2} = \sigma_\ell + \rho c\,[V_{n+1} - V_{n+2}] \qquad (7.27)$$

for

$$n = 1, 3, 5, \ldots$$

Figure 7.4 is a plot (using actual elevator weights and speeds) of the recursive relations (7.25), (7.26) and (7.27) and shows the total instantaneous force acting at the elevator end, as a function of time. These graphs show that for the worst case the peak load will significantly exceed 2.5 G's for 1/25th of a second when the static peak load of the tube is added. However, it is believed that the tube will split on the first reflection and that the load will drop off faster than that shown in the curves. In the event that this load-time requirement cannot be met, an attenuator pad similar to that presently used on the oil buffer can be used to reduce the initial spike to an acceptable level.

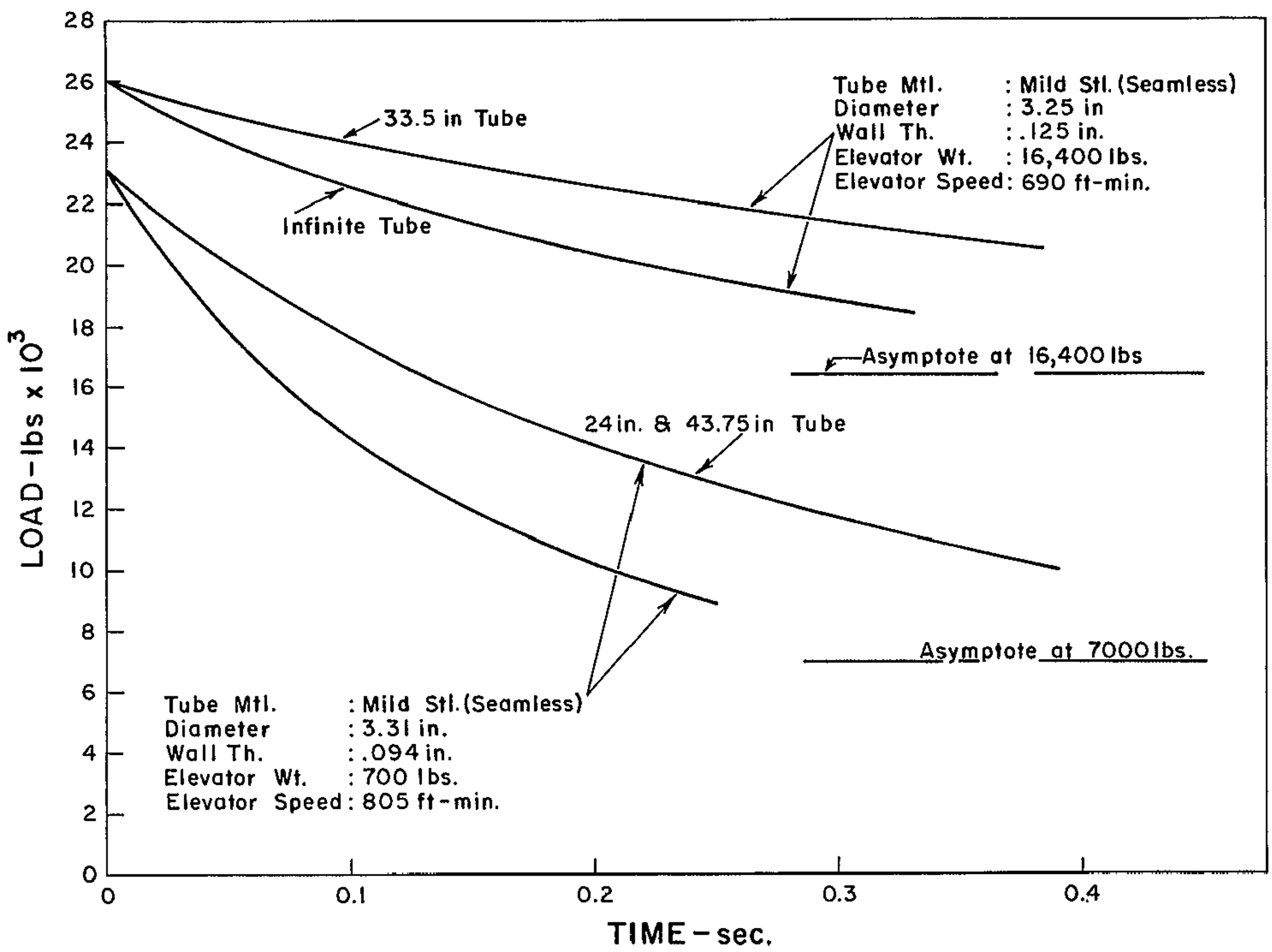


Figure 7.4 Plot of the Recursive Relations Showing the Total Instantaneous Load of the Elevator End of the Tube, Not Including the Static Peak Load of the Tube

2. Fabrication and Static Testing of a Prototype Tube and Mandrel

To coincide with the requirements outlined above a tube and mandrel device has been designed and statically tested for the three elevators listed in Table 1.

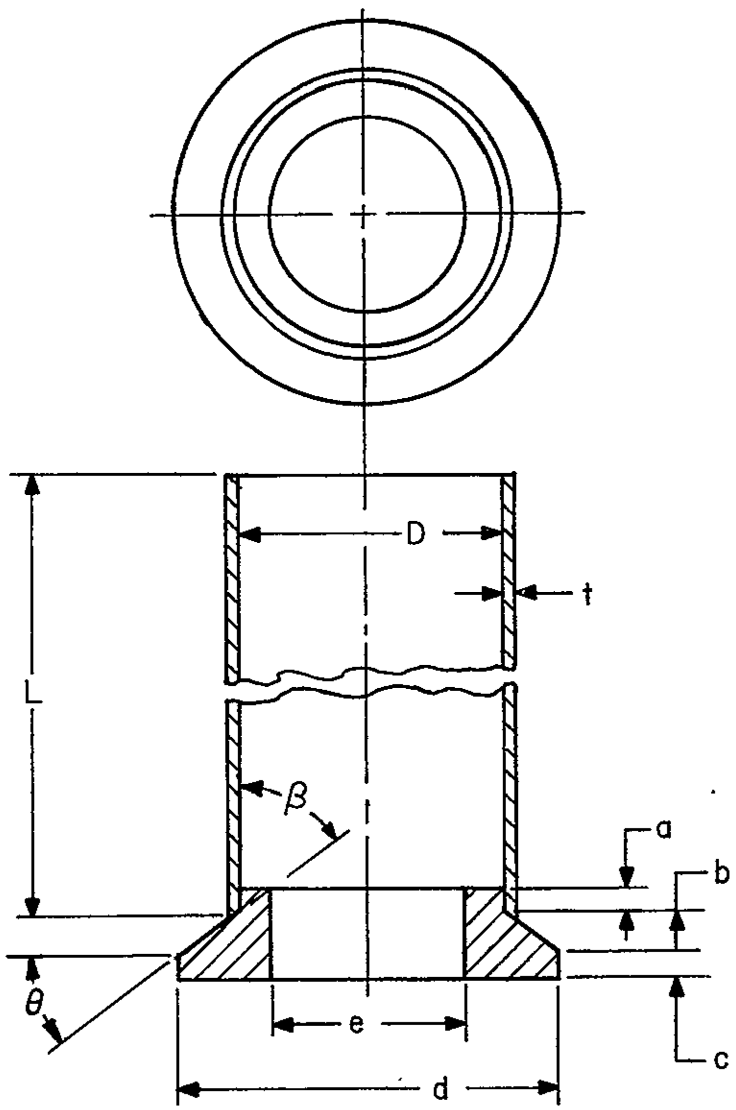
Figure 7.5 is a sketch of the device which, it is believed, will satisfy the requirements for the three cases. For case 1 the device supplies an average steady state retarding force of 7,000 lbs, a peak force of 18,000 lbs, and has a stroke of 15 in. This length allows for the small loss of stroke due to formation of curls. See Figure 7.1a. For case 2 the device differs from that in case 1 in the fact that the length of stroke is increased to 56 inches, which compensates for the increased speed. For case 3 the design had to be modified and uses a mild steel tube with a 1/8 inch wall. It supplies an average steady state force of 16,000 lbs and a peak force of 25,000 lbs. See Figure 7.1b. The length of tube required for this elevator in both the loaded and unloaded configurations is 40 inches. In all three cases the design is intended to decelerate the unloaded elevators at an average deceleration of 1 G while for the loaded elevators the decelerations are .71 G, .71G, and .67 G (cases 1 - 3 respectively).

It will be necessary to attach the tube to the mandrel in such a way that the two will be a unit and will only require the mandrel to be fastened (bolted or welded) to stands at the bottom of the elevator shaft. Fastening of the tube to the mandrel will probably be done using small tack welds which will not add significantly to the breakaway or peak load.

It should be reiterated that the device has not yet been tested under dynamic conditions, only static testing has been performed on devices of various lengths up to 18 inches. A field test and evaluation program is anticipated to experimentally determine the dynamic properties of the device.

3. Market Potential

The elevator industry is quite large and growing with the increasing number of tall buildings being constructed. Otis Elevator Company has an annual volume of approximately $500,000,000.00 in worldwide sales..



Case	L	D	t	β	e	a	b	c	d	θ
1	15"	3.31"	.094"	30°	2"	.25"	.278"	.25"	4"	37°
2	56"	3.31"	.094"	30°	2"	.25"	.278"	.25"	4"	37°
3	40"	3.25"	.125"	30°	2"	.25"	.196"	.25"	4"	30°

Figure 7.5 Tube and Mandrel Design for Use as an Elevator Buffering Device

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VIII. AUTO-HIGHWAY SAFETY APPLICATIONS

A team of faculty, students and professional staff are engaged in the development of attenuation systems for highway gores. The achievements to date as well as future aims are reported in this Section. A method for initially screening available energy absorbing devices is presented. The evaluation of the remaining devices by means of testing and analysis is discussed. The procedures for inexpensive and flexible testing of attenuation systems using scale modeling techniques are developed and applied to a case of head-on collision.

Two senior students projects for gore buffers are also described. These are applications of two out of three concepts for attenuation devices developed by three of the faculty and professional staff last September 1969.

A. BASIC CONSIDERATIONS

A current awareness of highway safety needs makes energy absorbing devices very timely. (1)* This awareness was pointed out in a paper given at a recent highway engineering conference. (2) The paper states in part that:

> "---the single vehicle "ran off the road" accident is a leading source of fatalities on our Interstate system. ---Elevated gore structures such as exit ramps on bridges are a prime example. The rather hostile nature of the nose of the bridge parapet and railing in such an area involving driver decisions at freeway speeds, and the large number of such structures in urban areas with high average daily traffic figures have combined to make real problem areas of such structures. ---Impact energy absorption barriers can be used to reduce the severity of these hazards. Conventional guard rail installations are not well suited for such areas, inasmuch as they are best suited for glancing impacts and are not satisfactory for the kind of high speed, near head-on collision which may occur at these sites."

Significant efforts to protect vehicle occupants from the lethal effects of impacting a fixed roadside obstacle were initiated by the Bureau of Public Roads in December of 1966. Under a program entitled "Structural Systems in Support of Highway Safety," a short-range study was initiated to develop a first generation of attenuators based largely on full-scale impact tests of systems developed from existing technology. (3) The criteria established for evaluating the tests were limited to preliminary estimates of vehicle weight ranges, maximum impact speeds, maximum angle of impact, and maximum average passenger deceleration rates. This program has resulted in the evolution of a number of devices that provide varying degrees of impact attenuation. (4) However, these devices are still in the experimental stage and subject to continual change. Also, a lack of information on adequate performance and cost criteria has made it almost impossible to establish design standards. Thus existing devices are each designed under different criteria, making it very difficult to compare their performance characteristics and overall cost factors. Based on this

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* Indicates the reference at end of part B

appraisal of the state-of-the-art, it was decided that the NASA study would concentrate on patents in the area of energy absorbing devices, directing a portion of our efforts toward two aspects of auto-highway safety: applications to the automobile structure, and applications to the fixed highway system. This Chapter concentrates on the latter area.

1. The Basic Problem

The division of the roadways on an elevated bridge structure necessarily creates an intersection of parapets and railings of the respective roadways. The critical problem in terms of impact attenuation is at this intersection or gore area as shown in Figure 8.A.1. The shaded area can be designed either as a recovery zone, or as an attenuation and/or redirection zone. Under normal conditions, vehicles pass to one side or the other of this zone; but in the event of an erratic maneuver caused by driver indecision or confusion, the vehicle may well enter the zone. If the vertical and horizontal alignment of the roadways permit the zone to be large enough, the driver may be able to bring his vehicle under control without striking the parapet along the edge of the structure. However, in most situations the available area is too restricted, thus making it necessary to redirect or stop the vehicle with tolerable decelerations to the occupants and minimum conflict with other vehicles traveling in the lanes adjacent to the gore area.

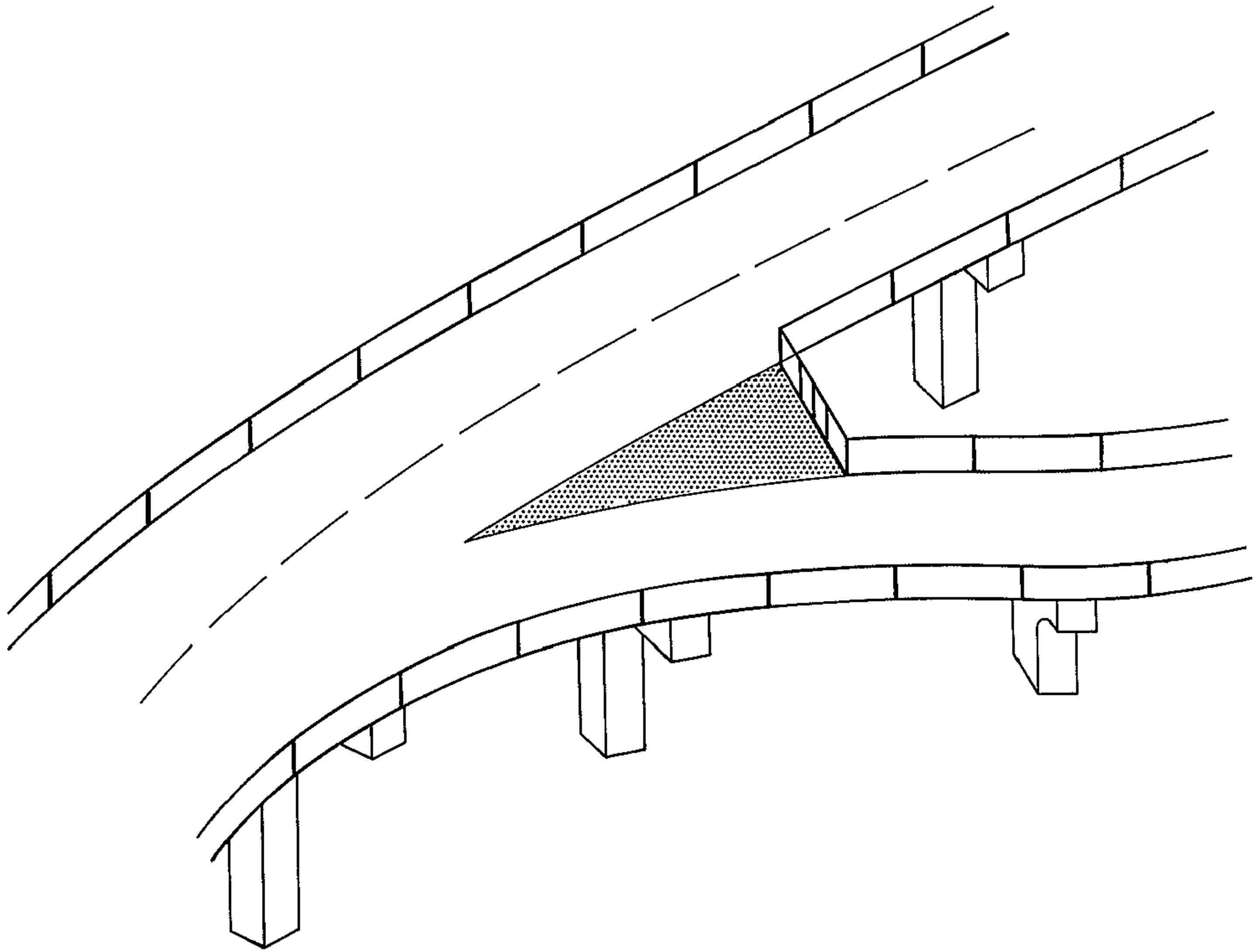


Figure 8.A.1

2. The Attenuation System

Protection from gore impacts can be provided by a buffering system. Such a system is composed of individual energy absorbing devices, tie down and connecting elements, and load distributing elements.

A generalized attenuation system is shown in Figure 8.A.2. The system is somewhat like a structural column that must absorb axial loads in stopping the vehicle, and bending and shear forces in redirecting the vehicle. It must also absorb shear and bending on the vertical plane to compensate for eccentric loadings due to differences in the relative heights of the mass centers of the vehicle and the attenuation system.

The magnitudes of the axial and shear forces imparted to the attenuation system by the vehicle are a function of the weight and speed of the vehicle, the angle of impact, and the point of impact. The actions of the vehicle during an impact are largely a function of the reacting forces generated by the attenuation system. Axial deceleration is governed by the collapse of the system and redirection is controlled by the bending and shear resistance.

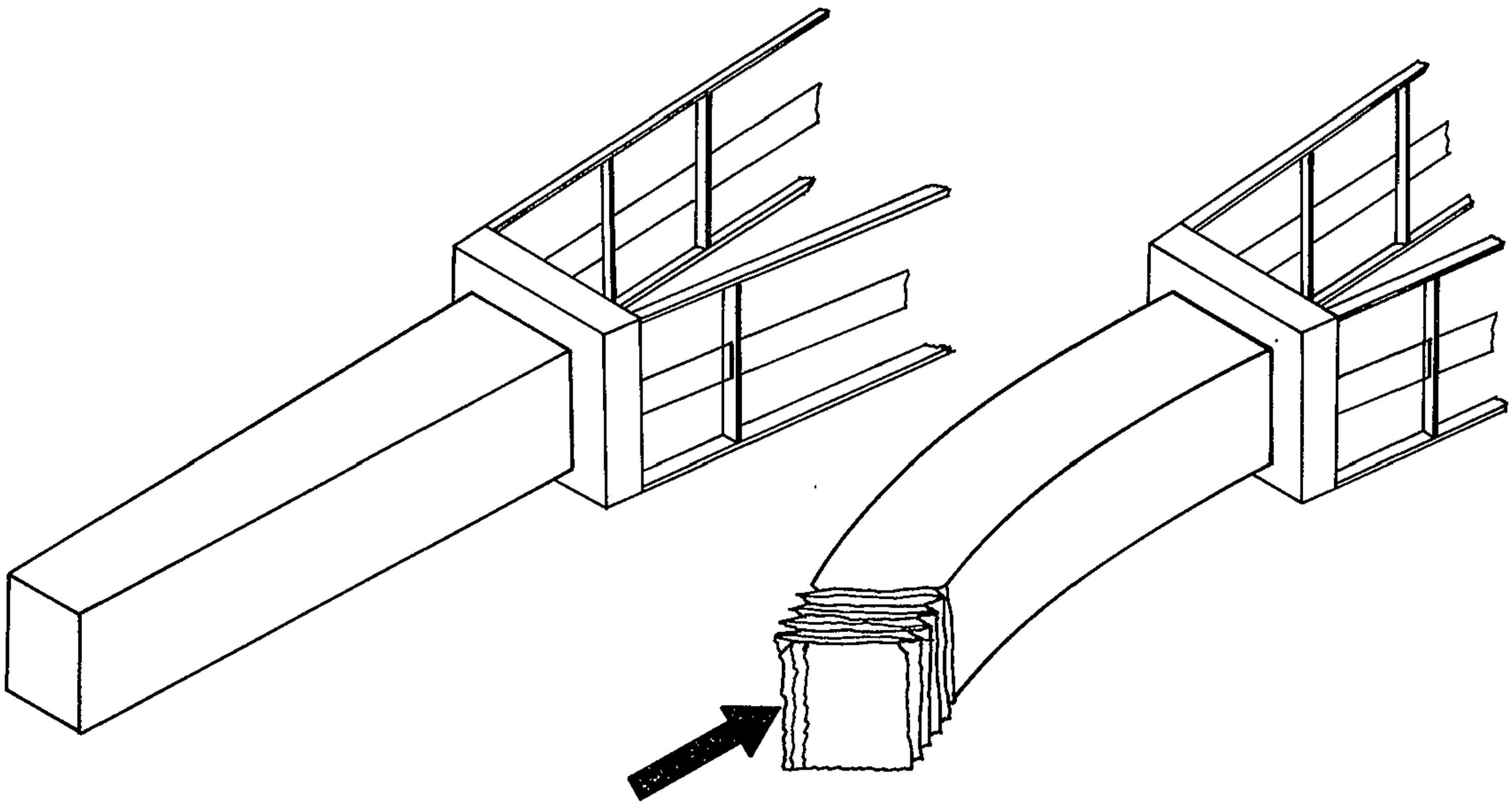


Figure 8.A.2

3. Identification of Performance Criteria

Operational buffering systems cannot be properly designed nor adequately tested until performance-design criteria have been established. To date, there are insufficient data on vehicle dynamics and driver responses during an actual impact to fully define such criteria. Full-scale simulated crash tests have been used to evaluate a number of the vehicle dynamic characteristics and passenger deceleration forces. (5), (6) However, the prohibitive costs and physical danger involved in such tests have not allowed testing over a wide range of conditions. In addition, the majority of such tests have presumed no driver response and a linear path of the vehicle prior to impact.

The "4-S" program of the Bureau of Public Roads referred to in the introduction has defined a partial set of criteria for purposes of evaluating the results of full-scale crash tests. (3) However, these criteria are not inclusive in terms of performance standards. Three major criteria have yet to be defined in terms of attenuator design. The first is the range and probability of vehicle dynamic conditions prior to impact. Cost-effective design must necessarily be based on the majority of actual conditions rather than all possibilities or only those which are easily tested in a simulated crash. It is suggested that the evasive actions of a driver prior to impact will significantly modify the dynamic response observed in driverless test vehicles.

A second set of criteria to be evolved should consider the relationship between the time-deceleration history of the vehicle as shown in Figure 8.A.3 and resulting injury potential to the occupants. Only through effective measures (indices) of injury exposure based on probable human tolerances will there be an adequate definition of deceleration constraints for attenuation systems. Without such defined constraints, valid comparisons of cost-effective designs are not possible.

The third set of criteria relate to the action of the vehicle during and after impact. No limitations on redirection angles or angular momentum imparted to the vehicle by the attenuation system have been established. Since the majority of bridge gore situations occur on heavily traveled freeways, the action of the vehicle after impact, may, in fact, be of greatest concern in terms of overall safety of the freeway system.

With the availability of performance criteria, it will no longer be necessary to subjectively evaluate the "effectiveness" of particular

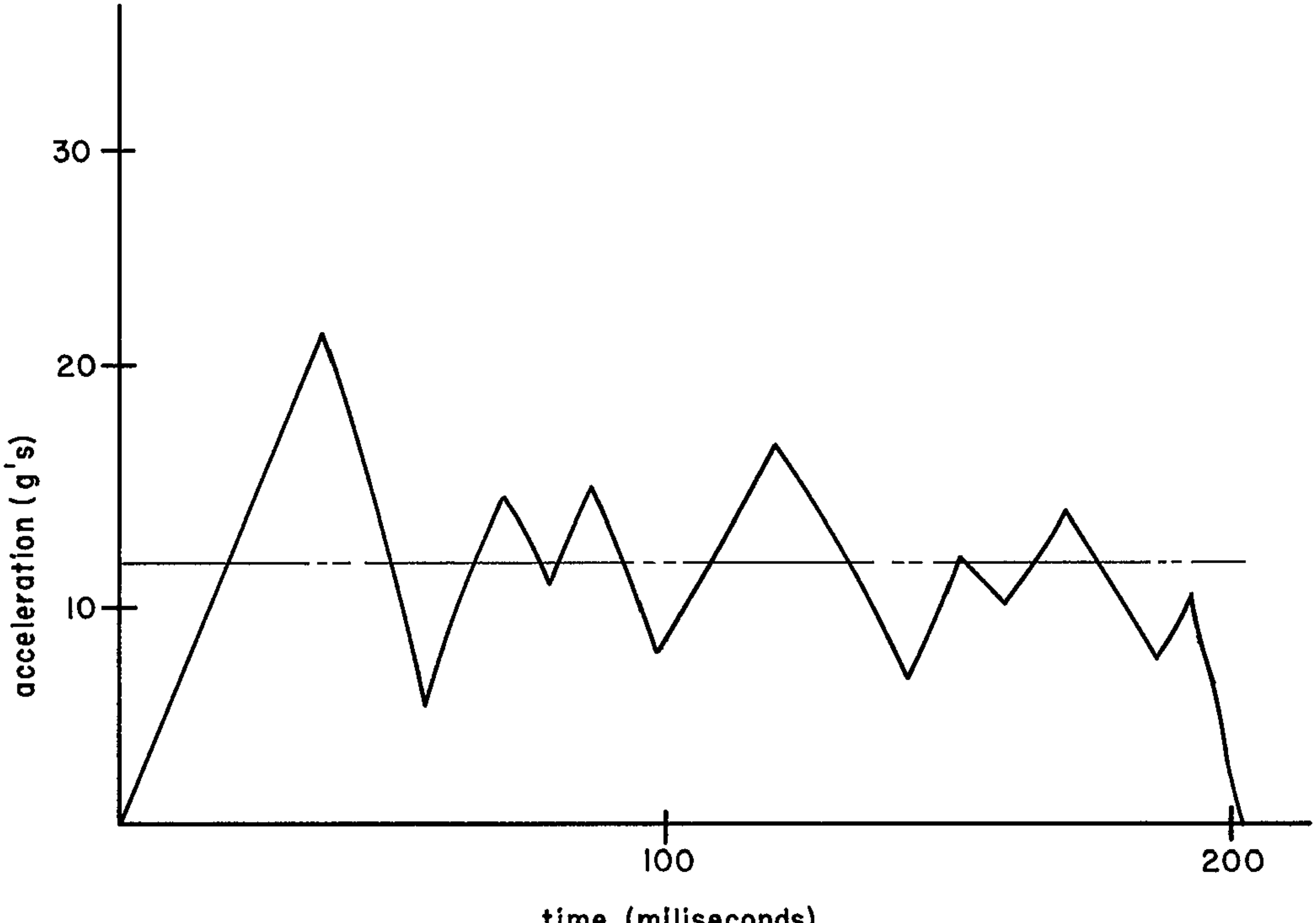


Figure 8.A.3

buffering systems. Systems shall be termed either effective or not effective depending, respectively, on whether or not they satisfy the existing performance criteria. System selection among the effective systems may then be made on the basis of size and total cost, i.e., some combination of initial and maintenance costs.

4. The Buffering System

In practice, a complete buffering system (one gore buffer) is a fairly complex arrangement of energy absorbing units and tie-down, interconnecting, and load-distributing elements. Attempts to design a buffering system to meet accepted performance criteria will, therefore, be difficult and will probably involve some degree of trial and error. Recognizing this, we have attempted to simplify the design problem by formulating some rough guidelines. We feel that buffers meeting these requirements will be more likely to satisfy a reasonable set of performance criteria.

1. The buffer mass activated at impact should be small compared to the weight of the impacting vehicle.

2. The impacting vehicle should be assumed to be rigid.

3. The force-displacement curve of the barrier should be optimal or near optimal.

4. Buffer deformation and motion should be localized to the immediate area containing the impacting vehicle.

5. The buffer should not eject material onto the traveled roadway.

6. The buffer should not store mechanical energy.

7. The center of gravity of each portion of the attenuator should be above the center of vehicle load application.

8. The buffer should not produce significant angular accelerations until the vehicle has been entrapped.

9. The lateral stiffness of the barrier should be greatly increased near its base.

At the moment of impact, the portion of the buffer in direct contact with the vehicle and the vehicle itself are rapidly brought to a common velocity. When the ratio of barrier mass activated at impact to vehicle mass is not small compared to unity, then this common velocity is significantly different from the vehicle's initial velocity, and large deceleration forces occur. It follows, for example, that buffers should not be designed with massive rigid bumpers (guideline 1).

The deceleration forces imposed on the passenger compartment after impact depend on the stopping force provided by the buffer as well as the crushing characteristics of the front end of the vehicle. Front end crushing is not easily incorporated into buffer design because of the variation in crushing with vehicle make and model. We believe that vehicle crushing should be ignored in buffer design, i.e., it should be assumed that the impacting vehicle is rigid(guideline 2). This assumption provides a margin of safety since vehicle crushing attenuates the forces felt in the passenger compartment. Full size or scale model acceptance testing could then be conducted with specially constructed rigid vehicles. The test vehicles would be reusable and comparisons between different tests would become more meaningful.

Vehicle response after buffer impact must be such that acceptable levels of human tolerance are not exceeded. This condition places an upper bound on allowable force levels and therefore on the force-displacement response of the buffer. The force-displacement curve for which the acceptable limits of human tolerance are attained (but not exceeded) at all times during the deceleration process is called the optimum force-displacement curve.

To illustrate, consider the simple criteria that the g loads on the vehicle not exceed some constant value $\overline{g}$. The buffer must safely stop all vehicles traveling less than $\overline{v}$ fps and weighing between W_0 and W_f lbs. The initial force F_0 cannot exceed $W_0\overline{g}$ until the lightest vehicles have been stopped. Thus the optimum force-displacement curve is constant at F_0 until the kinetic energy of a vehicle of weight W_0 traveling at velocity $\overline{v}$ has been dissipated. Equating the energy dissipated to the area under the force-displacement curve gives the penetration or stopping distance, $D_0 = \overline{v}^2/64.4\overline{g}$. Beyond D_0 the optimum curve rises continuously since the force level may be raised without exceeding $\overline{g}$ on the heavier vehicles that remain. The equation governing the shape of the optimum curve beyond D_0 is

$$F_0D_0 + \int_{D_0}^{X} F(\overline{X})\, d\overline{X} = \frac{W}{64.4}\, \overline{v}^2 \qquad (1)$$

where X is the distance necessary to stop an impacting vehicle of weight W traveling at the maximum velocity $\overline{v}$. The left side of (1) is the work done on the buffer while the right hand side is the maximum kinetic energy of a vehicle of weight W. At the point X where the vehicle has been stopped the force must produce the maximum allowable g load, i.e., $F(X) = W\overline{g}$. Substituting this into equation (1) and solving the resulting integral equation yields the optimum force-displacement relation for $X > D_0$.

$$F(X) = W_0\overline{g}\,(\exp)\,64.4\overline{g}\,(X - D_0)/\overline{v}^2 \tag{2}$$

The complete optimum force-displacement curve for this example is shown in Figure 8.A.4. The distance required to stop a vehicle of weight W impacting at $\overline{v}$ fps is found by letting $F(X) = W\overline{g}$ in (2) and solving for X. This result is

$$X = \frac{\overline{v}^2}{64.4\overline{g}} \left(1 + \log \frac{W}{W_0}\right) \tag{3}$$

The minimum stopping distance that the barrier must provide is found by putting $W = W_f$ into (3).

In practice, it will not often be feasible to design a barrier with an optimum force-displacement curve. However, the optimum curve may readily be approximated by designing the buffer as a series of energy absorbing units whose components operate at different force levels, as shown by the dashed curve in Figure 8.A.4.

Consider two buffers designed with optimum (OP) and non-optimum (NOP) force-displacement curves respectively. The force-displacement curve of NOP must be below the optimum curve (or the design criteria will be violated) and therefore buffer NOP dissipates less energy in a given displacement than buffer OP. Since both buffers must be capable of dissipating equal amounts of energy, the minimum stopping distance provided by buffer NOP is greater than that provided by buffer OP. Other things being equal, buffer NOP will be larger than buffer OP, with the difference in size proportional to the variation of NOP's force-displacement curve from the optimum.

In order to maintain simplicity in our example, we ignored many effects, e.g., dynamic loads, multi-directional response, dependence of injury on the time integral of a function of acceleration. The inclusion of these effects does not change the basic facts that the operating force levels of a buffer are constrained by the performance criteria and that the size of a buffer is minimized by optimizing its force characteristics within this constraint (guideline 3).

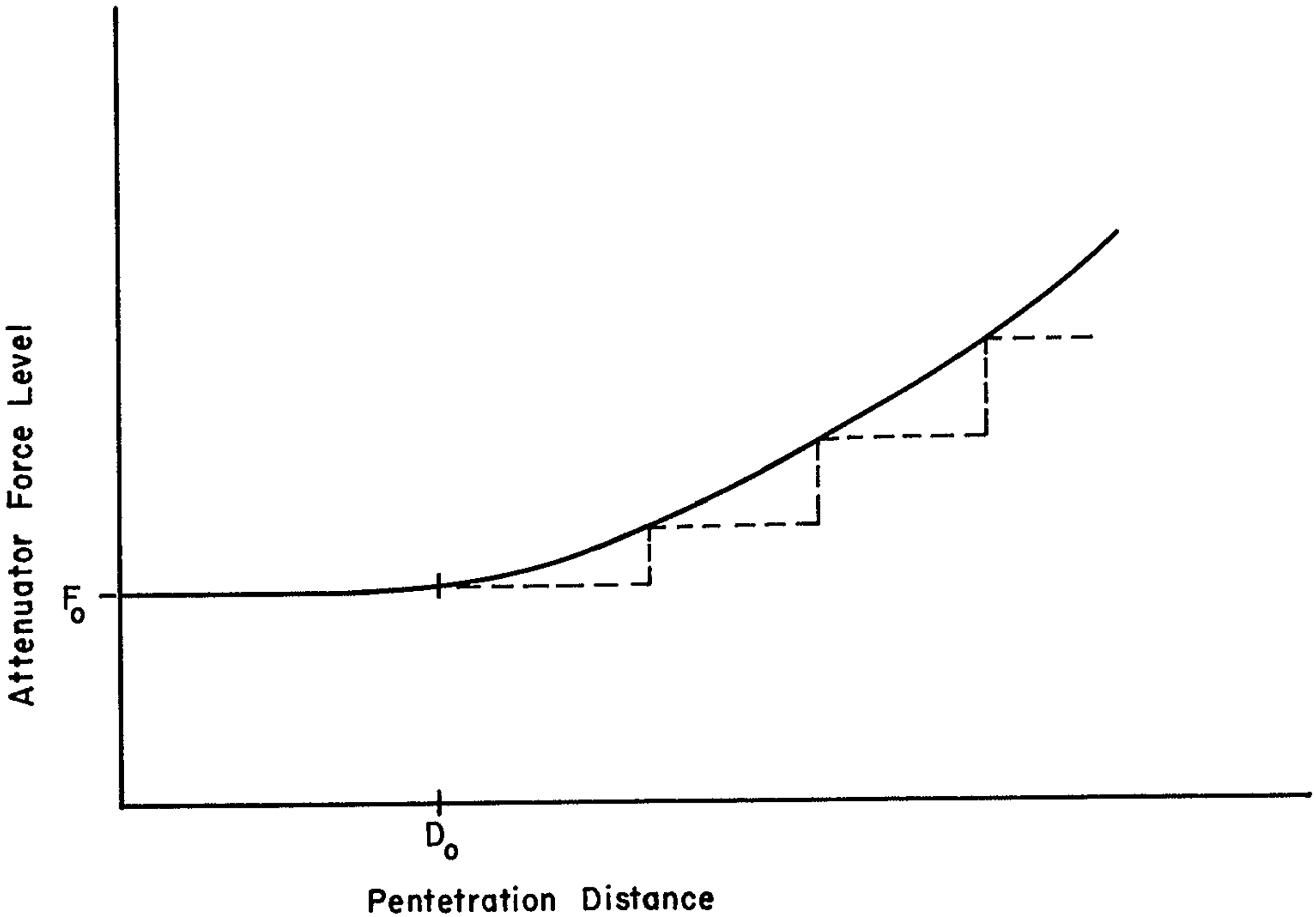


Figure 8.A.4

The importance of size in attenuator design follows from the consideration that smaller buffers will have a simpler support structure, a lower frequency of impact, and increased probability of acceptance for use in existing gore areas with limited placement area. A second parameter which is important in determining buffer size, namely, the length/stroke ratio, will be discussed in Section 5.

The impact of a vehicle with a gore attenuator should not create a hazardous environment for those vehicles in the vicinity of the impact area. Therefore, the impact should not move the buffer so that it interferes with the flow of traffic (guideline 4). It should not eject material onto the roadway (guideline 5). Furthermore, the impacting vehicle should not, under most conditions, be allowed to reenter the roadway. In a direct or semi-direct hit on the buffer, the driver is not likely to be in full control of the vehicle and the vehicle itself will be at least partially disabled by the impact. The most likely means of a vehicle escaping from the buffer after impact are by elastic rebound, ramping, and spinout. Elastic rebound is easily controlled by selecting energy absorbing units which dissipate rather then store energy (guideline 6). To prevent ramping, the center of gravity of the individual units in the buffer must not be lower than the center of force application so that the units have no tendency to rotate under the vehicle (guideline 7). Spinout is the most difficult of the three escape mechanisms to control. Spinout may occur when the vehicle, with no initial angular velocity, impacts the buffer off center. The resulting moment produces angular accelerations which can rotate the vehicle out of the barrier. It may also occur with an initially spinning vehicle which is simply redirected by the buffer. We feel that the probability of spinout in both these cases will be greatly reduced if the outside of the buffer is readily deformable under relatively small forces (guideline 8). Under these circumstances, the buffer should tend to wrap around the vehicle without redirecting it. As the vehicle penetrates to the stiffer part of the buffer, the tendency to spinout will be decreased by the lateral and friction forces provided by the material enclosing the front end of the vehicle.

In circumstances where the buffer cannot safely arrest the vehicle, the vehicle must be redirected. Large local deformations at redirection locations are undesirable. The buffer must therefore be designed with increased lateral stiffness in these areas (guideline 9).

5. Selection of Devices for Use in Gore Buffering Systems

Most of the available energy absorbing devices can be engineered to operate over a wide range of force and energy levels. Devices with this type of flexibility will, in all probability, be able to be incorporated into an effective attenuation system. Comparison between devices, therefore, will generally be made on the basis of their effect on total system cost.

The initial phase of our program was concerned with the identification of those characteristics of an individual device which play an important role in the cost of the attenuation system. Our purpose was to provide a means of qualitatively comparing the 53 different devices with which the program was begun without the expense of system design, fabrication, and testing. The characteristics of major importance were concluded to be:

a. Reliability

Gore attenuators are required to operate over a wide range of environmental conditions during a time period measured in years. Some devices are virtually insensitive to weather and time effects and require no special care. Devices, however, which rely on friction and/or close tolerances for proper operation, and devices which contain weather sensitive materials will require special care to insure reliability. This will be reflected in higher unit costs.

b. Cost of Manufacture and Assembly

The relationship of this item to initial cost is obvious. Devices which use standard materials and do not require close tolerances tend to have low manufacturing costs. Those devices which use off-the-shelf items are particularly attractive in this regard.

c. Reusability

Devices which are completely or partly reusable can be expected to exhibit lower maintenance costs.

d. Multi-Directional Load Capability

Buffer impacts occur over a range of positions and directions. Proper functioning of devices which only operate under a restricted range of loading directions requires the use of support and load directing structures which add to system cost.

e. Material Efficiency

Material efficiency is defined as the energy absorbed per pound of device. The minimum weight of a buffer is the total energy to be dissipated divided by the material efficiency of the basic device. Devices with low material efficiency tend to have high material costs.

There is a large variation in material efficiency with changes in materials and geometry. In general, however, when the weight of the deformed material is not a significant percentage of the total weight of the device, material efficiency is low.

f. Force-Displacement Curve and Length/Stroke Ratio

These characteristics have been grouped since together they approximately determine the minimum size of the attenuation system. System cost increases with size because of the increase in necessary supporting structure (including the foundation or pad for the attenuator) and the increase in maintenance costs due to a higher frequency of impact. We have already discussed the role of the system force-displacement curve in determining size. In order to design a buffer, the force-displacement relations of the individual devices as a function of material properties, geometry, and direction of load application must be known. These are determined by a combination of analysis and experimentation. Typical force-displacement curves (for a folding tube and metal shearing device) under axial loading are shown in Figure 8.A.5. The oscillatory nature of these force-displacement relationships are common to many energy absorbing devices.

The maximum useful displacement provided by a device is called its stroke. The strokes of three energy absorbing devices are shown in Figure 8.A.6. The minimum length of a device necessary to produce one foot of stroke is its length/stroke ratio. The minimum length of an attenuation system is the product of the minimum stopping distance which must be provided and the length/stroke ratio of the device being used in the system.

The use of these characteristics has enabled us to eliminate most of the original devices from consideration for use in attenuating systems for gores. The five patents which survived this critical review

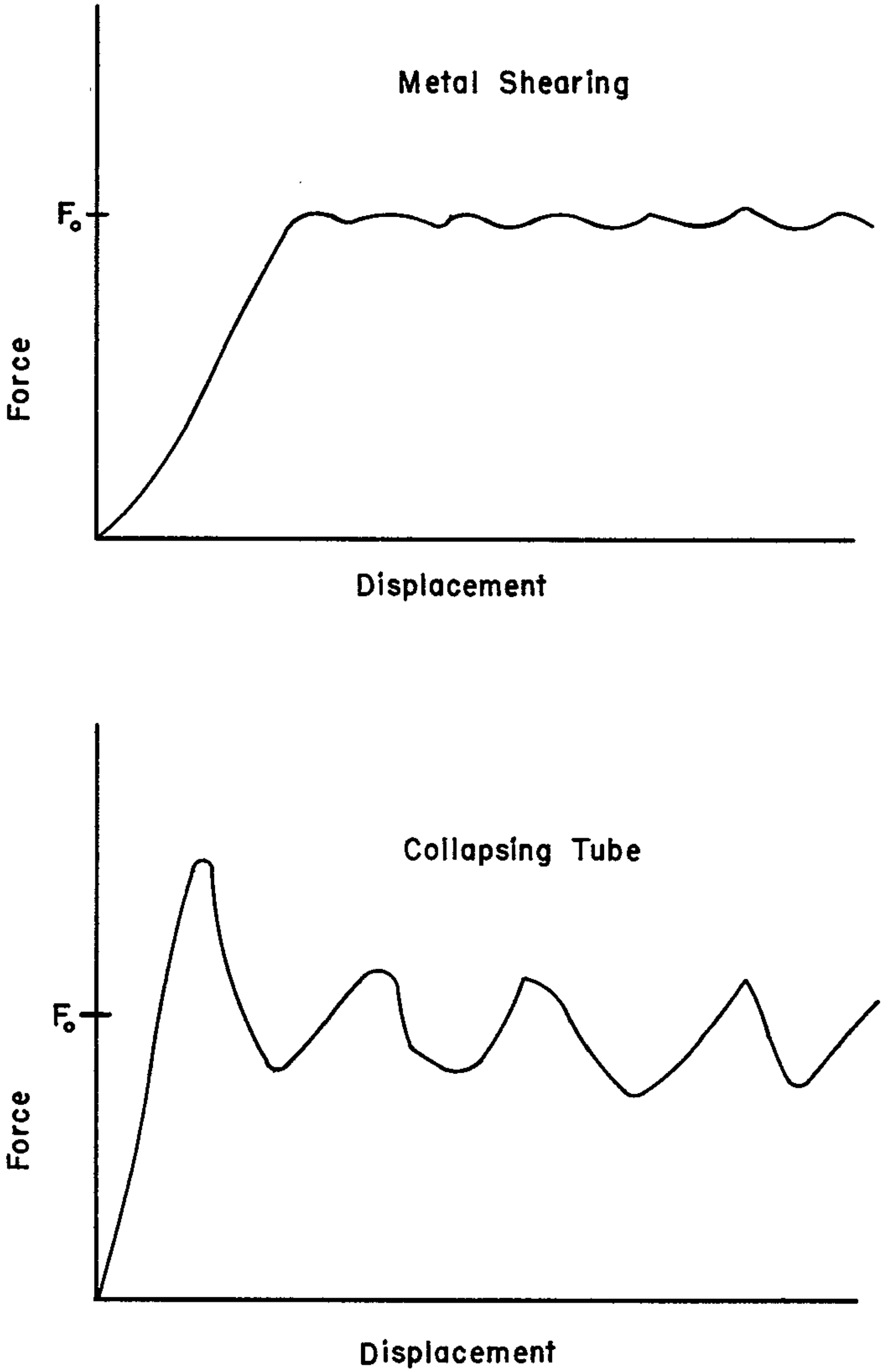
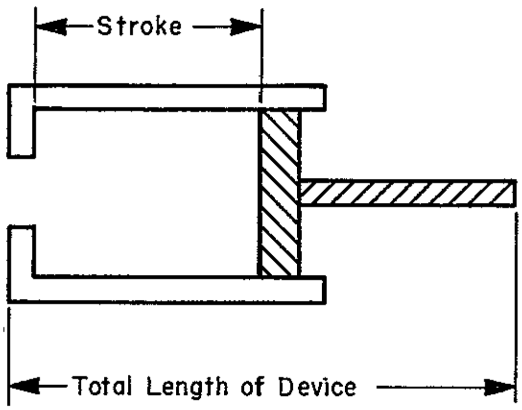
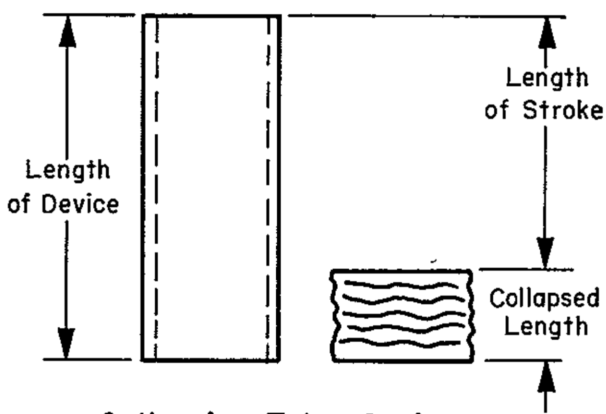

Metal Shearing
F_o
Force
Displacement
Collapsing Tube
F_o
Force
Displacement

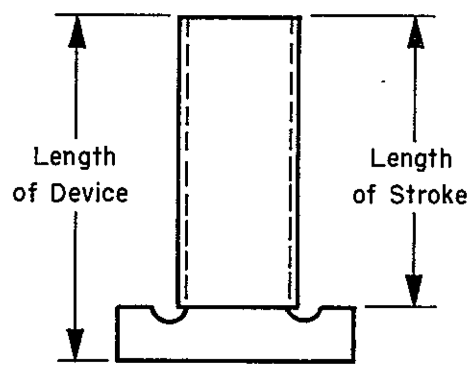
Figure 8.A.5



Extrusion Device



Collapsing Tube Device



Tube-Mandrel Device

Figure 8.A.6

are the frangeable tube, the collapsing tube, a metal strip bending device, a metal shearing device, and an extrusion device. We are also considering other energy absorbing devices which have been invented during the course of the program.

6. Current Research Program

The properties of the selected energy absorbing devices are being developed by analytical and experimental analysis. The knowledge generated by these studies will be used in the design of highway buffers. Scale models of these highway buffers will be built and tested.

a. Experimental Studies

Experimental evaluations of the energy absorbing devices are being made using both quasi-static and dynamic tests. The quasi-static tests are performed with a hydraulically operated testing machine equipped with electronic instrumentation which records forces and deflections. This equipment, with the exception of the electronics, is standard equipment in most materials testing laboratories. Dynamic testing is also required to define the dynamic characteristics of absorbing devices. A unique dynamic testing machine was designed and constructed for this purpose. This testing machine is equipped with a 35 lb. ram which is accelerated to speeds of up to 60 mph by a specially designed pneumatic cylinder. Directional control of the ram is provided by shock mounted linear bearings which operate on hardened rods. The testing machine is equipped with a test specimen mounting table which is supported on shear pins which release the table in the event of an overload; the energy in the system is then absorbed by a hydraulic shock absorber positioned under the table. Electronic instrumentation provides a record of the velocity of the ram before impact with the energy absorbing device and the deceleration of the ram as a function of time during the operation of the energy absorbing device. Additional instrumentation has been developed which gives force-time data. High speed photography (400-3500 frames per second) is used when necessary, to observe the energy absorbing devices during operation.

b. Scale Model Testing

Considerable advancement in the field of highway vehicle attenuators has been made through the use of full-scale tests. However, these tests are quite expensive and time consuming to set up and run. Much of the data which is required for the development of highway vehicle attenuator systems can be obtained from tests with scale models. Some scale model testing has been done of two auto collisions by Emori at UCLA (7). Scale model tests were also conducted concerning the interaction of autos with the GM redirecting median barrier by Jurkat and Starrett of Stevens Institute of Technology (8). The information obtained from such scale model tests can be directly related to the behavior of a full-scale prototype by observing appropriate scaling laws and similitude requirements. Scale modeling has been used extensively in the design of aircraft and is presently being used rather extensively in the field of explosive metal forming.

A scale model testing facility has been constructed at the University of Denver for the purpose of evaluating highway vehicle attenuation systems. This facility is shown in Figure 8.A.7. The facility consists of a table on which scale model autos are accelerated to desired speeds and directed into scale model highway buffers. It is equipped with electronic and high speed photographic instrumentation for recording the crash data. A high speed camera is mounted above the table and can be seen in Figure 8.A.7. Other cameras can be mounted on tripods near the table. A velocity measuring system employing light beams and photo transistors is used to measure the velocity of the auto prior to impact with the attenuator. Accelerometers and load cells are available for measuring accelerations and forces in the interaction. The scale model auto is launched with a pneumatic catapult. The present launching system is capable of launching the auto straight ahead only; future plans call for a launching system which can launch the model with both forward and rotational velocities and also simulate swerving maneuvers prior to impact.

A more complete description of the use of scale modeling in the study of vehicle-attenuator interactions is presented in the following article.

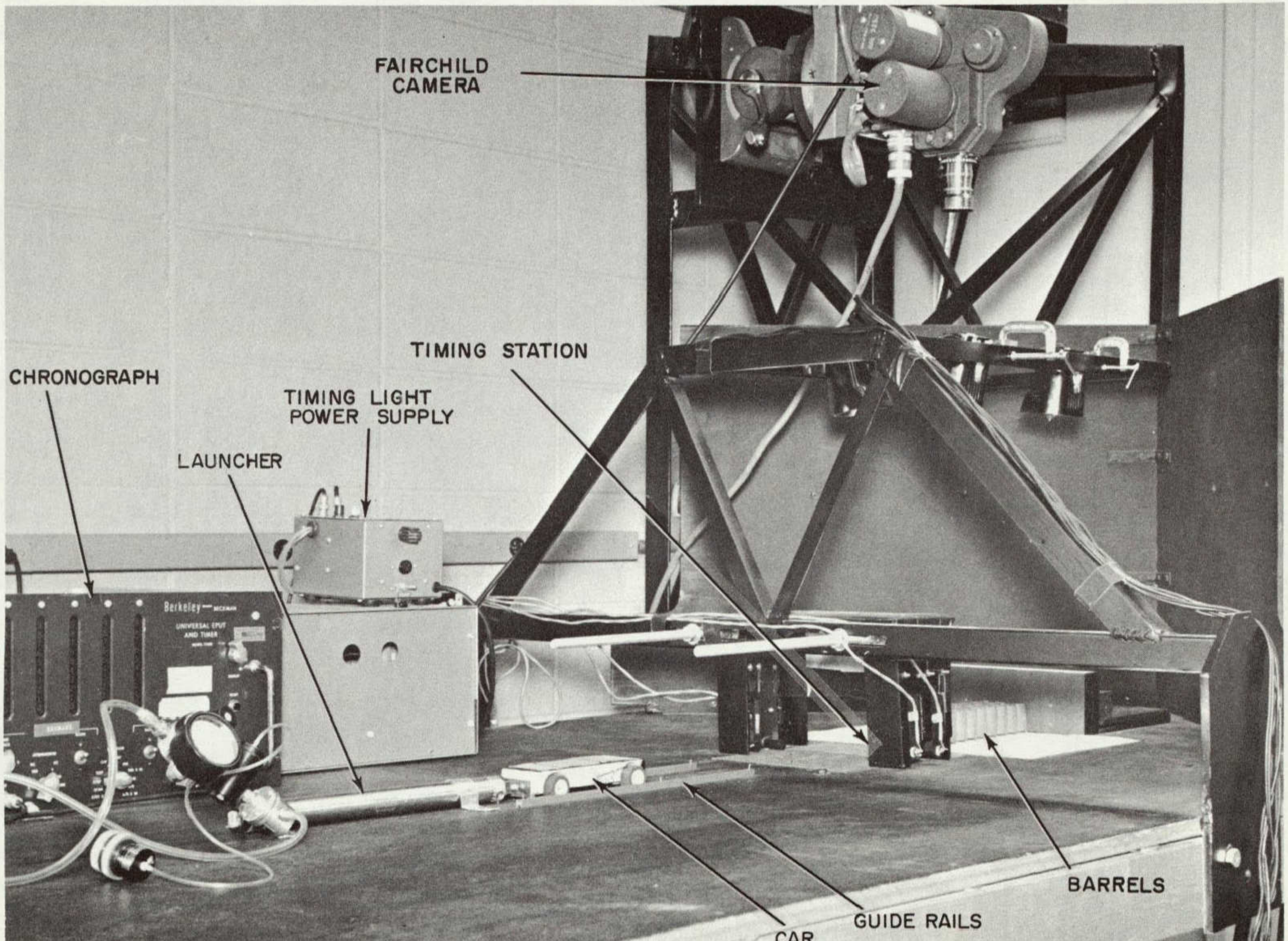


Figure 8.A.7 Overall View of Model Crash Barrier Test Facilities.

B. SCALE MODEL TESTING

1. Introduction: Scaling Law and Similitude Requirements

A physical phenomenon, y, may depend on several independent variables $x_1, x_2, \ldots, x_n$ in some unknown manner. In general a functional relationship exists between y and the independent variables which can be expressed as

$$y = f(x_1, x_2, \ldots, x_n) \tag{1}$$

This could be the mathematical expression of a physical law governing the dependence of y on the independent variables, $x_1, x_2, \ldots, x_n$. This physical law is valid regardless of the units of measurement and applies equally well to the model and the prototype. Since the variables in the physical problem (excluding electromagnetic phenomena) can be described in terms of the four basic dimensions of mass, length, time and temperature, then according to Buckingham's Pi Theorem, the $n + 1$ variables in equation (1) can be combined into exactly $n + 1 - k$ dimensionless groups, where k is the number of basic dimensions used to define the n variables. Therefore, equation (1) can be put into the dimensionless form

$$\pi_1 = f(\pi_2, \pi_3, \ldots, \pi_{(n+1-k)}) \tag{2}$$

The physical law expressed in equation (2) is the same for the model and the prototype. If each of the dimensionless variables on the right side of the equation is the same for the prototype and the model, then π_1 will be the same also. This equality between π_1 for the model and for the prototype defines the scaling law for the dependent variable, y. The requirement that the π terms on the right side of the equation be the same for the model as the prototype determines the similitude requirements for the independent variables.

Using the subscripts m and p to refer to model and prototype, the scaling law and similitude requirements are as follows:

$$\text{Scaling Law:} \quad \pi_{1p} = \pi_{1m} \tag{3}$$

Similitude Requirements:

$$\begin{aligned} \pi_{2p} &= \pi_{2m} \\ \pi_{3p} &= \pi_{3m} \\ &\cdot \\ &\cdot \\ &\cdot \\ \pi_{(s)_p} &= \pi_{(s)_m} \end{aligned} \tag{4}$$

The number of scale factors which may be chosen arbitrarily is equal to the number of basic dimensions in the physical problem. For example, if all the variables in the problem can be expressed in terms of three basic dimensions, i.e., mass, length, and time, three scale factors may be chosen arbitrarily which correspond directly or indirectly to them. The remaining scale factors can be derived in terms of one or more of these factors by using equations (3) and (4).

2. Summary

To demonstrate the validity of scale model testing a scale model test of the "BPR Modular Crash Cushion" was conducted. The test was restricted to a head-on collision of a model car impacting an array of scale model 55-gallon drums. The results were compared to the results of the actual tests conducted by the Texas Transportation Institute. (10, 11)* Specifically, the test modeled by DRI was No. 1146-5, (11) which used a Dodge sedan weighing 3360 lbs., impacting head-on at 52.5 mph into a modular crash cushion. This cushion or barrier consisted of a rectangular array of 16 gage 55-gallon drums, 9 drums long by 3 drums wide, with a 10th row, 2 drums wide, placed at the impact point.

The scaling laws were determined using standard techniques discussed in Section II. For the specific test conducted, emphasis was placed on accurately scaling the dimensions, weight and velocity of the car, and the dimensions, weight and static force-deflection curve for the drums. Briefly, it was found that if the linear dimensions were scaled in the ratio of

$$\frac{\text{Model dimensions}}{\text{Prototype dimensions}} = n_1$$

the velocity would scale as

$$\frac{\text{Model velocity}}{\text{Prototype velocity}} = (n_1)^{\frac{1}{2}}$$

and the forces would scale as

$$\frac{\text{Forces on model}}{\text{Forces on prototype}} = (n_1)^3$$

The results of the test are shown graphically in Section IV. It is apparent from these results that even a simplified model such as that used in this test can give valid results provided strict attention is paid to the governing parameters.

* Numbers refer to references.

3. Dimensional Analysis

The variables assumed to be sufficient to describe the impact problem under discussion are tabulated below.

Dependent Variable	Name	Basic Dimensions
a	Deceleration	LT^{-2}
Independent Variables		
V	Velocity (initial)	LT^{-1}
w	Rotational Velocity (initial)	T^{-1}
I	Polar Moment of Inertia of the Automobile	ML^2
θ	Obliquity Angle at Impact	--
u_1	Coefficient of Friction Between Tires and Road Surface	--
γ_i	Characteristic Dimensions of Automobile	L
F_i	Force to Operate Barrier over the i^{th} Increment of Stroke	MLT^{-2}
L	Stroke of Barrier	L
u_2	Coefficient of Friction Between the Barrier and the Roadway	--
λ_i	Characteristic Dimension of Barrier	L
M	Mass of Automobile	M
ρ	Density of the Barrier Material	ML^{-3}
g	Acceleration Due to Gravity	LT^{-2}
ϕ	Yaw Angle at Impact	--
k	Percent Springback of Barrier	--

Using standard techniques the variables may be combined into 13 dimensionless groups. For the present purpose, these have been taken as follows:

$$\pi_1 = aM/F_i$$
$$\pi_2 = \omega L/V$$
$$\pi_3 = I/M\lambda_i^2$$
$$\pi_4 = \theta$$
$$\pi_5 = u_1$$
$$\pi_6 = \gamma_i/L$$
$$\pi_7 = L/\lambda_i$$
$$\pi_8 = u_2$$
$$\pi_9 = \rho V^2\gamma_i^2/F_i$$
$$\pi_{10} = \phi$$
$$\pi_{11} = gM/F_i$$
$$\pi_{12} = \lambda_i F_i/V^2 M$$
$$\pi_{13} = k$$

The design conditions for a true model are that the π terms for the model equal the corresponding π terms for the prototype, i.e.,

$$(\pi_n)_m = (\pi_n)_p$$

where the m and p subscripts refer to the model and prototype respectively.

There are 3 basic dimensions (mass, length and time) for the variables considered and therefore, 3 scale factors may be chosen arbitrarily which scale these basic dimensions. The scale factors chosen for the present case are

$$n_1 = (\lambda_i)_m/(\lambda_i)_p \quad \text{(scales length)}$$
$$n_2 = g_m/g_p \quad \text{(scales time)}$$
$$n_3 = (F_i)_m/(F_i)_p \quad \text{(scales mass)}$$

For our laboratory, a desirable linear scale for the model is $(\lambda_i)_m/(\lambda_i)_p = n_1 = 1/25$, while the most practical value for the ratio of the gravitational constants is $g_m/g_p = n_2 = 1$. The force ratio is conveniently chosen as $(F_i)_m/(F_i)_p = n_3 = n_1^3$, thus preserving the natural relationship between the length, volume and weight or mass of the model. Using these scale factors and equating the terms given the complete similitude requirements below.

(1) $a_m = a_p$

(2) $V_m = n_1^{\frac{1}{2}} V_p$

(3) $\omega_m = n_1^{-\frac{1}{2}} \omega_p$

(4) $I_m = n_1^5 I_p$

(5) $\theta_m = \theta_p$

(6) $u_{1m} = u_{1p}$

(7) $\gamma_{im} = n_1 \gamma_{ip}$

(8) $F_{im} = n_1^3 F_{ip}$

(9) $L_m = n_1 L_p$

(10) $u_{2m} = u_{2p}$

(11) $\lambda_{im} = n_1 \lambda_{ip}$

(12) $M_m = n_1^3 M_p$

(13) $\rho_m = \rho_p$

(14) $g_m = g_p$

(15) $\phi_m = \phi_p$

(16) $k_m = k_p$

These values (with $n_1 = 1/25$) were used in the model test described in the following sections.

4. Facilities

a. Model Car

The car used in these tests (Figure 8.B.1) was modeled to a scale of 1:25 and was specifically weighted to simulate the 3360 lb. Dodge used in test 1146-5. (11) The scale factors for the car's mass, length, width and velocity were:

$$M_m = n_1^3 M_p = M_p/15,600$$

$$L_m = n_1 L_p = L_p/25$$

$$W_m = n_1 W_p = W_p/25$$

$$V_m = n_1^{\frac{1}{2}} V_p = V_p/5$$

As a first cut at the problem of modeling the impact on a modular crash barrier, the linear dimensions, the weight and the velocity of the car were given primary consideration. Other factors such as tire friction, tire spring constant, ground pressure, and center of gravity were not considered significant due to the fact that the test was confined to a straight, head-on impact where it could be assumed that these factors would play a negligible role. It was believed that this model was the simplest one which could be relied upon to produce valid data and would therefore require the least expenditure of time and money to construct and operate.

One factor which was deemed to be of significance was the resistance of the car to motion with the motor turned off and the gear shift lever in the drive position (automatic transmission) or in high gear (standard transmission). In the actual test, (1146-5) (11) the motor is turned off (brakes not applied) just prior to impact causing the rear wheels to run against the compression of the motor down to about 20 mph for an automatic and all the way down to a dead stop for a standard drive. How far the car rebounds depends on these factors since, a car with automatic transmission, in drive for example, will roll backward rather easily while a car with standard transmission engaged in 3rd, say, offers a very great resistance to any force trying to accelerate it backwards. Thus, it was felt that for purposes of initial testing some method of braking the model in the direction of rebound should be incorporated. This was done by fastening a wire to each side of the car (Figure 8.B.1) in such a way that the front wheels could not rotate backward. This also took into account some of the differences in the

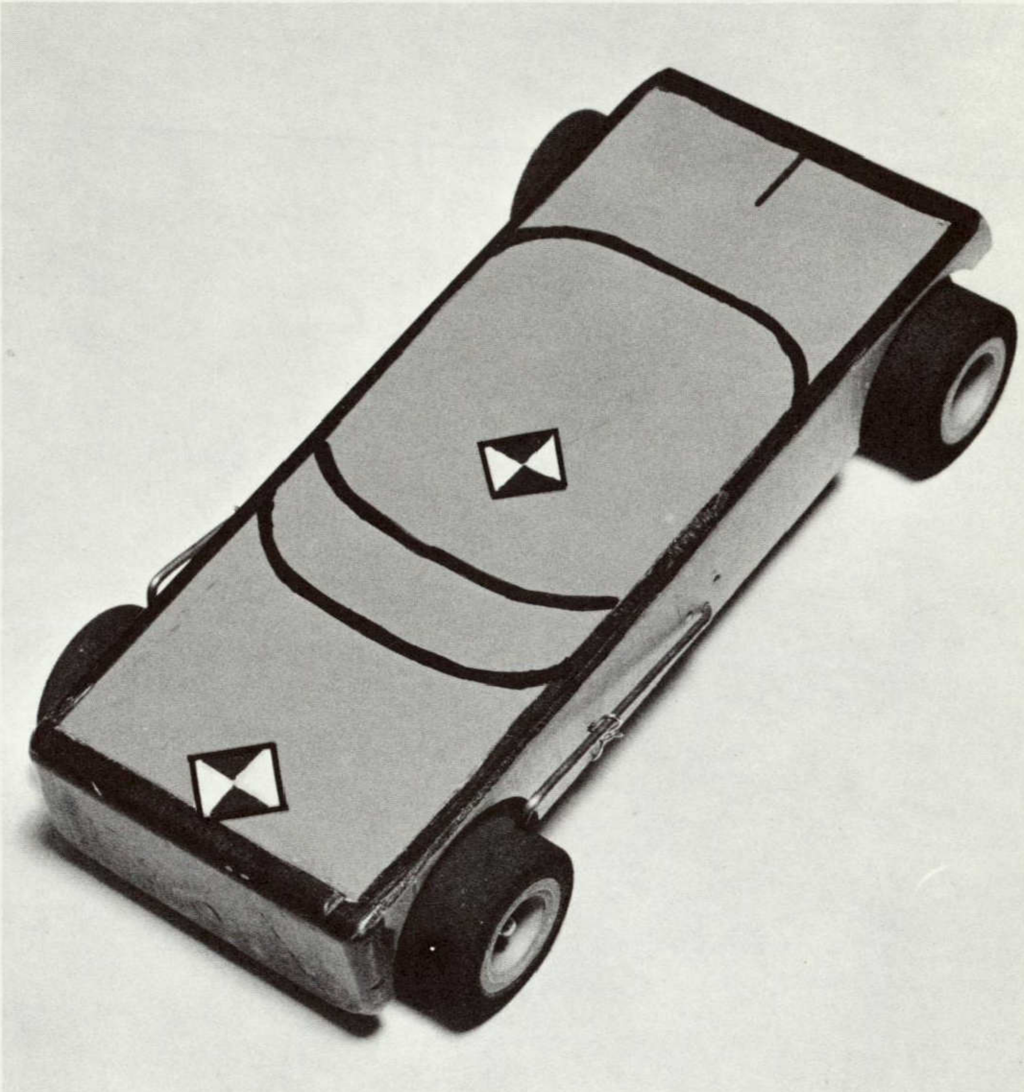


Figure 8. B. 1 Scale Model Car. The Car Measures 6 3/8" × 2 1/2". The Targets on the Car (Black and White Triangles) were Added After the Test Described in This Paper to Facilitate More Accurate Measurements from the High Speed Film.

road surface used in the two tests which in the case of the model was simply a wooden table top covered with grid paper.

The car was constructed of a solid block of redwood with axle holes drilled completely through the block front and rear. To model the weight correctly it was necessary to hollow out a large portion of the underside. The axles were one piece and threaded on both ends. The wheels were a common type used for "slot cars," and were attached directly to the axles.

b. Drums

Using the simulitude requirements discussed earlier, models of the 16 gage, 55-gallon drums were constructed, (Figure 8.B.2).

The mass, static peak load, diameter, height, density and springback of the model drum was, respectively, given by

$$M_m = n_1^3 M_p = M_p/15,600$$

$$P_m = n_1^3 P_p = P_p/15,600$$

$$D_m = n_1 D_p = D_p/25$$

$$H_m = n_1 H_p = H_p/25$$

$$\rho_m = \rho_p$$

$$K_m = K_p$$

The material selected was dead soft aluminum foil (since the springback characteristics of the real drums was not known but was assumed to be small) .003" thick which was formed into cylinders (1" dia. × 1.4" long) by wrapping the material around a 1" dia. tube and gluing the seam with contact cement. The top was modeled by gluing an aluminum strip 0.005" thick by .1" wide, over each end of the model barrel. The model was then compressed in a static testing machine to obtain the load deflection curve. The results of this test are shown in Figure 8.B.3, which also shows a plot of the load deflection curve of an actual drum scaled down to model dimensions. The data shown in Figure 8.B.3 shows that the load deflection curve for the model has the same prominent features as the prototype, these being the sharp initial slope and the peaking effect with the subsequent rapid decrease in the load with continued increase in displacement. (Note that these curves were both obtained from only one test. To properly characterize the barrels a

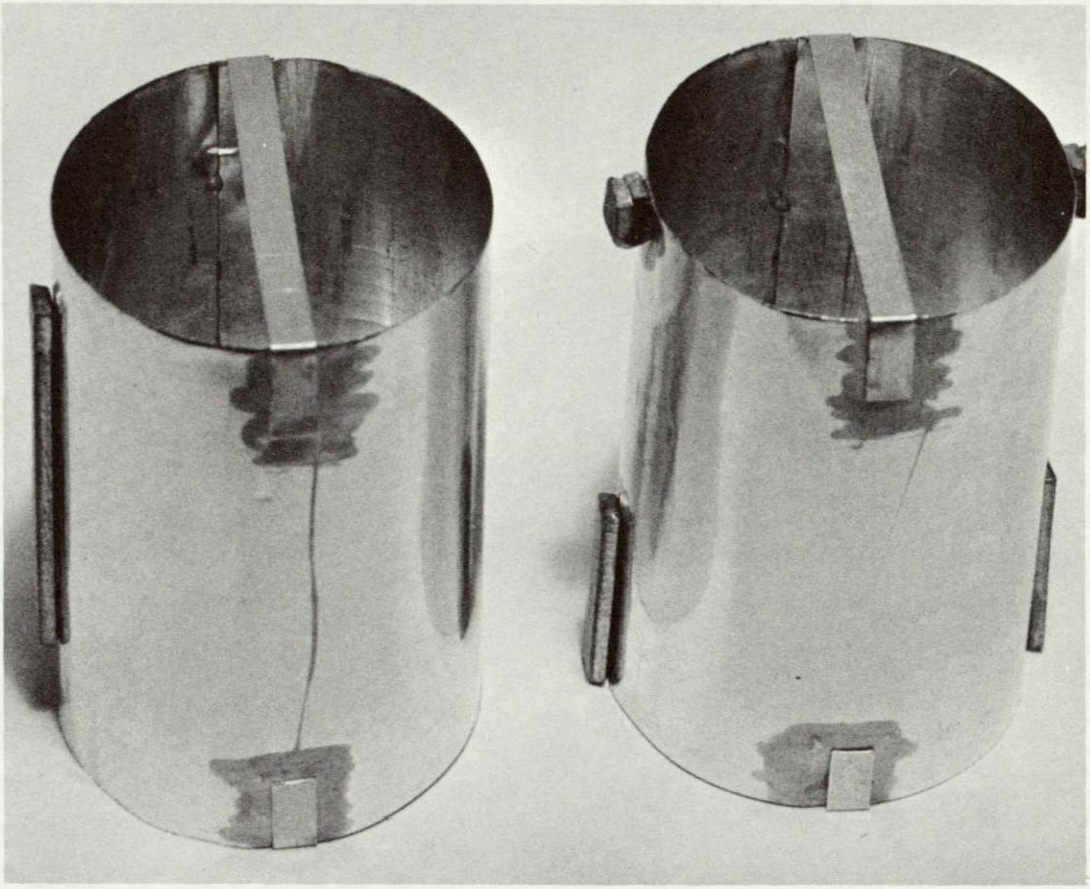


Figure 8. B. 2 Scale Models of the 55-Gallon Drums Used in the BPR Modular Crash Cushion. A "Center" Drum is Shown on the Right and an "Outside" Drum on the Left. Note Small Lead Weights Used as Spacers on the Center Drum. (This Photograph was Taken Prior to Painting the Barrels).

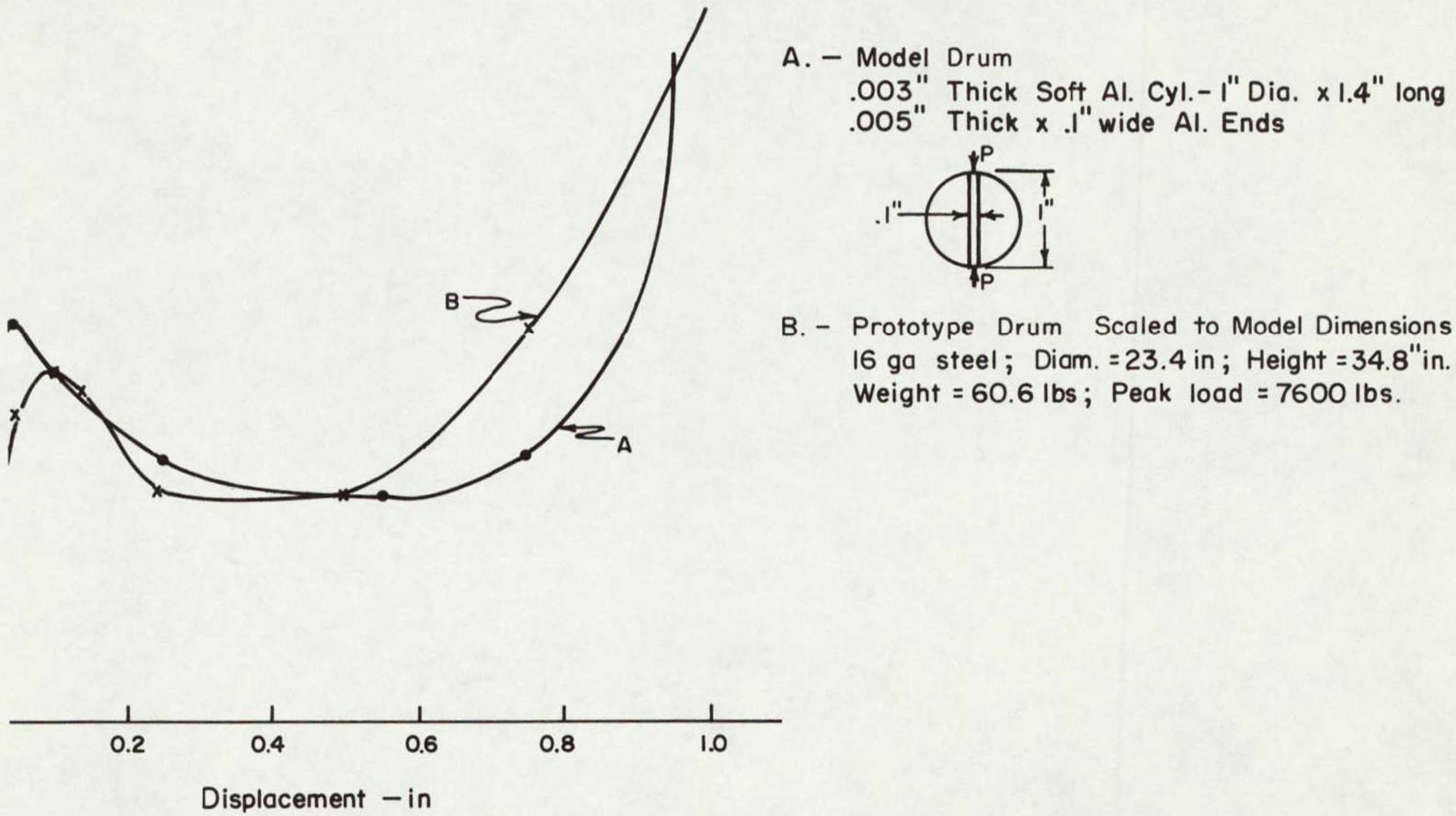


Figure 8. B. 3 Plot Showing the Relationship of the Load vs Deformation Curves for the Model and Prototype Drum.

statistical sample would be required). It was found that considerable change in the peak load for the model could be achieved by varying the width of the end strip, making it possible to simulate a large number of designs. Furthermore, as indicated by the results of the present work, the load-deflection curve for the model drum need not be an exact duplicate of the prototype curve, but must retain the essential character of the curve, i.e., initial slope, etc. It is, therefore, reasonable to assume that the top of the model drum need not duplicate, physically, the top of the prototype.

The drums used in the model test incorporated spacers which simulated rolling hoops. These were formed by gluing lead weights to each side of the center barrel, in each row. This provided the necessary space, between the center and outside drums for the wire used to simulate the steel cable used in the actual test. Also, the lead weights were necessary to bring the weight of the models up to the necessary values since the aluminum, alone, was too light. The weights which were used on the outside drums also served to raise them off the table in a manner similar to that of the "Re-Bar" chairs used in the actual test. Thus, the model 55-gallon drums, individually as well as collectively, showed a close similarity to the actual drums.

c. <u>Model Car Launching Facilities</u>

The model car discussed previously impacted the barrier at 10.6 mph (equivalent to 53 mph for the prototype). This velocity was attained by a compressed air launcher operating at 150 psi air pressure. (Figure 8.A.7). The car was maintained on course for the first 12 in. by two guide rails. On leaving the guides the car passed through the timing station where it interrupted two light beams, starting and stopping a Beckley (100,000 cps) chronograph. The time required for the car to traverse the distance between the light beams was used to determine the velocity of the car.

The motion of the car and the barrier was recorded by a 16mm high-speed camera. This camera, a Fairchild Model HS101A, capable of film speeds up to 10,000 frames per second, is shown in Figure 8.A.7 mounted to take a plan view of the collision. From the film obtained in this fashion an accurate analysis was made of the motion of the car and the barrier. See Figure 8.B.4. A mirror was used to give a side view of the event making it possible to observe vertical motion of the car and drums. This side view can be seen in Figure 8.B.4. Also refer to the photographs in Figure 8.B.5.

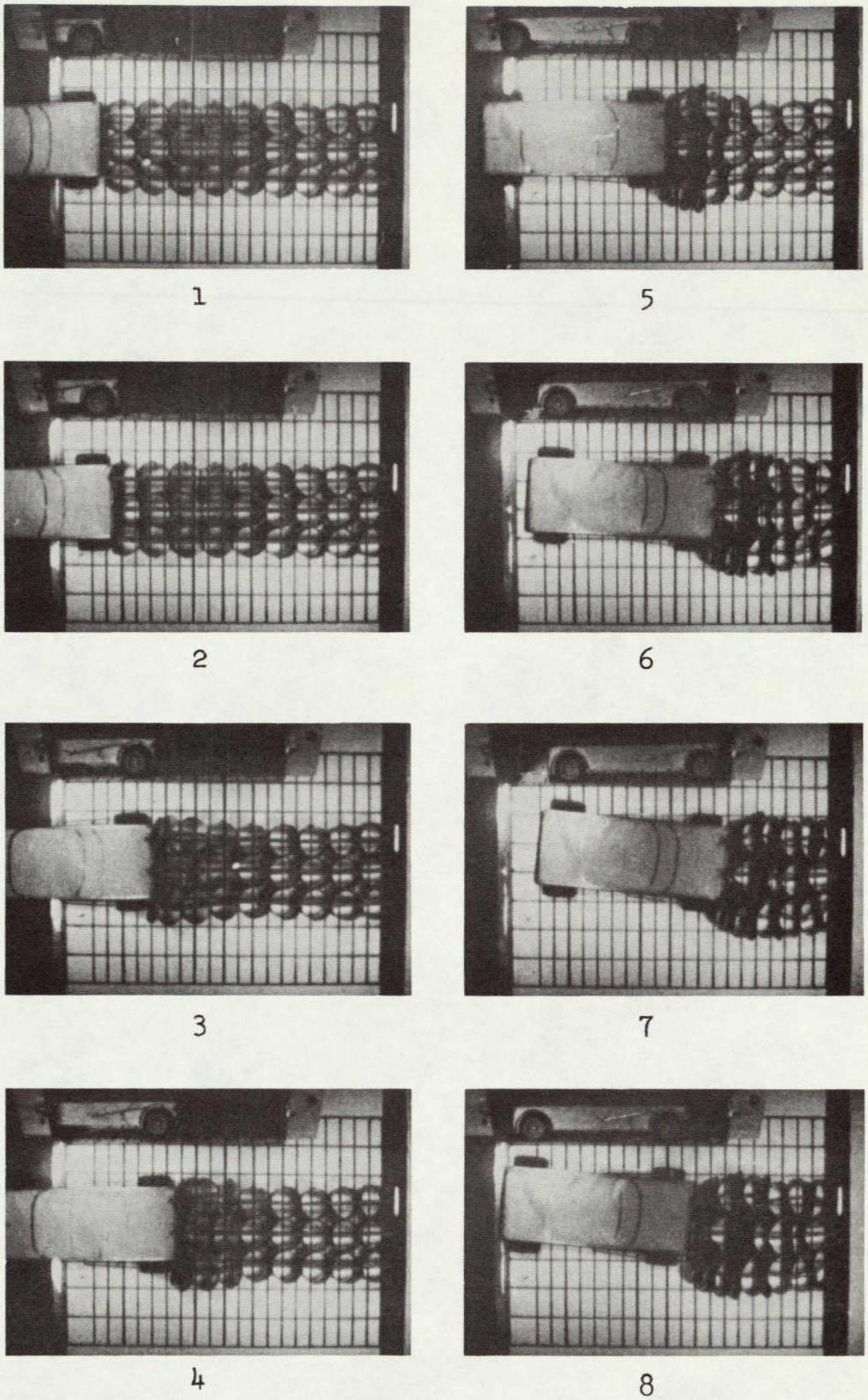
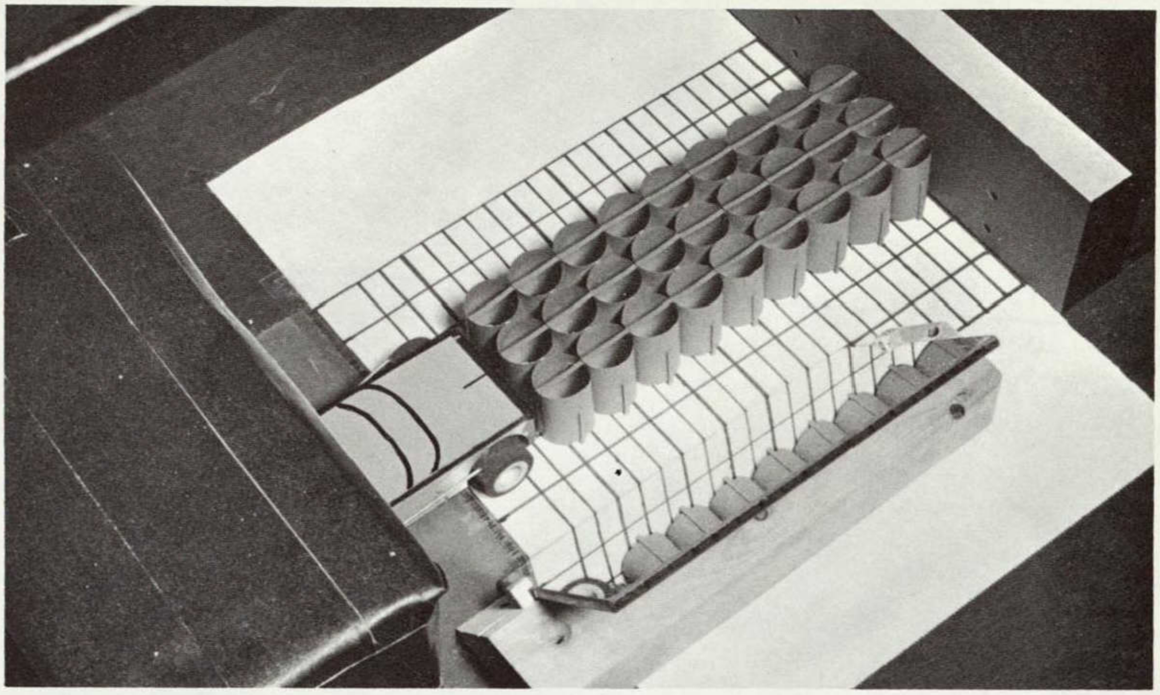
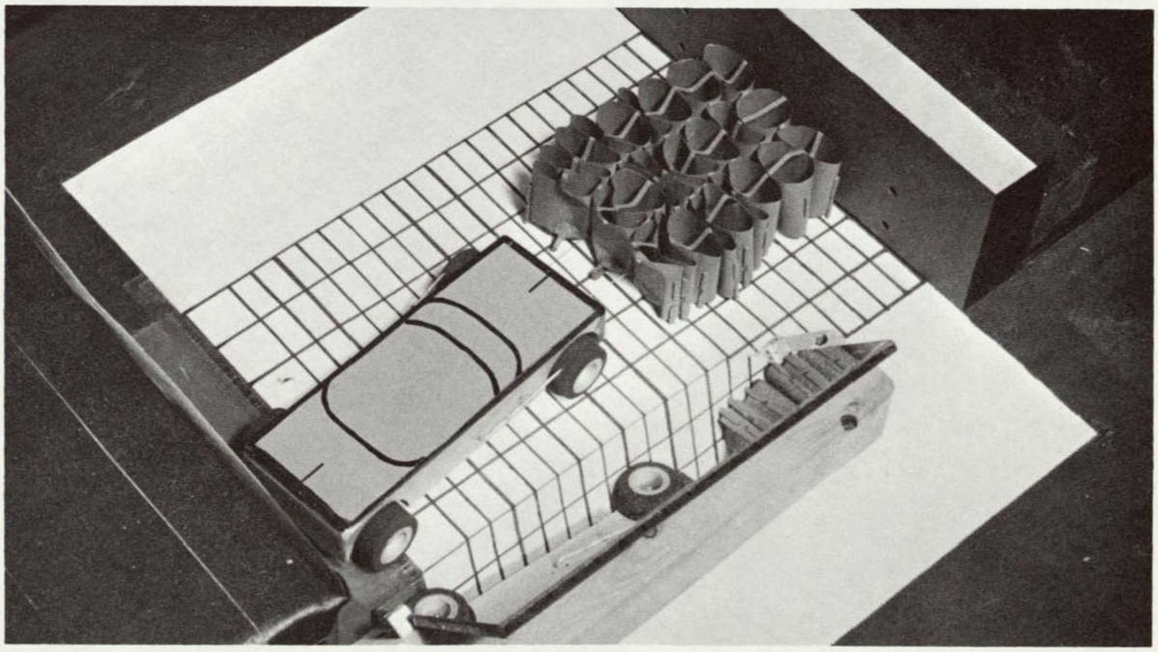


Figure 8. B. 4 Sequence Taken from 16mm High Speed Movie Film (3, 000 fps), During the Scale Model Test of the BPR Modular Crash Cushion.



(a)
Car and Barrier at Impact. (Simulated)



(b)
Car and Barrier After Impact. (Simulated)

Figure 8. B. 5

d. Results

The results obtained in the test just discussed are presented graphically in Figure 8.B.6. The distance, velocity, and G curves are plotted as a function of time after impact for the model car. Also, the distance-time plot for the prototype (1964 Dodge - 3360 lbs.) is shown to the scale of the model. Because of the good correlation of the model data to the prototype data, only the data for the model was actually differentiated to produce the velocity and G curves. The average deceleration in G's for the model was computed from the equation (11)

$$G_{ave} = \frac{V_0^2}{2gS}$$

where

V_0 = initial velocity at time of impact

S = movement of the vehicle's center of gravity in feet from its position at first contact to its position when its longitudinal velocity is zero.

g = 32.2 ft/sec

Thus, for the model

$$(G_{ave})_m = 7.35$$

while for the prototype the value was given as (11)

$$(G_{ave})_p = 7.6$$

A further indication of the validity of the test can be obtained from a visual comparison of the barrier after the test to the "after" photograph of the prototype barrier. (Both cases are for the rectangular array of 3 × 9). In these photos (Figure 8.B.5 in this report, photo on page 10 of Reference 10, and Figures 6 and 11 of Reference 11) the first two rows appear to crush somewhat uniformly while thereafter there is a distinctive pattern wherein alternate rows do not crush uniformly. This behavior is very evident in the high-speed movie (3000 fps) taken during the model test. An explanation for this nonuniform behavior lies in the manner in which the force is applied to successive elements of the barrier. The first two rows undergo catastrophic deformation due to the relatively high impact velocity. By the time the car travels

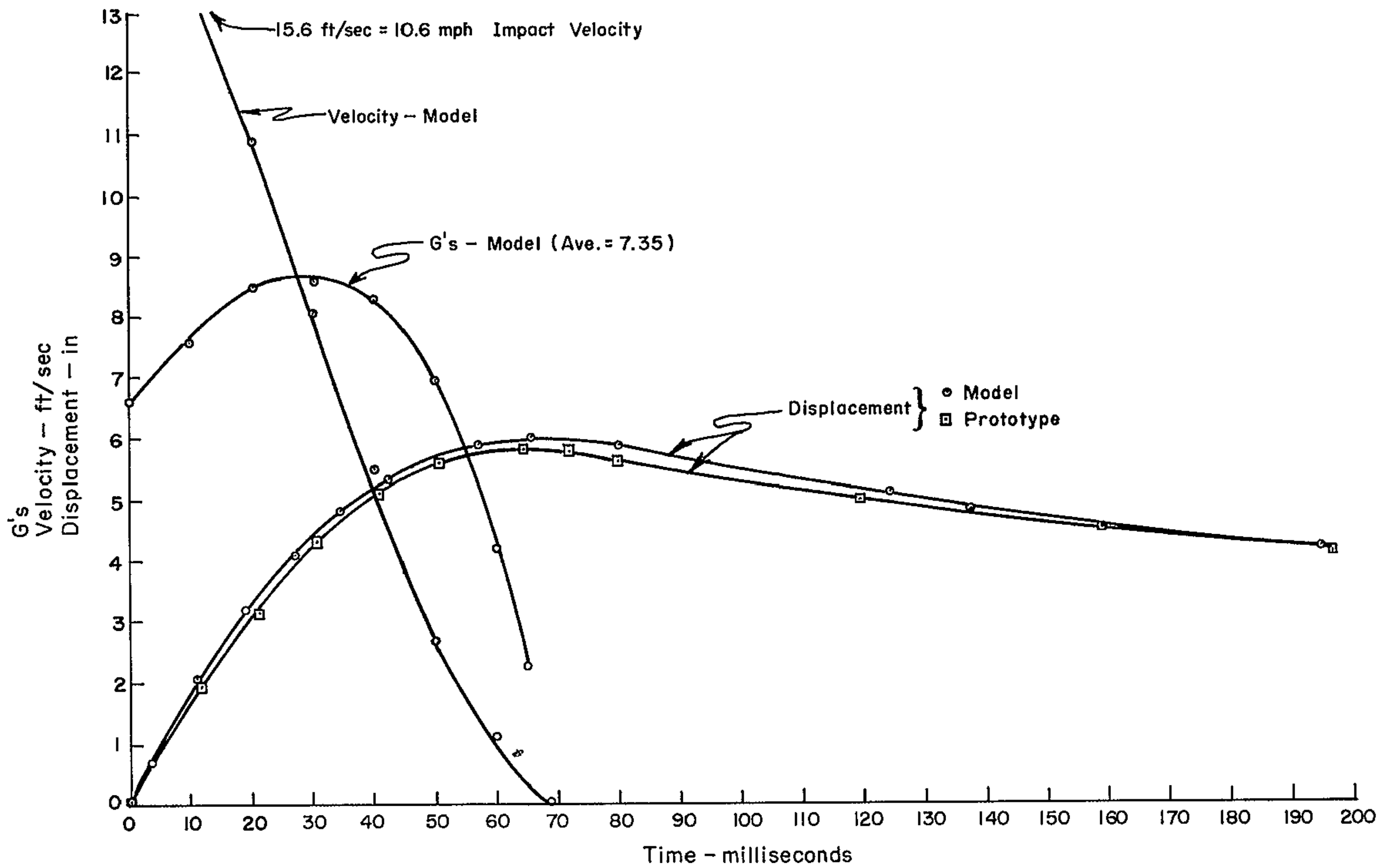


Figure 8.B.6 Results of the Scale Model Impact Test Compared to the Results of the Actual Test (1146-5).(11) The Actual Test Data Has Been Scaled Down to the Model Dimensions of Distance and Time.

approximately the distance equivalent to these two drum diameters the second row has wrapped back, around the third row, causing the load to be spread out over a much greater area. This row, seeing a more distributed load is not as easily crushed. (Note that the kinetic energy of the car at this time has decreased by about 30%). The fact that the third row is not crushing as rapidly implies that the fourth row is now being affected by a more concentrated load and it therefore deforms more readily. This chain of events continues until, in the end, it appears that every other row is affected in this manner. Similar behavior, but to a lesser extent, may be seen in Figure 9 of Reference 11 (Test 1146-3). However, this last set of barrels was originally a slightly different array which may account for the more uniform deformation.

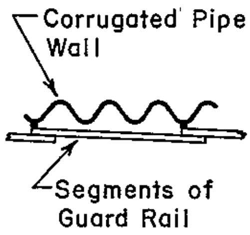
The fact that the drums behave in this nonuniform manner indicates clearly an important role which modeling can play in barrier design. That is, a crash cushion can be studied experimentally to determine changes in design required to obtain uniform deformation, thereby applying the most uniform decelerating force to the vehicle over the shortest distance. Simultaneously, the data from all tests may be compiled for use in developing and correlating analytical models. The same techniques may be applied to barriers of any design with the ultimate goal of designing a barrier which is optimized with regard to overall efficiency, including the cost of materials and labor to build and install the barrier. These points and others such as angle impacts and side impacts resulting in auto redirection can be studied experimentally, using models, at a fraction of the cost of using full size cars and barriers. For example, the cost of running a test such as that described in this paper is estimated at less than $400 including direct labor, complete film coverage, materials and data reduction. Some barrier tests would be expected to be considerably less expensive because with the modular barrier the largest expense is involved in construction of the individual elements. Simpler designs would cost less for labor and hence would be less expensive to test and evaluate.

C. VEHICLE ATTENUATION SYSTEMS

As mentioned earlier, a group of three faculty members conceived three attenuation and energy absorption devices. These are described briefly in the following pages.

1. Large Collapsing Tube Vehicle Attenuation System

This device concerns an energy dissipation cushion which can be mounted in front of fixed highway structures such as bridge columns and parapets to provide a controlled deceleration for errant autos which would otherwise impact the structure. The device consists of a large pipe made of corrugated sheet metal such as those which are used for culverts. It is fitted with bulkheads and short segments of overlapping guardrail as shown in Figure 8. C. 1. The bulkhead on the front end of the pipe is rounded slightly and designed to deform so that the impact force is transferred rather uniformly into the corrugated pipe. The kinetic energy in the auto is absorbed by the pipe as it collapses by folding as shown in Figure 8. C. 2. Directional control is maintained during the collapse by guides operating in a submerged rail as shown in the Figure or by tie-down cables. The guardrail segments on the side of the pipe operate in conjunction with the pipe as an energy absorbing guardrail system under the influence of heavy side loads. Under the influence of some impacts, the pipe will operate in both modes simultaneously with part of the pipe collapsing on the side and part of the pipe folding.



TOP VIEW OF DETAIL

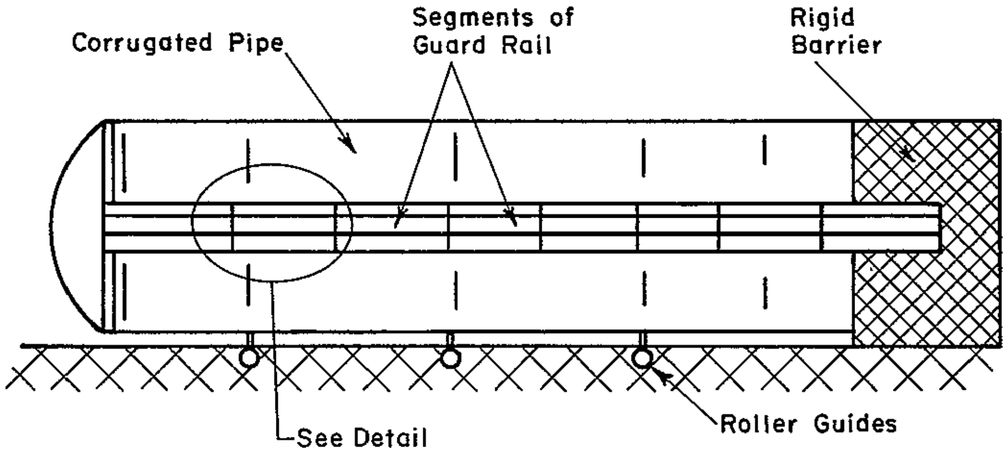
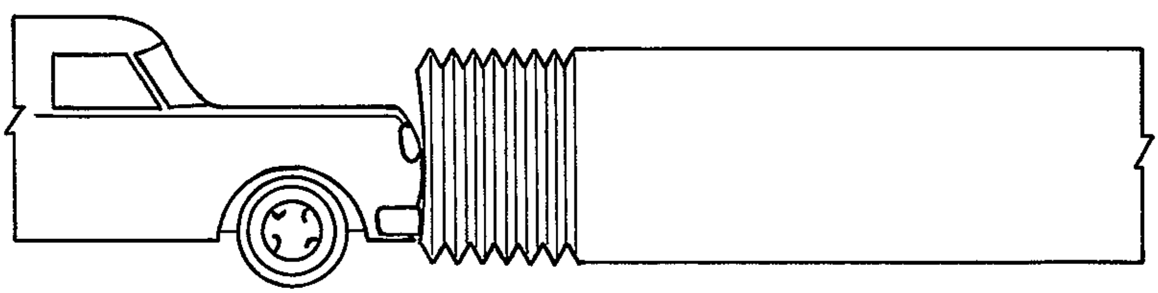


Figure 8.C.1 Side Elevation.



Note: Guard rail segments not shown

Figure 8.C.2 Side Elevation at Impact.

2. Collapsing Tube Vehicle Attenuation System

The accompanying sketches shown in Figures 8. C. 3 and 8. C. 4 illustrate a general system to be used as a safety device. The system serves as an energy dissipation cushion that is installed in front of fixed bridge columns and parapet railings on a highway. Its function is to provide controlled deceleration of a vehicle that would otherwise strike the column or railing.

The system consists of a series of energy devices held in alignment by anchored cables. The energy devices consist of a series of tubes and mandrels or folding tubes oriented in the preferred mode (axially loaded) and sandwiched between diaphragms.

The system is actuated by an out-of-control vehicle striking the front energy device. This device begins to crush at the predetermined, controlled force level. The succeeding devices then crush in order from front to rear, each at its own predetermined force level. The force levels are defined in the initial design on the basis of standards set for the particular highway location and human safety.

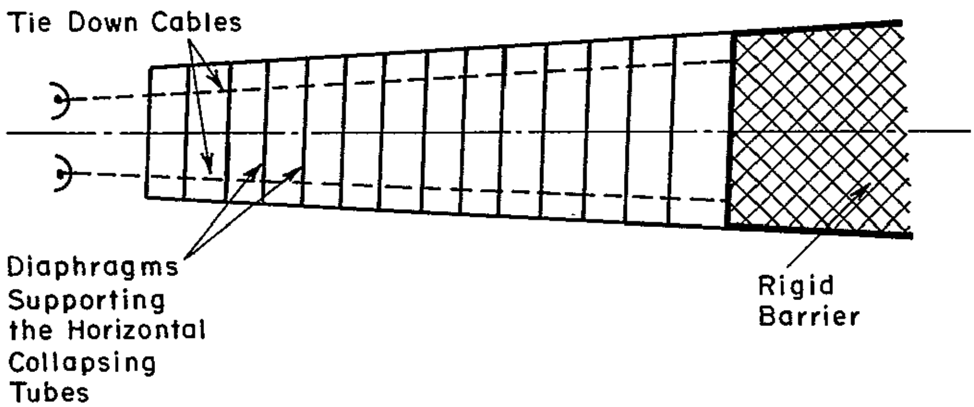


Figure 8.C.3 Top View.

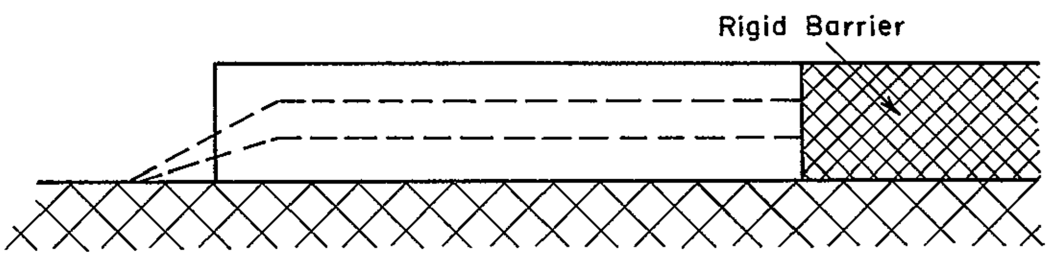


Figure 8.C.4 Side View.

3. Energy Absorbing Device for the Protection of Automobiles From Collisions with Highway Gores

The purpose of the energy absorbing device shown in Figures 8. C. 5 and 8. C. 6 is (a) to absorb the kinetic energy of an automobile before it impacts a highway gore and (b) to limit the acceleration of the automobile to levels which are tolerable by humans.

Description: (See Figures 8. C. 5 and 8. C. 6)

The device is constructed of several corrugated sheet metal strips which are rigidly attached to the concrete gore. The thickness of each strip is varied in order to provide the desired decelerating force. Required thicknesses are obtained by varying the thickness of the sheet metal and/or by using two or more sheets in a single strip. The purpose of the corrugations is to provide increased bending strength.

Features of the device:

a. The force-displacement curve required for a particular gore area is readily obtained by variation of the number, lengths, and thicknesses of the sheet metal strips.

b. There is virtually no deceleration spike at the moment the automobile strikes the energy absorber because the amount of activated mass is quite small.

c. The device tends to wrap around an impacting automobile and, therefore, tends to prevent the automobile from leaving the gore area and reentering the highway in a disabled condition.

d. The device requires no extensive hold-down mechanisms.

e. The device is considerably stiffer near the gore than it is at its nose. Thus, glancing impacts near the base of the device will lead to redirection of the automobile past the gore.

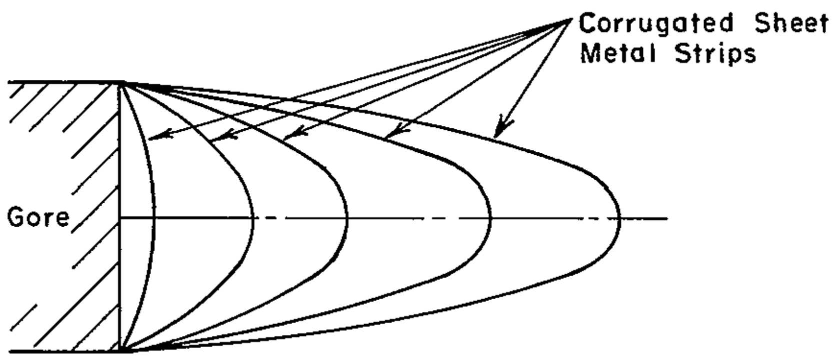


Figure 8. C. 5 Top View.

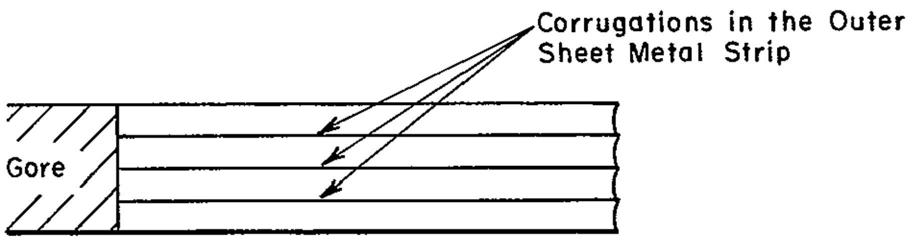


Figure 8. C. 6 Side View

D. GORE AREA BUFFER

1. Introduction to the Problem

The design of a highway safety device to absorb the kinetic energy of a moving vehicle without exceeding force levels tolerable for passengers has been the focus of activity for the gore buffer area design group of the NASA patent project.. The need for such a device was the result of an insufficient original design at the Broadway Exit on the southbound Valley Highway (Interstate 25) in Denver, see Figures 8.D.1 to 8.D.3. Five fatal accidents have occurred at this site since the highway's opening. This is considered to be above the acceptable limits of safety for an Interstate highway.

It was the intention of the design group to arrive at a design that would be applicable to the particular situation at the Broadway Exit gore area but general enough that it would be applied to other situations where there is a high probability of collision with a rigid barrier.

The device has to serve a dual purpose as a highway buffer. First, the device must be able to absorb the kinetic energy of a vehicle during a head-on (direct) collision. Second, the device must be able to redirect the vehicle on to the highway without interfering with the flow of traffic. In the case of redirecting the vehicle, damage to the vehicle must be minimal in order to allow the vehicle to reenter the system while still under control.

Several types of gore area buffers are already available and in use in various locations throughout the country. These include the Fitch Inertial Barrier, Tor-Shok, and a Modular Crash Cushion consisting of a series of 55-gallon oil drums bolted together and held in alignment with steel cables.

The most effective of these barriers at the present is the Modular Crash Cushion barrier design. It has several advantages over the other types since it is composed of standard parts which are easily installed or replaced, is adaptable to many types of configurations and fits within reasonable space limitation because of its adaptable geometry. It also has minor limitations since lateral stability against angle impacts is provided by installing a wire cable which is anchored on the rigid barbier, runs along the rolling hoops on the side of the barrels and then is anchored to the ground at the nose of the device. These cables may provide a force acting below the center of gravity of the car on angle

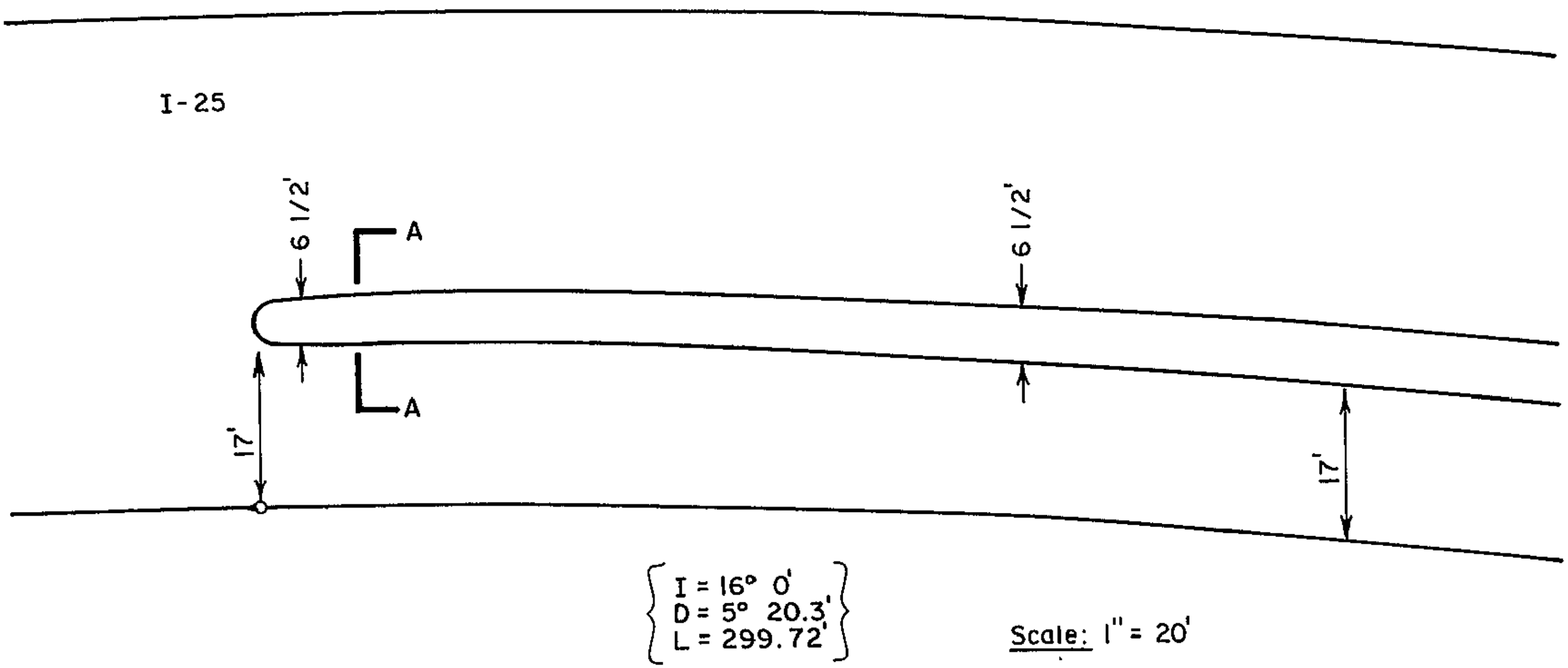


Figure 8.D.1. Plan View of Gore Area (I-25 and Broadway, Denver, Colorado)

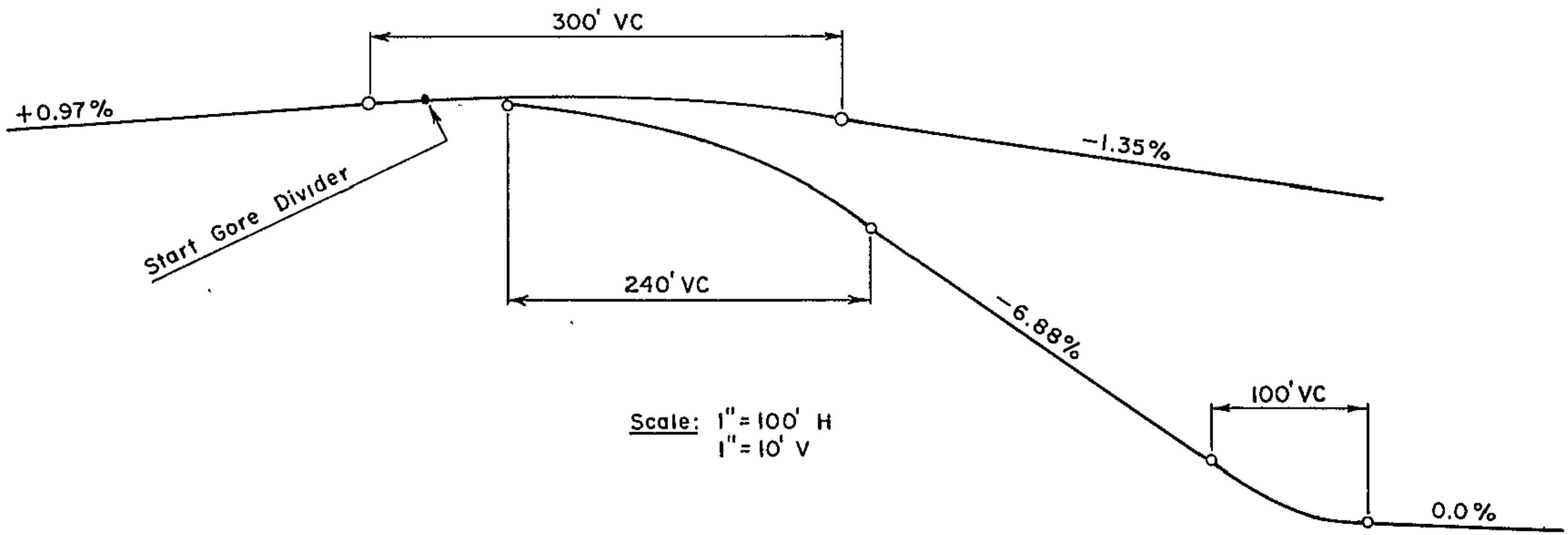
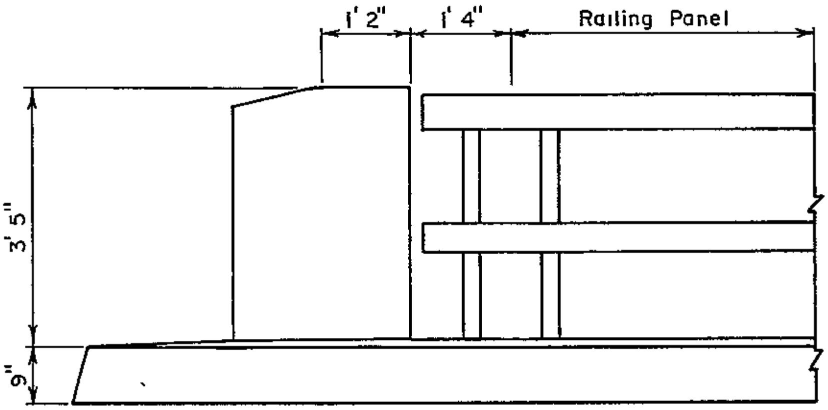
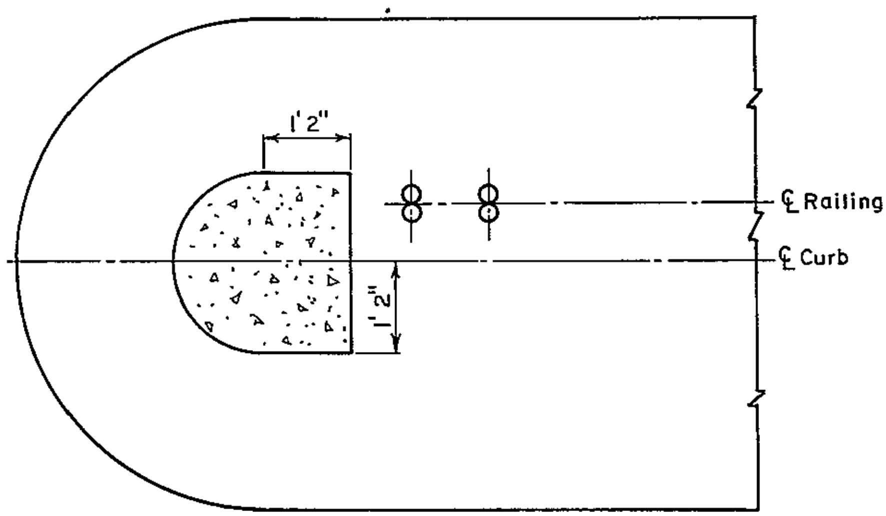


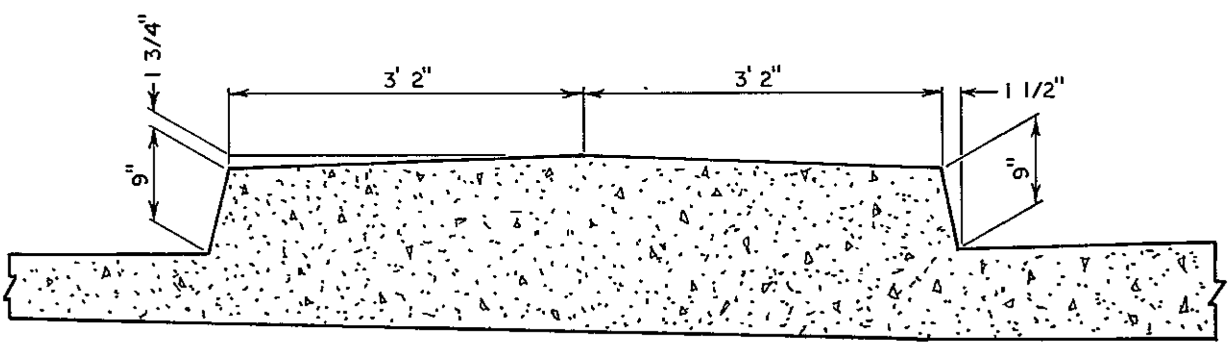
Figure 8.D.2. Profile



ELEVATION



EXISTING END POST PLAN



SECTION A-A (See plan view 8.D.1)

Figure 8.D.3. Details of Existing Gore Area

impacts near the nose of the device, providing a ramping action tending to lift the car off the roadway or overturn the vehicle. A second disadvantage of the barrel device is excessive deflections near the rear of the system, tending to pocket the vehicle and causing excessive decelerations. This may occur under certain angle impacts near the nose of the device. A third limitation of the barrel device (in particular the one recommended for use of the southbound Broadway Exit) was that the total length of the device is nearly 25 feet. Because of space limitations at the site it was felt that a shorter device would be more appropriate, leaving more space for a vehicle to maneuver either on or off the exit ramp area. It should be noted that the total stopping distance required for a vehicle at 60 mph under the current restraints imposed for passenger safety is only 10 feet.

It was decided in attempting to design a gore area buffer to use concepts from both the barrel and other types of devices to arrive at a device that would perform the two required functions (energy absorption and vehicle redirection) and yet remain within the constraints currently recommended for passenger safety. The barrel barrier uses increasingly heavier gauged barrels in order to increase the force level imposed on the decelerating vehicle. This allows the smaller car with relatively small kinetic energy to be stopped by the barrels exerting the smaller forces with an allowable deceleration (determined by penetration into the device). The heavier gauged barrels located at the rear of the device are capable of absorbing larger amounts of energy and hence stop heavier vehicles with larger kinetic energies. The maximum force level obtained with the barrel device is limited since the heaviest gauge barrels collapse under a force of 21K. Width limitations allow the use of only three barrels in a row, limiting the maximum force level applied to 63K. It was decided to continue to increase the force level at the back of the device. This was accomplished by using bulkheads filled with tube and mandrel energy absorbers. The basic design is discussed in detail in the section on design elements. Force-displacement curves and velocity-g values for both the barrel device and the composite barrel and folding tube device are shown in Figures 8.D.4 to 8.D.9. Integrating the area under the force deflection curve gives the amount of kinetic energy absorbed by the device for a given penetration.

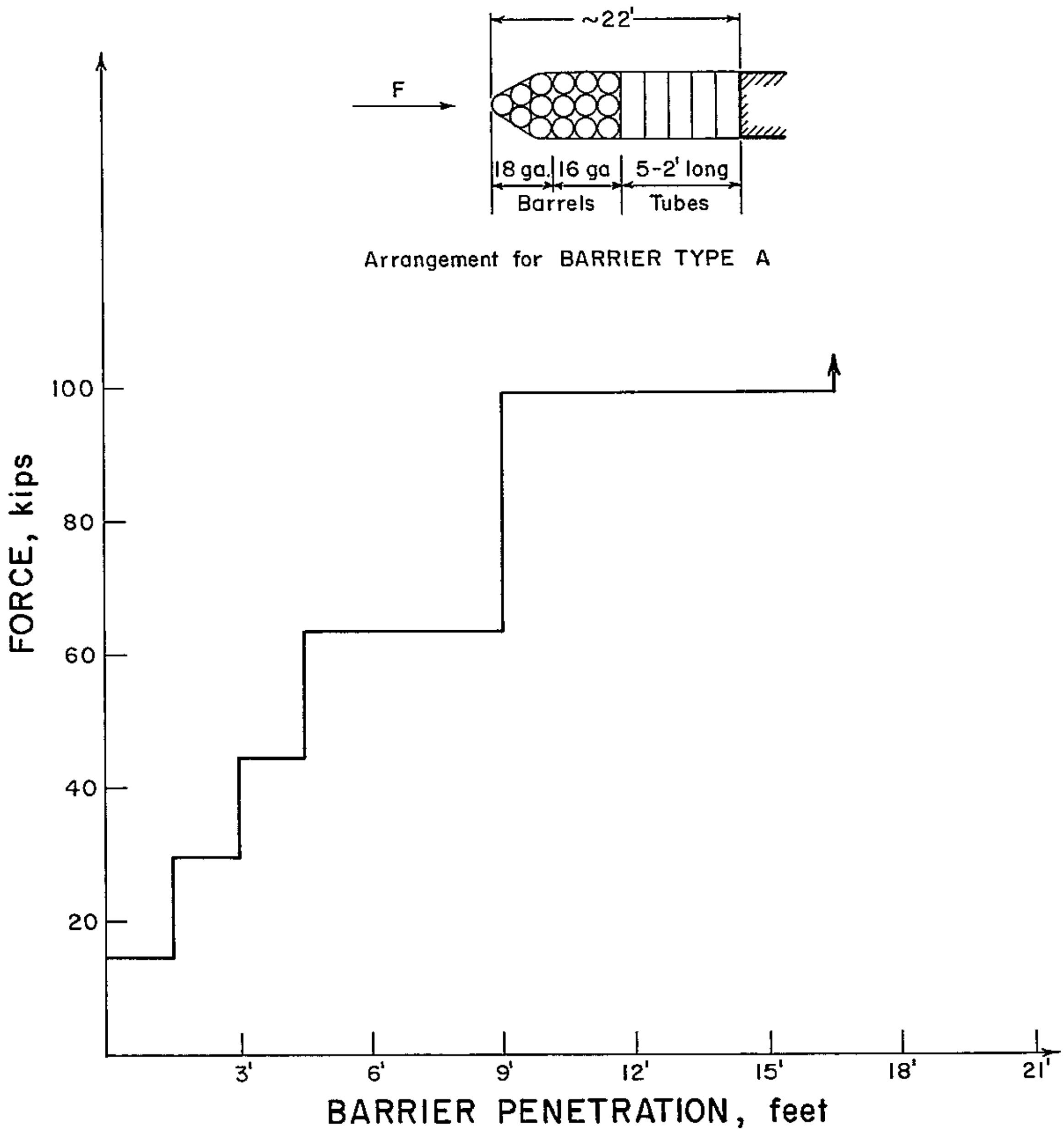
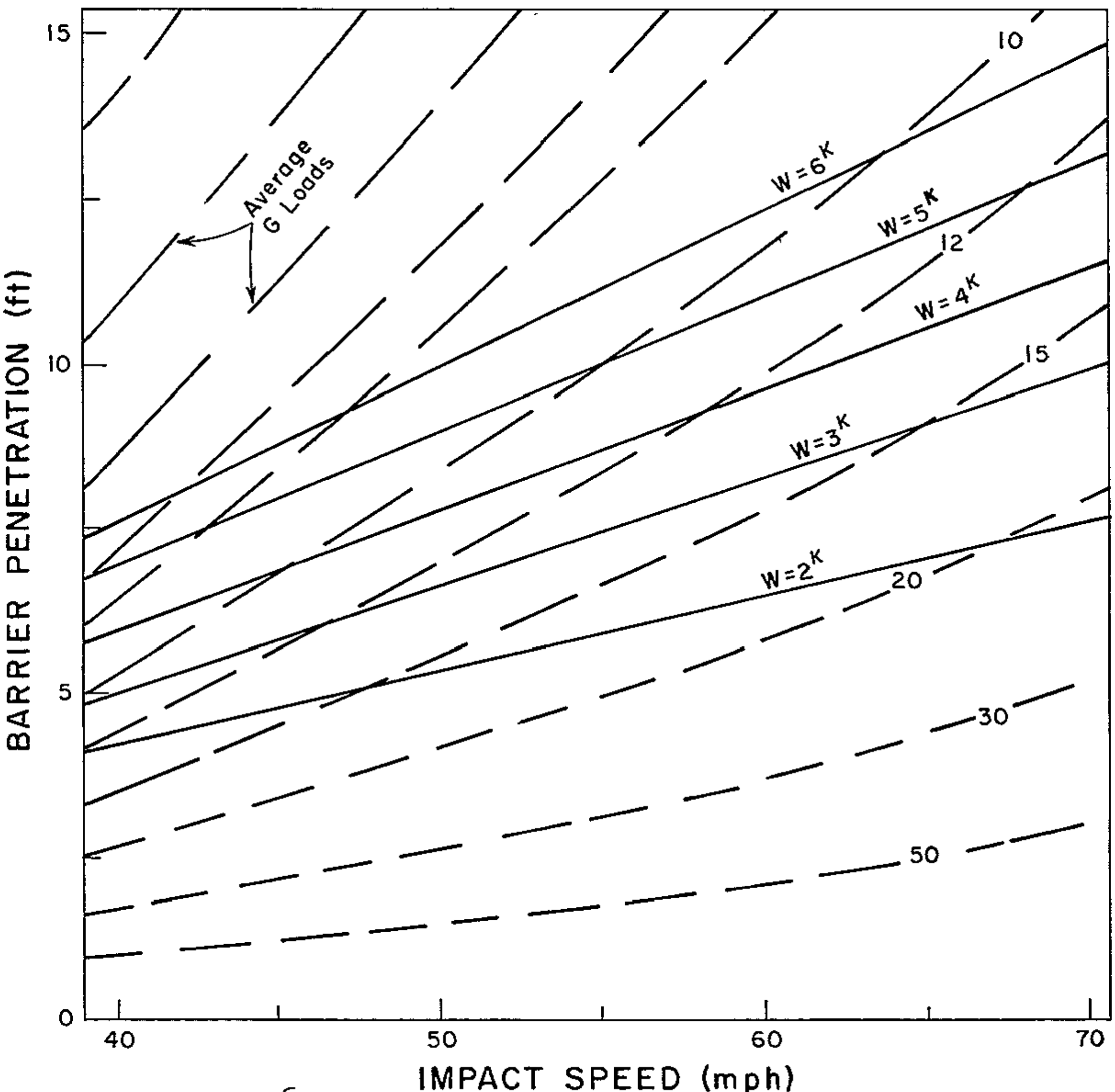
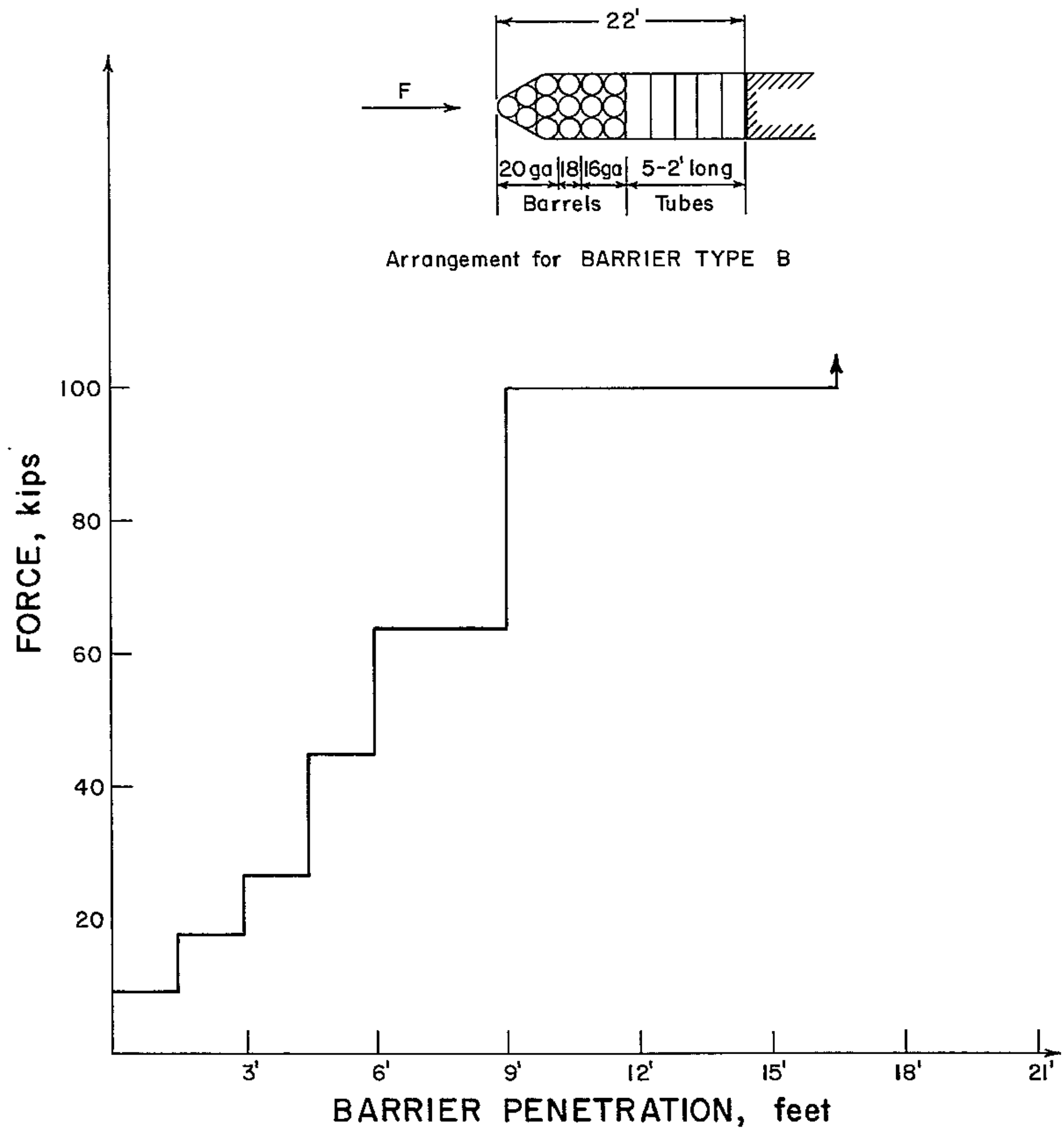

~22'
F
18 ga.
16 ga
5-2' long
Barrels
Tubes
Arrangement for BARRIER TYPE A
FORCE, kips
100
80
60
40
20
3'
6'
9'
12'
15'
18'
21'
BARRIER PENETRATION, feet

Figure 8.D.4



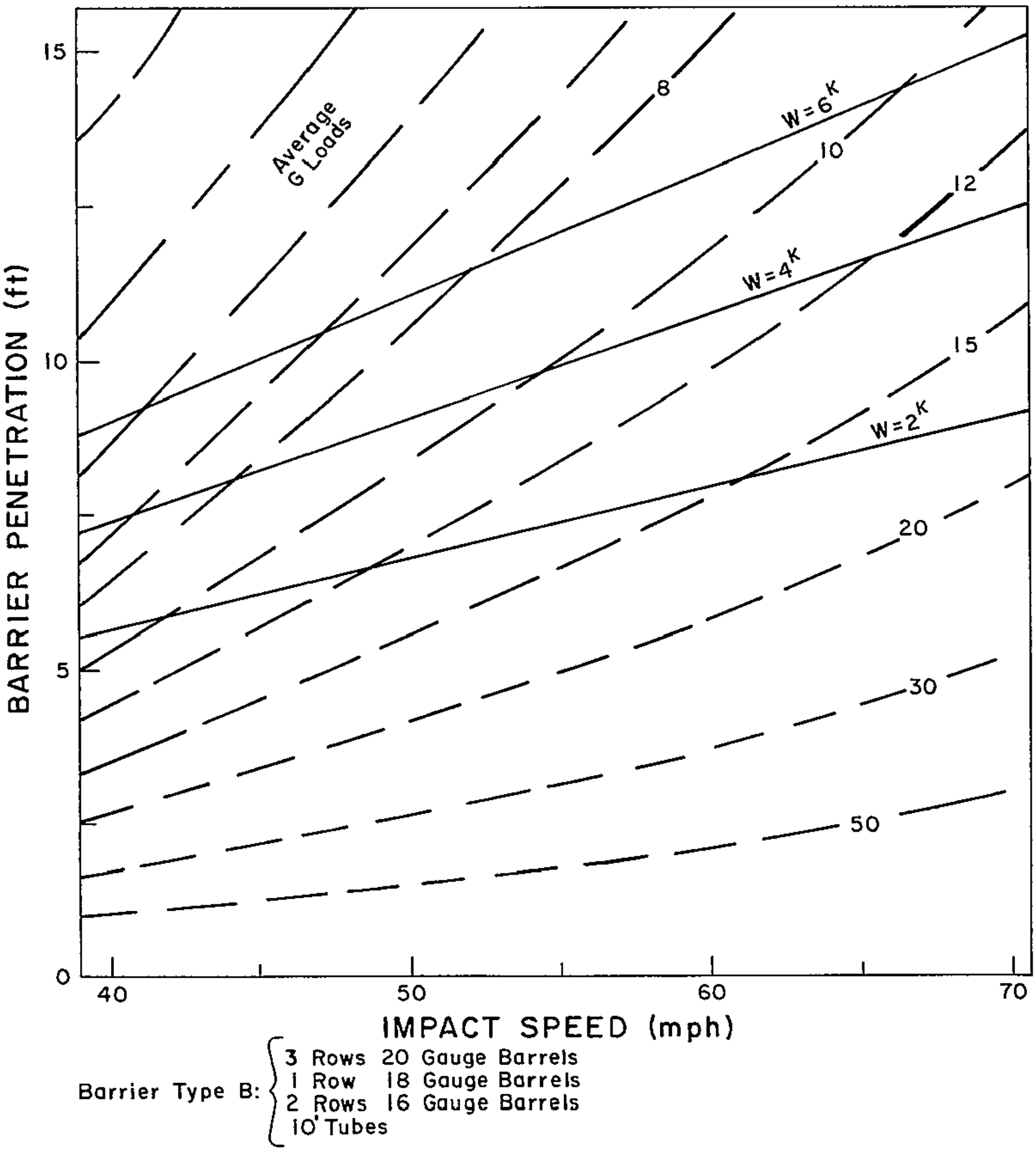
Barrier Type A: 3 Rows 18 Gauge Barrels
3 Rows 16 Gauge Barrels
10' Folding Tubes

Figure 8.D.5



22'
F
20 ga
18
16ga
5-2' long
Barrels
Tubes
Arrangement for BARRIER TYPE B
FORCE, kips
100
80
60
40
20
3'
6'
9'
12'
15'
18'
21'
BARRIER PENETRATION, feet

Figure 8.D.6



Average G Loads
8
10
12
15
20
30
50
W=6K
W=4K
W=2K
BARRIER PENETRATION (ft)
0
5
10
15
IMPACT SPEED (mph)
40
50
60
70
Barrier Type B:
3 Rows 20 Gauge Barrels
1 Row 18 Gauge Barrels
2 Rows 16 Gauge Barrels
10' Tubes

Figure 8.D.7

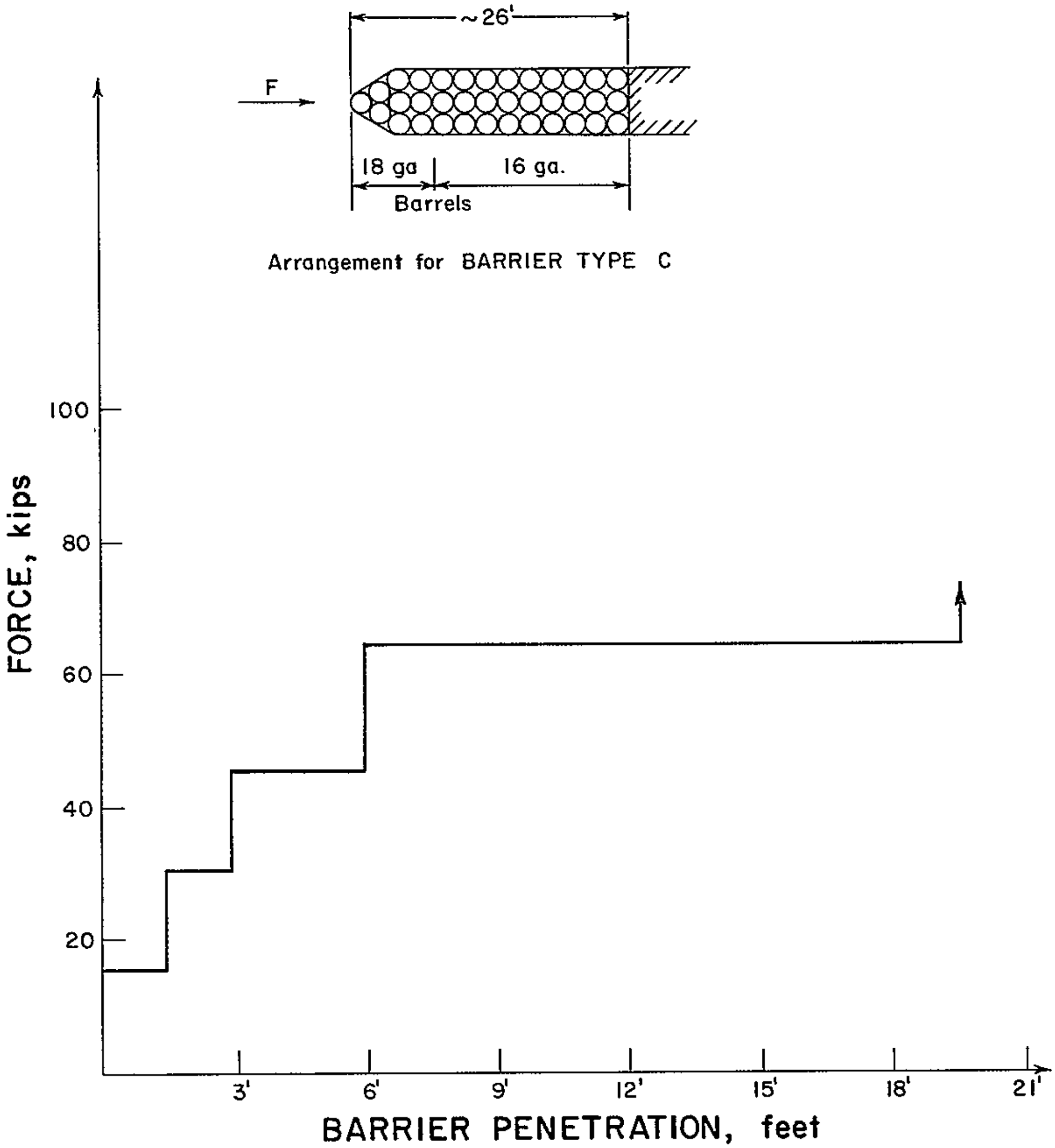

~26'
F
18 ga
16 ga.
Barrels
Arrangement for BARRIER TYPE C
FORCE, kips
100
80
60
40
20
3'
6'
9'
12'
15'
18'
21'
BARRIER PENETRATION, feet

Figure 8.D.8

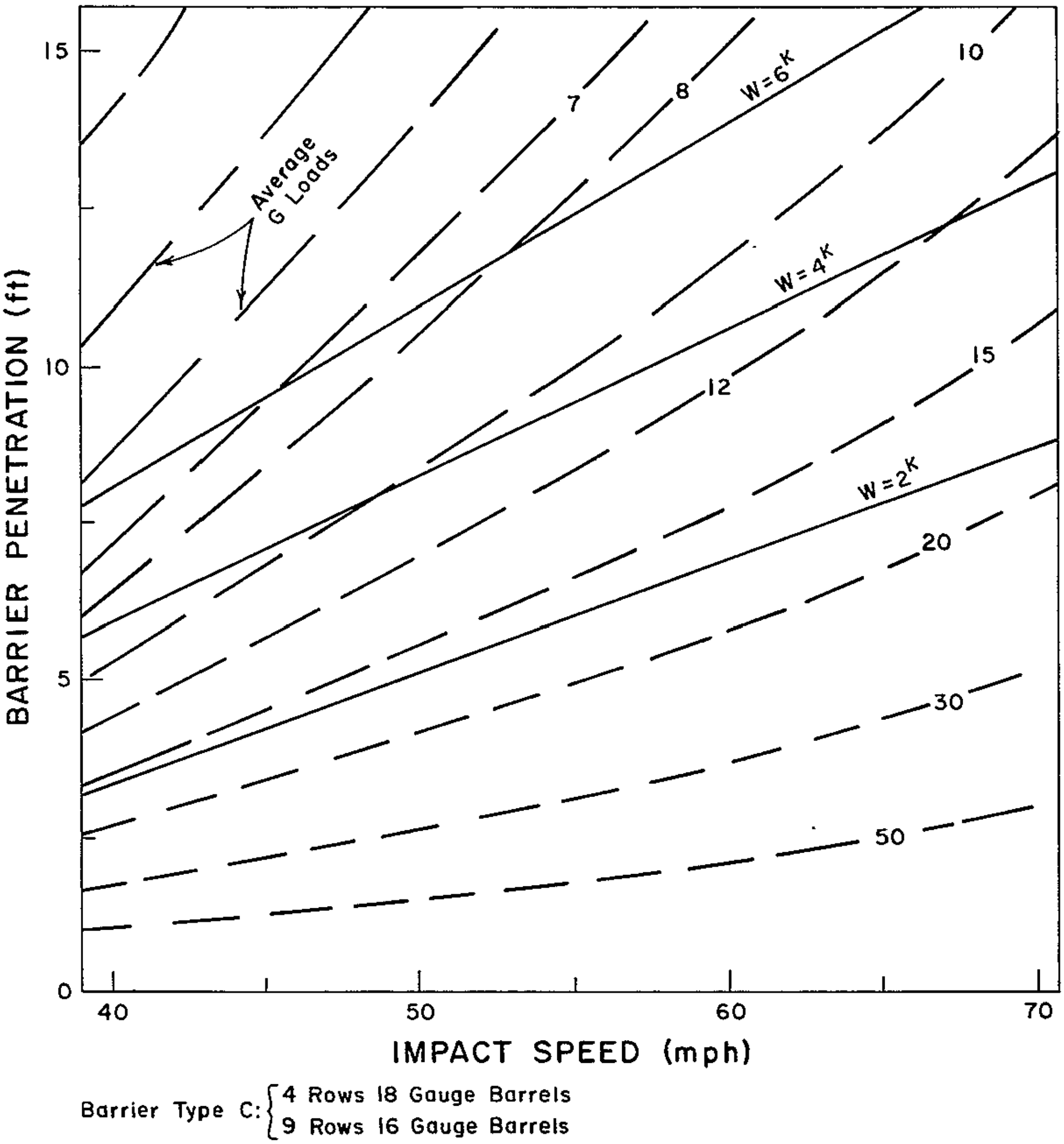

BARRIER PENETRATION (ft)
0
5
10
15
IMPACT SPEED (mph)
40
50
60
70
Average G Loads
7
8
10
12
15
20
30
50
W = 6K
W = 4K
W = 2K
Barrier Type C: 4 Rows 18 Gauge Barrels
9 Rows 16 Gauge Barrels

Figure 8.D.9

2. Design Considerations

From the previous research and experimentation in the area of energy absorbers many constraints and design limitations have been formulated. The Bureau of Public Roads has laid down human deceleration limits:

1. Average seat belted passenger deceleration of less than 12 g's.
2. Crash duration of 300-400 milliseconds.
3. Maximum force onset of less than 500 g's/sec.

Another important limitation is that of space at the site, i.e., the area for the device is only 6 feet wide and much less than 100 feet long.

A constraint that many other proposed barriers violate is that of few fragmenting particles. This is obviously a major constraint, because the particles (sand, water or concrete chunks) can be very dangerous to other vehicles in the system.

Maintenance and replacement costs must be minimum in the final device. In the proposed device there is no maintenance, except painting, necessary; and the replacement costs are not great.

3. Design Elements

To design an energy absorbing device which is within all the constraints and limitations is not easy, if ever possible.

The design developed last winter quarter fitted many constraints; some improvements and corrections were made last April in order to have a better lateral stability and minimize the "pocketing" that results in case of lateral hit.

The suggested arrangements A & B shown in the Figures 8.D.4 - 8.D.6 serve the following purposes:

a. Absorb the kinetic energy from a direct hit, and

b. Redirect a vehicle back into the system after an angle impact. (In both cases the vehicle should remain somewhat undamaged. In the second case, the car should not have lost too much speed).

Objective a

In both arrangements A & B, the nose is composed of empty 55-gallon oil drums (with 7" holes in top and bottom). The drums stand on a slightly sloping timber platform. They are held in place by two wire ropes stretched around the group perimeter and fastened to the rigid concrete barrier. The barrels can absorb the following energy amounts each - for a stroke of 1.5':

20 gage barrel: 13.5 K ft
18 gage barrel: 22.5 K ft
16 gage barrel: 31.5 K ft

The nose of the device absorbs energy in a direct hit and acts as an attenuator.

The back half of the device is made up of aluminum tubes about 20 to 24" in length connected to vertical steel diaphragms spaced 24" center to center. The bulkheads are about 3' X 5' in size. A typical connection between tubes and bulkheads is shown in Figure 8.D.10. These energy-absorbing tubes may be either folding tubes or tube and mandrel. Test results and analysis will be used to determine the suitable number of tubes and thicknesses between each pair of diaphragms.

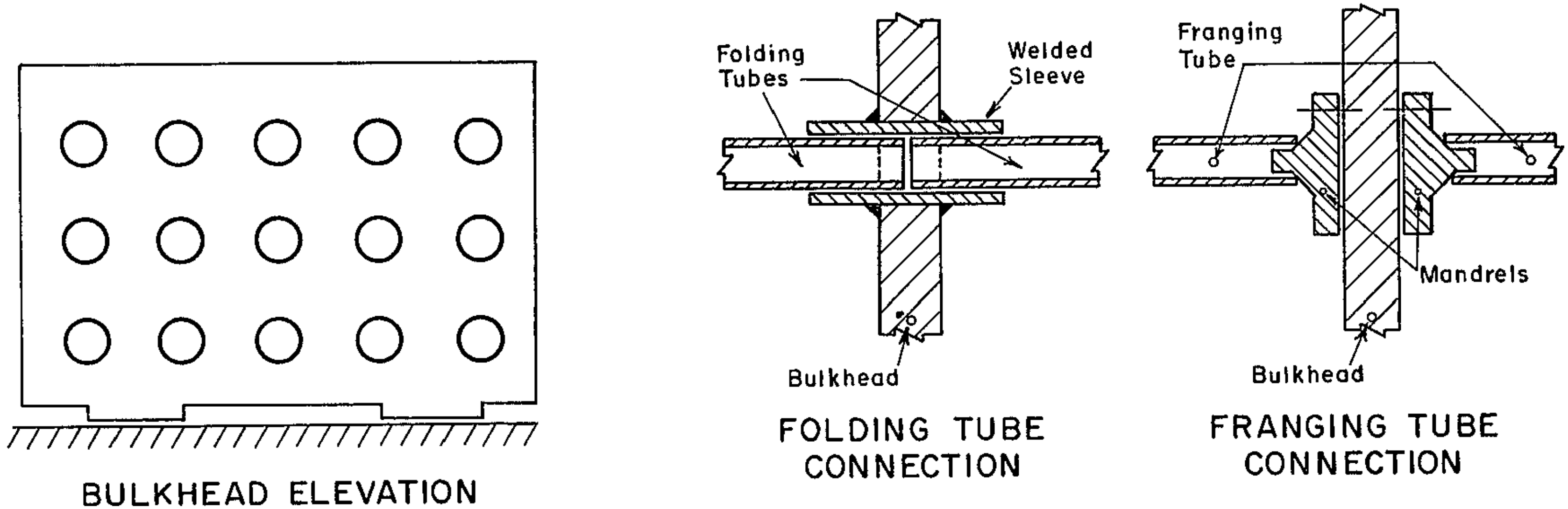

BULKHEAD ELEVATION
Folding Tubes
Welded Sleeve
Bulkhead
FOLDING TUBE CONNECTION
Franging Tube
Mandrels
Bulkhead
FRANGING TUBE CONNECTION

Figure 8.D.10

The nearer to the rigid barrier, the thicker the tubes should be in order to offer increasing force levels. The purpose of the tubes is anyway to absorb energy from direct hits only.

Objective b

To redirect a vehicle and minimize lateral displacements of the system, while eliminating the "pocketing" effect of common barrel barriers, a new device has been added. It consists of three parts:

i) A horizontal truss beneath the back half:

This truss lies at ground level. Chord members consist of rectangular tubing and diagonals are made from steel rope. A plan view is shown in Figure 8. D. 11. On top of the truss, timber boards are laid horizontally so as to cover its whole area. The previously mentioned bulkheads rest on top of the timber platform.

ii) A special U-shaped member:

See Figure 8.D.12 whose horizontal part lies on the ground and vertical parts serve as posts for the W guard-rail, has the function of transmitting horizontal shear forces to the truss and to resist bending moments. The cross section of this member should be flexible enough to eliminate too large decelerations from angle hits and stiff enough, however, as to limit deflections to about 3" to 5". The purpose of the latter limit is to prevent crushing of the tubes and bulkheads part in case of average angle hits and redirect the vehicle in the proper direction.

The W guard-rail is split into three overlapping parts bolted together in "fish scale" fashion for safety reasons. In fact, this arrangement will prevent the guardrail from being axially loaded when hit on the end by a vehicle.

iii) A special hold-down device for the cable:

Figure 8.D. 13 shows the detail of the front end of the tie-down cables. The special trough and roller guides allow the cables ends to "fly" when hit laterally. This arrangement of the ends adds to the overall safety and greatly improves on the original Texas barrier cables.

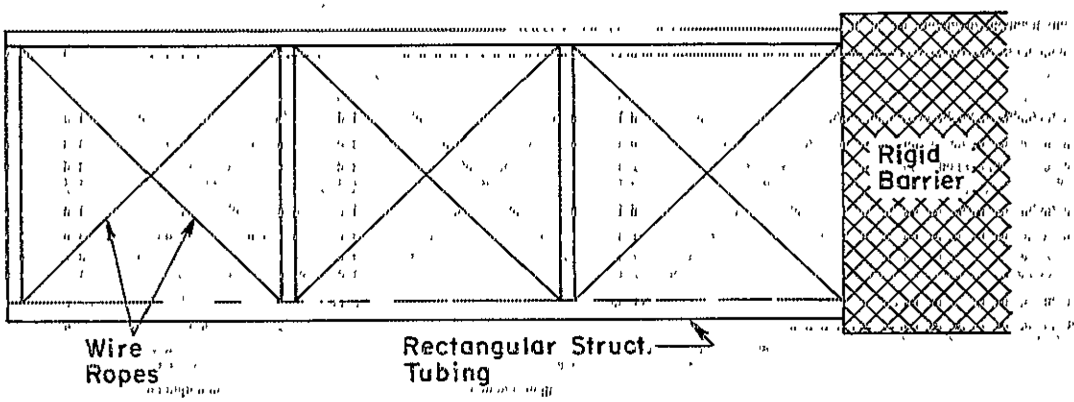


Figure 8.D.11. Plan View of Truss

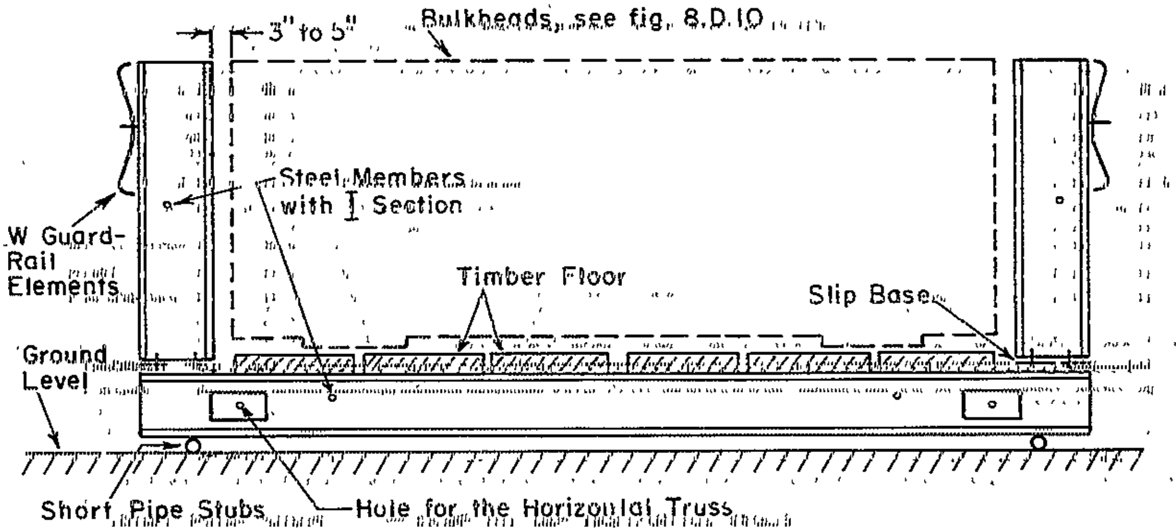


Figure 8.D.12. U-Shaped Member

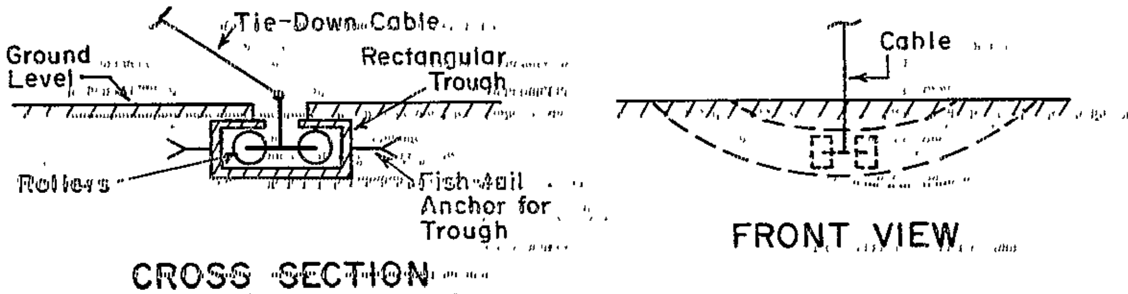


Figure 8.D.13

4. Conclusions and Recommendations

The system represents the group efforts for the winter and most of the spring quarters. All constraints, limitations and past experimental results and failures were considered and this design was developed. It is our opinion that the device will be capable of handling most of the probable accident situations.

A preliminary cost estimate for either one of arrangements A or B amounts to about $1,500. This seems relatively high when compared to arrangement C shown in Figure 8.D.8, the simpler Texas modular cushion barrier. The latter barrel barrier is the most efficient means of absorbing kinetic energy in terms of economy and simplicity of design. It is nevertheless our opinion that A & B could be useful for conjested gore areas due to the saving in overall length and the avoidance of most of the "pocketing" effect.

At any rate, it is felt that the addition of the horizontal truss and other device previously mentioned under "objective b" (paragraph 3) will greatly improve the Texas barrier.

The "pocketing" effect will be eliminated as well as the possibility of vehicle "tripping" at the nose of the hold-down cable.

E. CORRUGATED METAL STRIP BUFFER

The sheet metal strip buffer is an experimental buffer which was conceived during the course of the current program. It consists of a series of corrugated metal strips which are rigidly attached to the concrete gore (Figure 8.E.1). The desired force-displacement curve for a particular gore area is obtained by variation of the number, lengths, and thickness of the sheet metal strips and by the depth and frequency of corrugation. The purpose of the corrugations is to reduce material costs by increasing the bending stiffness of each strip.

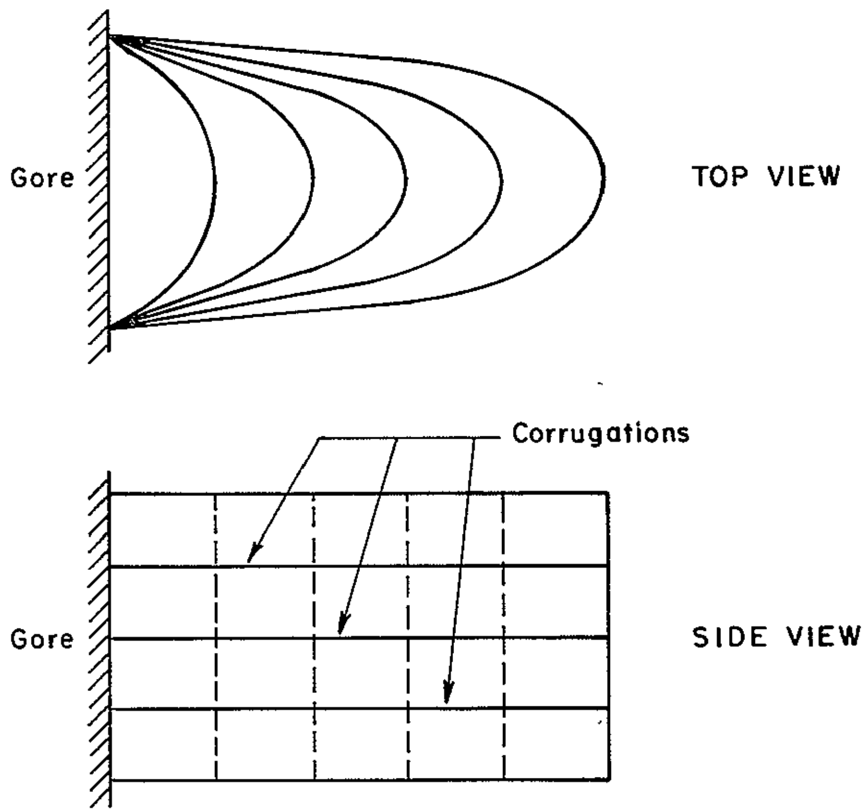


Figure 8.E.1

The possible advantages of this type of buffer are:

1. There is virtually no deceleration spike at the moment of impact since the buffer mass activated at this time is negligible compared to the mass of the impacting vehicle.

2. There is a tendency for the buffer to wrap around an impacting vehicle and therefore to prevent the vehicle from leaving the gore area and entering the highway in a disable condition.

3. The buffer requires no extensive hold down and interconnecting elements.

4. The buffer is considerably stiffer at its base than it is in the nose area. Thus, glancing impacts near the base will tend to redirect the vehicle past the gore.

Work on the development of the buffer has been limited thus far to a scale model experimental program whose purpose is to investigate the role of individual parameters in the force-displacement response of the buffer. These parameters are the yield strength σ_y of the strip material, the thickness t of the strip, the depth c of corrugation (for a sinusoidally varying corrugation), and the point and direction of load application.

The scale model strips of 1100-0 aluminum were made using the rolling device shown in Figure 8.E.2. The depth of corrugation, controlled by the position of the threaded dowels, may be varied between zero and .050", although the depth was not consistent for corrugations below .020". The curvature of the barrier is determined by the curvature and spacing of the rollers and the thickness of the material, but can be easily altered by hand pulling the sheet as it passes through the rollers. The roller produces circular strips, but will be modified so that parabolic strips can be obtained.

Test results have been obtained thus far for circular corrugated arches compressed by a static load applied axially at the midpoint of the arch. The test results are summarized in Table 8.E.1. The parameters π_1, π_2 and π_3 are dimensionless parameters which are defined on the following page.

CORRUGATION DEVICE

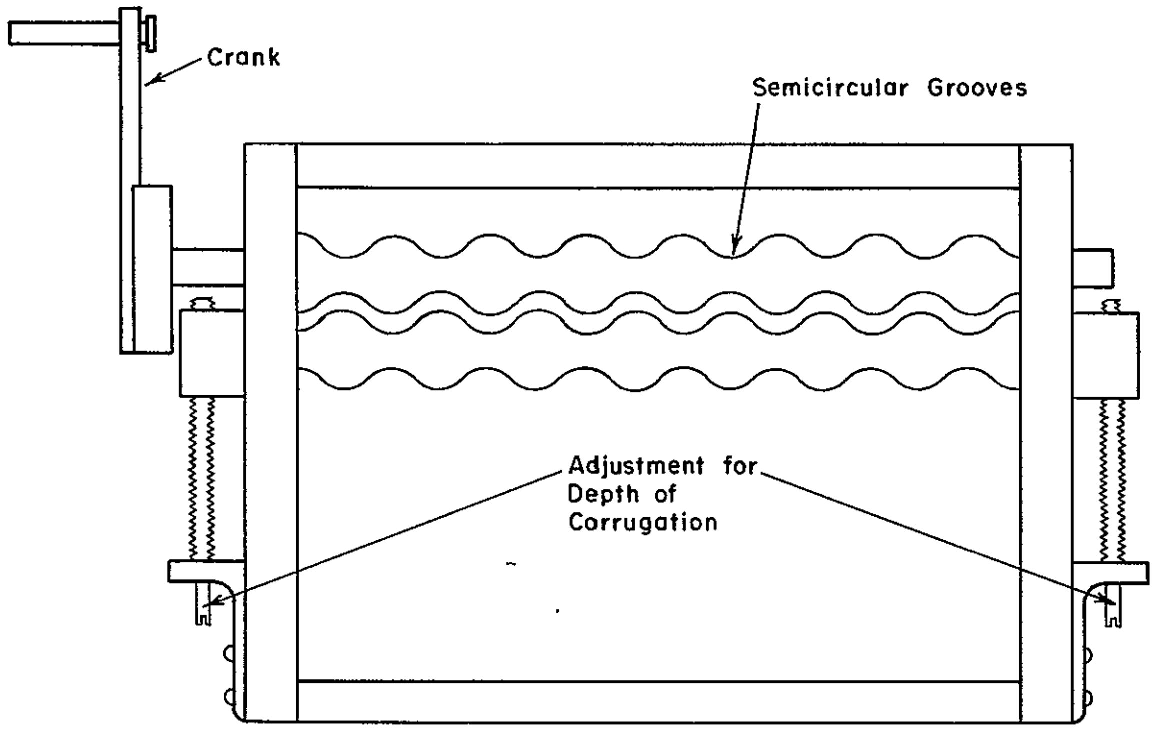


Figure 8.E.2

An empirical relationship between the important parameters was obtained by the use of the experimental results together with dimensional analysis. The dimensionless variables which were chosen were:

$$\pi_1 = \frac{P}{\sigma_y \sqrt{c^3 t}}$$

$$\pi_2 = \frac{t}{R}$$

$$\pi_3 = \frac{t}{c}$$

where P is the average force, σ_y is the yield strength of the buffer material, t is the sheet thickness, c is the corrugation height and R is the radius of curvature of the arch. Examination of π_1 as a function of π_2 and π_3, revealed that π_1 is nearly constant with changes in π_3, and

TABLE 8.E.1

Corrugated Barrier Test Results

Test No.	Radius of Curvature	Corrugation Depth	Thickness	Average Force	π_1	π_2	π_3
1	3 in.	.040 in.	.003 in.	2.6 lbs	1.3	.001	.075
2	3 in.	.040 in.	.004 in.	4.0 lbs	1.97	.0013	.1
3	4 in.	.020 in.	.003 in.	.4 lbs	.53	.00075	.15
4	4 in.	.040 in.	.003 in.	1.5 lbs	.685	.0075	.075
5	4 in.	.040 in.	.004 in.	2.3 lbs	1.03	.001	.1
6	4 in.	.050 in.	.004 in.	4.0 lbs	1.18	.001	.08
7	5 in.	.020 in.	.003 in.	.3 lbs	.4	.0006	.15
8	5 in.	.030 in.	.003 in.	.8 lbs	.53	.0006	.1
9	5 in.	.040 in.	.002 in.	1.2 lbs	.67	.0004	.05
10	5 in.	.040 in.	.003 in.	1.4 lbs	.64	.0006	.075
11	5 in.	.040 in.	.004 in.	2.1 lbs	.85	.0008	.1
12	5 in.	.040 in.	.005 in.	3.0 lbs	1.06	.001	.125
13	5 in.	.050 in.	.003 in.	2.4 lbs	.784	.0006	.06
14	6 in.	.020 in.	.003 in.	.21 lbs	.27	.005	.15
15	6 in.	.040 in.	.003 in.	.6 lbs	.3	.005	.075
16	6 in.	.040 in.	.003 in.	.9 lbs	.4	.0006	.1
17	6 in.	.040 in.	.005 in.	2.7 lbs	.955	.00083	.125
18	6 in.	.040 in.	.002 in.	.9 lbs	.5	.00033	.05
19	6 in.	.040 in.	.004 in.	.8 lbs	.443	.00066	.1
20	6 in.	.050 in.	.003 in.	1.6 lbs	.57	.0005	.06
21	6 in.	.050 in.	.004 in.	2.3 lbs	.707	.00066	.08
22	7 in.	.040 in.	.003 in.	.7 lbs	.32	.00043	.075
23	7 in.	.040 in.	.004 in.	1.2 lbs	.466	.00057	.1

varies linearly as a function of π_2. The resulting relationship, in terms of the basic parameters is

$$P = \sigma_y \sqrt{c^3 t} \left[\frac{2000t}{R} - 0.3\right]$$

Thus, a 1/16" mild steel buffer whose length was 10' and whose corrugation depth was 1", would provide an average stopping force of 7500 lbs.

REFERENCES

1. Saugatuck, "Traffic Safety, A National Problem," Eno Foundation for Highway Traffic Control, Connecticut, 1967.

2. Viner, J. G., and Tamanini, F. J., "The State-of-the-Art of Impact Energy Absorption Barriers," paper presented at the Sixth Annual Louisiana Highway Engineers Conference, Louisiana State University, Baton Rouge, Louisiana, February 1969.

3. Tamanini, F. J., and Viner, J. G., "Structural Systems in Support of Highway Safety," paper given at the ASCE Meeting on Transportation Engineering, Washington, D. C., July 1969.

4. J. D. Lacy, Director Office of Traffic Operations, Circular Memorandum to: Regional Federal Highway Administration and Division Engineers, December 12, 1968.

5. Record Number 222, Highway Research Board, Washington, D. C., 1968.

6. Record Number 259, Highway Research Board, Washington, D. C., 1969.

7. Emori, Richard I., and Link, Dan, "A Model Study of Automobile Collisions," Society of Automotive Engineers, International Automotive Engineering Congress, Detroit, Michigan, January 13-17, 1969.

8. Jurkat, M. Peter and Starrett, James A., Automobile-Barrier Impact Studies Using Scale Model Vehicles HRB Record 174, pp. 30-40, 1967

9. Murphy, Glenn, "Similitude in Engineering," The Ronald Press Company, New York, 1950.

10. Hirsch, T. J., "Barrel protective barrier," Technical Memorandum 505-1, Texas Transportation Institute, Texas A & M Research Foundation.

REFERENCES (Cont.)

11. Hirsch, T. J., Ivey, Don L. and White, Monroe C., "The Modular crash cushion, Research Findings and Field Experience," Texas Transportation Institute, Texas A & M University.

12. Murphy G., "Similitude in Engineering," The Ronald Press Company, New York, 1950.

IX. MOTOR VEHICLE SAFETY APPLICATIONS

1. GM Energy Absorber Application

a. Introduction

A visit was made to the Saginaw Gear Division, General Motors Corporation, Saginaw, Michigan to make a presentation of the work being done on energy absorbers, and to determine if GM had any applications for these devices. It was learned that GM does have a requirement for an energy absorber having the idealized characteristics shown in Figure 9.1.1. (Although GM did not specify the application of this device it is assumed that it is to be used in automobile steering columns to promote passenger safety.)

b. Tube and Mandrel

The device which we propose, as a solution to this problem, is the modified tube and mandrel made from 1018 mild steel, shown in Figure 9.1.2(b). This device retains the basic characteristics of the tube and mandrel with the exception that the tube wall is tapered (1.1° taper angle), with a .060 inch long straight section at the top. It also has eight notches, equally spaced (.050 inch long), cut into the base, causing the tube to split into eight equal segments. This design has been tested statically and demonstrates the force-deflection curve shown in Figure 9.1.3. The main features of this curve are similar to those of Figures 9.1.1(c), i.e., the sudden jump to 700 lb, the linear rise to 1800 lbs at 1.0 inch deflection and the leveling off for a short distance after reaching 1.0 inch deflection. The prominent feature which is not duplicated by the data of Figure 9.1.3 is the increase in load after the deflection has reached 1.06 inch. It is anticipated that this feature can be duplicated with a modification to the upper rim of the proposed tube. This modification will consist basically, of leaving more material around the top to keep the ring from spreading when the deflection is maximum. A second requirement which has not been fully met is the solid height after collapse. To date, the least solid height which has been obtained is nominally .215 inch (Refer to Figure 9.1.2(b)), (Figures 9.1.4 and 9.1.5 are photos of the actual device before and after static testing). The volume requirement of Figure 3.1(b) is satisfied by the proposed device since it measures 1.6 inch minimum ID and 2.6 inch maximum OD.

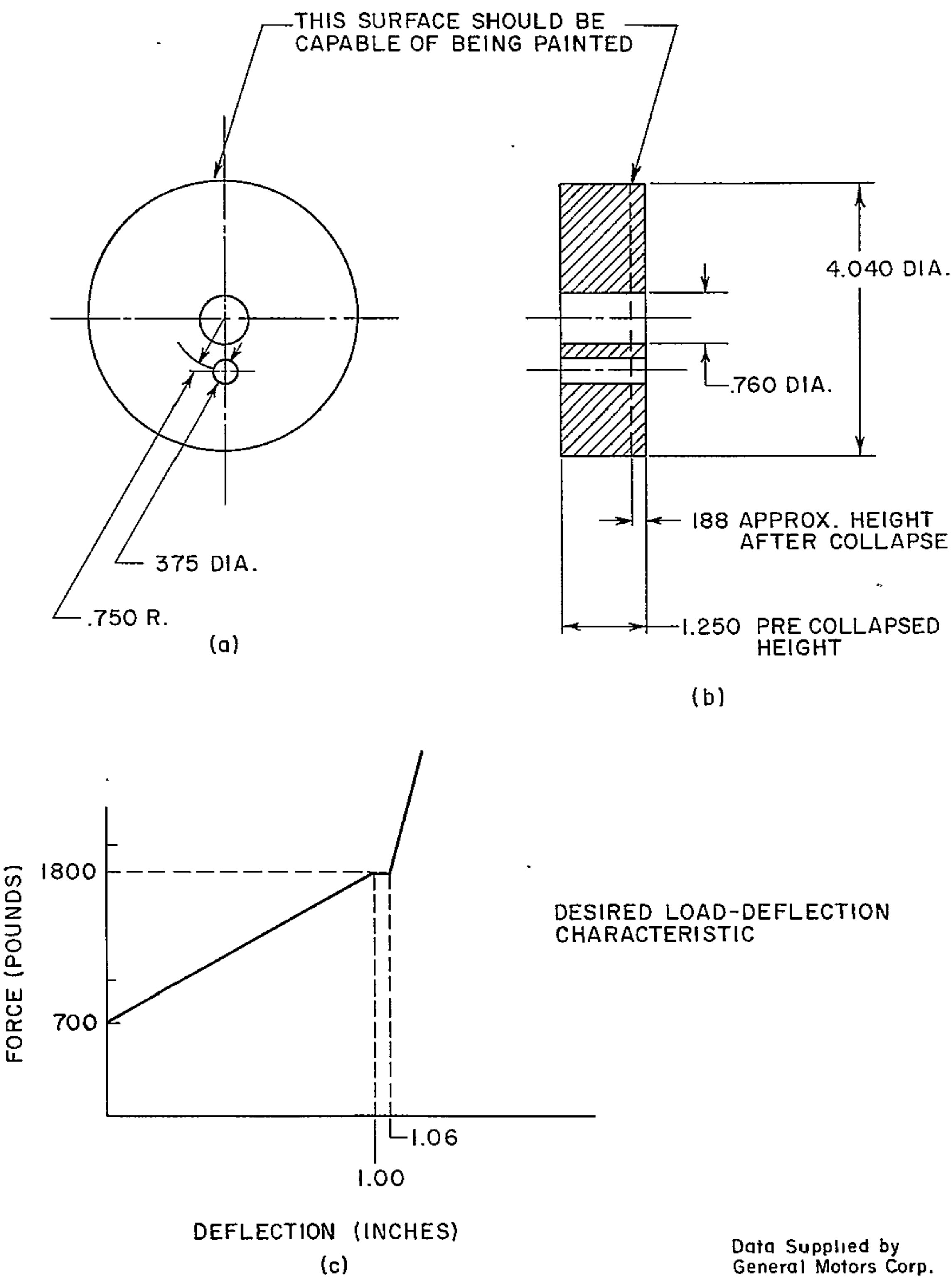
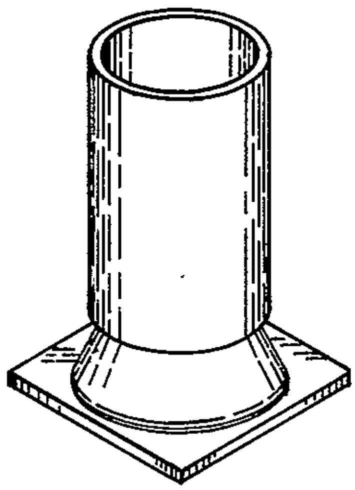
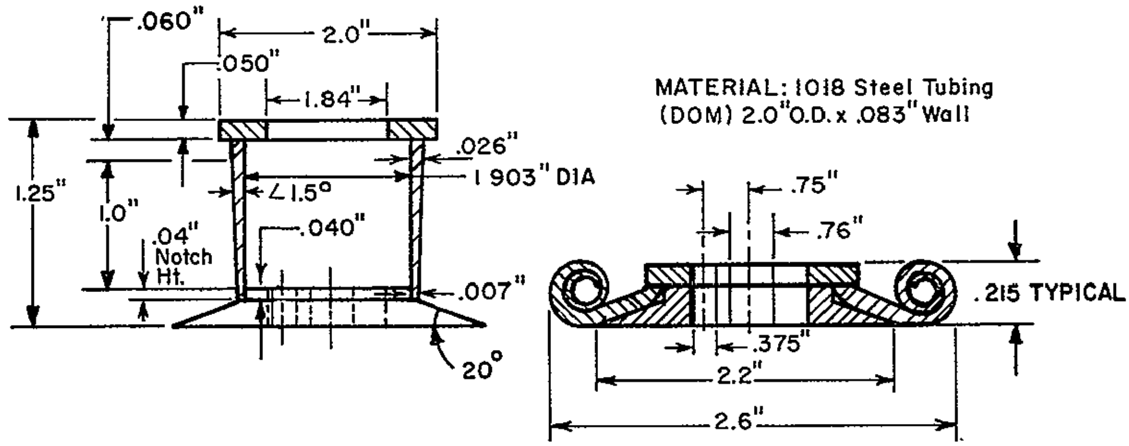


Figure 9. I. 1 General Motors Corp. Energy Absorber Requirements



(a)

Typical Tube and Mandrel Energy Absorber Using a Conical Mandrel



(b)

Tube and Mandrel Used for Small Solid Height and Linear Increasing Load Deflection Characteristics

Figure 9.1.2 Tube and Mandrel Energy Absorbers

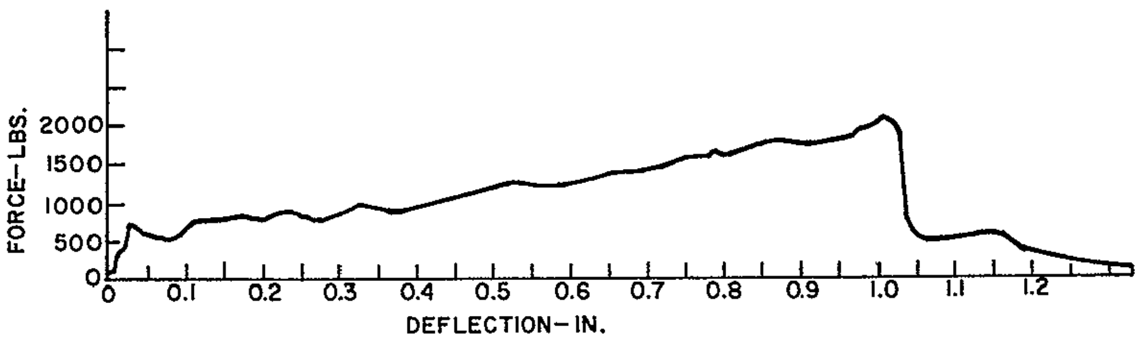
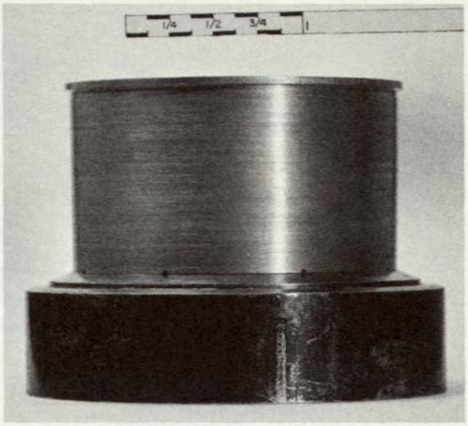
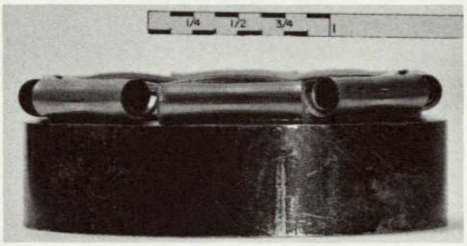


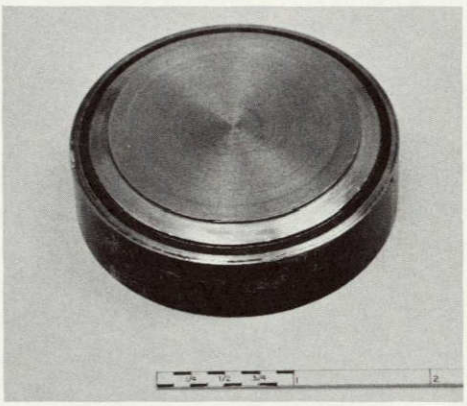
Figure 9.1.3 Force-Deflection Curve for the Tube and Mandrel Energy Absorber Designed to Meet the Requirements Given in Figure 9.1.1



(a)

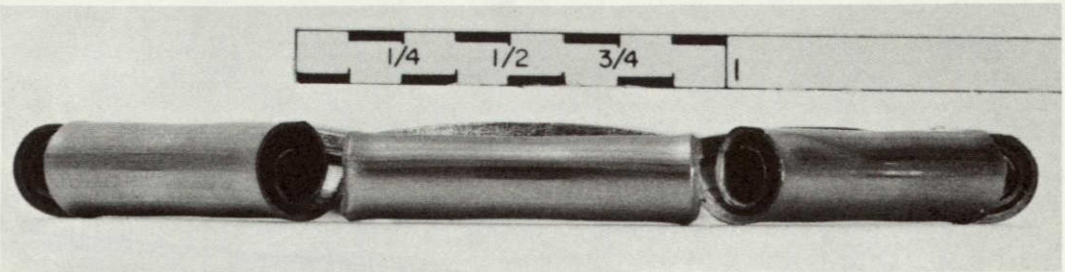


(b)

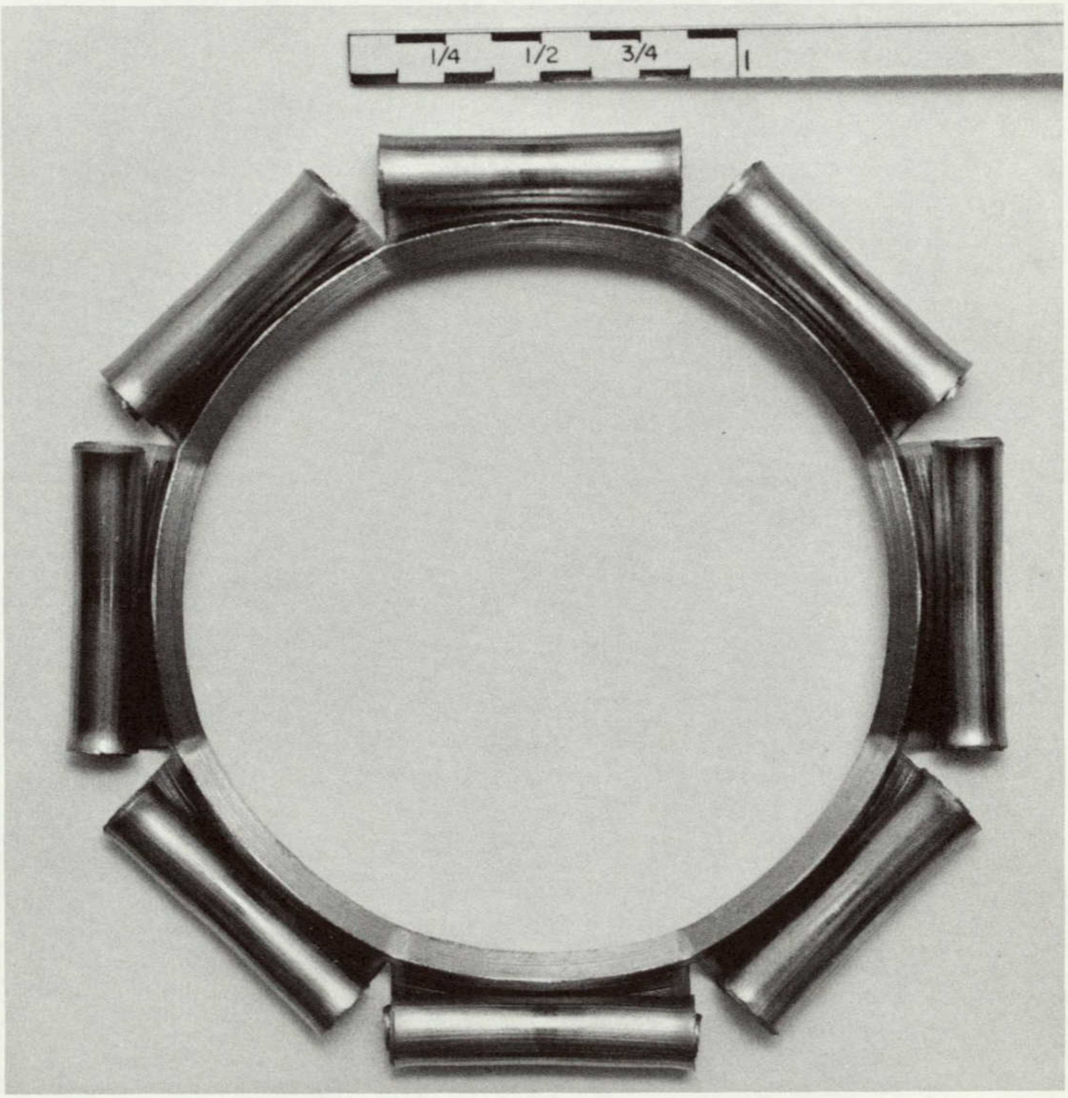


(c)

Figure 9.1.4 Proposed Tube and Mandrel Energy Absorber To Satisfy the GM Requirements. (a) Before Static Testing, (b) After Test (c) Mandrel. In Practice the Mandrel is Cut Off Just Below the Small Step. (Refer to Figure 9.1.2)



(a)



(b)

Figure 9. 1. 5 Two Views of the Collapsed Tube (Figure 9. 1. 4a) Used in the GM Energy Absorber

c. Manufacturing Requirements

The requirements put out by GM do not contain any manufacturing data such number required or production methods to be used. It is believed however, that the design discussed above does not present any unusual manufacturing problems and can be mass produced using standard techniques.

2. Motor Vehicle Crash Worthiness

a. Discussion

Few people seem to be aware that the passenger compartment of a private automobile is a very hazardous environment. Instead, passengers insist on the right to lounge in unrestricted comfort as in the living rooms of their homes. It is indeed unfortunate that these people refuse to take the hazard seriously. Over one and a half million people have been killed by vehicles.[1]

It is the so called "second collision" which kills or injures many passengers. A car in a 30-mile-per-hour, front-end crash into a solid barrier, experiences a front-end crush which allows the passenger compartment of the average American auto to move forward some two feet before it finally stops. This takes about 1/10 second. In the meantime, the unrestrained passenger moves forward in the passenger compartment at the unslackened pace of 30-miles-per-hour. After the instrument panel and windshield stop, the passenger crashes into them at 30-miles-per-hour. He then stops in, say three inches as he dents the instrument panel and puts his head through the windshield. This is the "second collision." The fully restrained passenger does not experience the "second collision." Instead he is decelerated with the car over a distance of two feet rather than three inches.

In spite of the fact that many organizations are working on the problem of protecting the auto passenger from the second collision, no true breakthrough solution has resulted. Such a solution would have to depend upon passive restraints which would function for the person who refuses to bother with any conscious action to protect himself. The mandatory, factory-installed headrest is an excellent example of such a passive restraint. It is a simple and effective device, but it functions only to prevent whiplash from a rear-end collision. Another passive device which is reasonably effective is the energy absorbing steering column. It converts the steering wheel from a driver-only hazard to a driver-only protection. It works best if the driver is belted, but also offers considerable protection to the unbelted driver.[2]

Some have hailed the instantly (1/40 second) inflated, impact-actuated air bags now under serious study as the final link in the protection of the passenger. If the problem of reliable actuation for this device can be solved, it will probably be a big step forward, particularly for the moderate straight-ahead impacts. It appears, however, that a lap belt would also be necessary to prevent an up-and-over pivoting of the passenger about his feet in a violent collision or sliding under the air bag in a more moderate one. And, in an angular or side impact an upper torso harness would also be required to prevent the passenger from being thrown against a side window or a roof post.

One approach to the passive protection of the passenger could be called encapsulation. If the automobile structure were extended until it contacted the passenger, in effect encapsulating him, he would be in contact with the structure at the beginning of the collision and would decelerate with it, thus avoiding the "second collision." It is not likely that passengers would tolerate such confinement although at present the driver does accept something similar in the form of the steering wheel.

Several arrangements of the above nature have been suggested. One suggested by J. D. Frost, involves air bags continuously inflated by the car's ventilating system.[3] These bags would hang limp until the car's engine was started. They would then promptly inflate, filling the space between the passenger and the car structure in front of him. An advantage over instantly inflated, impact-actuated bags would be the visual evidence that the bags were functioning. The bags would have to be kept low enough to avoid obstructing vision. Therefore, seat belts would certainly have to be worn to avoid up-and-over pivoting about the passenger's feet during collision.

Another arrangement is the "Cushion Car" suggested by David Foster.[4] Mr. Foster would pad the entire interior of the car with a thickness of about three inches of energy absorbing padding (material which slowly returns to its original form after deformation). Passengers would be restrained by a lap-bench configuration designed to prevent the up-and-over pivoting in a front-end collision. To prevent the passenger from being thrown to the side by angular collisions the seat cushion and a portion of the seat back would be mounted on tracks incorporating a rachet mechanism. As the passenger moved forward under the influence of a front end, or partially frontal, collision, this section of the seat would follow him forward. The rachet mechanism would lock the seat forward, pinning the passenger against the lap bench and preventing him from being thrown sideways. The forward motion of the pneumatic seat would open vents in it so that in a few seconds the seat would collapse allowing the passenger to escape.

Foster points out that lap belts and shoulder harness would not be needed in the cushion car thus making it entirely a passive device. Foster[4] speaks of the passenger being brought to rest after moving nine inches forward. Since the lap-bench padding is three inches thick, it would seem that there would be only six inches clearance between the lap-bench and the passengers' chest. This would make entry to the cushion car difficult, although not impossible. That such confinement and impediment to motion would be unacceptable to the motoring public is about as certain as any proposition which one could advance.

A suggestion made by Gregory Batten, a student on the project, merits further study. He suggested a pull out lap-bench. This would require active participation of the passenger. But the action of pulling out the lap-bench would probably be less troublesome than finding the two ends of a lap belt and buckling them. Certainly, it would be easier than the buckling of both a lap belt and a shoulder harness.

The pull out lap-bench concept could be combined with the cushion-car concept. The pull out lap-bench could be shaped so as to provide restraint for sideways motion as well as forward motion of the passengers. Foster's sliding seat feature would not be needed.

The design of the mechanical features of the pull out lap-bench may be a problem. An even larger problem would be an evaluation of the flailing of the face and head downward onto the padding of the lap-bench. A first reaction would be to doubt that three inches of padding could reduce decelerations to an acceptable value. However, the remarkable survival of eggs and glass balls when dropped onto energy absorbing padding as reported by Foster,[4] Frost[5] and Campbell,[6] has not been satisfactorily explained. Campbell dropped eggs in the Colorado State Capitol dome 135 feet onto a pad one inch nominal thickness of Ensolite resting on a concrete floor. The eggs did not break. The human head, resembling an egg somewhat, might exhibit similar remarkable survival.

Without a satisfactory total passive restraint system in view, it becomes necessary to decide what application should be made of the various voluntary and involuntary protection means available.

In an automobile crash at present we have four classes of people. (The advent of the instantly inflated air bag will modify this list somewhat.) We have:

1. Passengers who make full use of available restraints, namely, both the lap belt and the chest strap.

2. Passengers who use only the lap belt.

3. Passengers who refuse to use any voluntary restraints. (It is well known that the majority of passengers are in this group.)

4. The pedestrian who may be struck by the car.

We conclude that passengers in group 1, those who will cooperate by using available restraints, should be provided with the best protection the technology can devise. This protection would include:

(a) A passenger compartment which remains intact during collision. The passenger compartments of most modern automobiles should be improved.

(b) The energy of the crash should be managed to give the most tolerable possible decelerations to the fully restrained passenger while absorbing the maximum amount of energy.

(c) Full restraints, including head rests, lap belts and upper torso restraints, should be required as factory installations.

In the interest of those in group 2 who wear only lap belts, measures such as the removal of sharp objects and corners, the installation of flexible glass and the development of dash board energy absorbers should be pushed. Success in all of these measures should give the lap-belted passenger a good chance of survival.

The majority of passengers, those in group 3 who refuse to use restraints, cannot be given very good protection, but the things which will benefit groups 1 and 2 will give some benefit to group 3, particularly in the less severe collisions. Developments to make restraint systems more acceptable should be pushed to promote passengers from group 1 to group 2 or group 3.

The plight of the pedestrian should not be ignored. On Manhattan Island 80% of all auto accident fatalities are pedestrians. The figure for the nation as a whole is 20%.[7]

It is further concluded that reduction of the enormous costs incurred in the repair of damage in the five- and ten-mile-per-hour crashes should be included as a goal in the energy management approach to the auto crash. The additional costs involved in managing energy to reduce this damage may very well pay for itself in the savings on repairs.

The basic goal of this project is to find applications for energy absorbers. This means that principal attention should be directed toward the management of energy.

What is optimum in the management of energy in the automobile crash constitutes a complex problem. Intuitively it would seem that a constant crushing force should be provided in the front end of an automobile. But the Ford Motor Company disagrees.[8] In the development of the Ford "S" frame it was concluded that an early spike in the force-displacement curve would result in a reduced deceleration peak for the passenger restrained with both lap belt and upper-torso harness. However, the results of a computer analysis by one of the students on the project shows that minimum peak loading on the passenger occurs if a square wave collapse characteristic is used. The "S" frame (1969 production model) was designed to introduce such an early spike and resulted in a measured reduction of 18% in the peak deceleration experienced by the passenger compared to the 1966 model.

3. The Combined Effect of the Auto Collapse and Restraint Characteristics on Passenger Loading

If the integrity of the passenger compartment is assumed, the problem of energy management can be separated into two parts. Various deceleration-time histories can be assumed for the passenger compartment and the effect of these upon the restrained passenger deceleration-time history determined. This leads to a trial and error determination of an optimum combination of compartment deceleration history and restraint system characteristics. The development of such a restraint system could then be pursued together with the development of a frontal structure which would give the desired passenger compartment deceleration history.

A division of the energy management problem as described in the above paragraph is the method being followed in this project. A restraint system which keeps the passenger in his normal upright position is assumed. The system consisting of the passenger and the restraint is forced by various passenger compartment deceleration histories. The characteristics of the restraint are also varied.

To date, twenty-four combinations have been subjected to computer study, using an analogue simulating digital computer program called "SLASH" which was developed at the U.S. Air Force Academy.

These combinations are listed in Table 9.3.1. All of the deceleration histories used, bring a car to rest from 30-mile-per-hour in the same length of time. In all cases the restraint system connects the occupant directly to the car structure. For each run, a graph of passenger deceleration versus time results. The maximum passenger accelerations in G's recorded in Table 9.3.1 do not tell the whole story because some of the peaks are very brief while others are sustained. A complete presentation of these data is not given here because the work is still in progress and it is expected that a better presentation can be made when it is completed. The computer program is now being developed to deal with a two-mass system. One mass is the passenger attached to the seat by a conventional harness which keeps him in an upright position. The second mass is the safety seat which will be attached to the car structure by means of an energy absorber or a dash pot.

TABLE 9.3.1

Forcing Function	Restraint System		Maximum Passenger G's
Square Wave	Undamped	K = 800 lb/in.	30
Square Wave	Undamped	K = 1500 lb/in.	30
Square Wave	Undamped	K = 2400 lb/in.	30
Square Wave	Crit. Damp	K = 1000 lb/in.	17
Square Wave	Crit. Damp	K = 1500 lb/in.	17
Square Wave	Crit. Damp	K = 2400 lb/in.	17
Early Spike	Undamped	K = 800 lb/in.	38
Early Spike	Undamped	K' = 35(y-x) + 35 lb/in.	41
Early Spike	Undamped	K' = 50(y-x) + 50 lb/in.	49
Early Spike	Undamped	K' = 85(y-x) + 85 lb/in.	45
Early Spike	Crit. Damp	K = 1000 lb/in.	41
Late Spike	Undamped	K = 800 lb/in.	25
Late Spike	Crit. Damp	K = 1000 lb/in.	46
Ford "S" Frame	Undamped	K = 300 lb/in.	27
Ford "S" Frame	Undamped	K' = 35(y-x) lb/in.	44
Ford "S" Frame	Undamped	K' = 50(y-x) lb/in.	44
Ford "S" Frame	Undamped	K' = 85(y-x) lb/in.	40
Ford "S" Frame	Undamped	K' = 35(y-x) + 35 lb/in.	43
Square Wave	Unrestrained	Initial Space = -10"	42
Early Spike	Unrestrained	Initial Space = -12"	42
Early Spike	Unrestrained	Initial Space = - 8"	42
Late Spike	Unrestrained	Initial Space = - 8"	58
Late Spike	Unrestrained	Initial Space = -12"	42
Ramp-Square Wave	Unrestrained	Initial Space = -10"	42

4. Safety Seat

a. Introduction

As discussed previously, the energy in an automobile crash should be managed to give tolerable decelerations to fully restrained occupants while absorbing the maximum amount of energy. To optimize the crash characteristics of the automobile it is necessary to maximize the stopping distance of the occupant. This requires that as much of the distance as possible between the occupant and the front bumper of the automobile be used as stopping distance in the event of a frontal crash. An untapped source of stopping distance exists within the passenger compartment. According to auto manufacturers the front seat passenger can move forward an average of 8.5" without impacting the interior structure of the passenger compartment.

In order to adequately utilize this 8.5" of forward clearance as stopping distance during a crash the occupant must be allowed to move forward in a controlled fashion. One possible method for doing this is to equip the seat belts with tension type energy absorbing devices so that the seat belt elongates in a controlled fashion, however, this would allow the occupant to move out of his seat where subsequent lateral movements of the auto could throw him against the door. A better method involves the use of an integral bucket seat and restraint system such as the ones designed by Liberty Mutual Insurance Company, Boston, Mass., and the Allied Chemical Company, Detroit, Mich., with the added feature of an energy absorbing system to allow the entire seat to move forward in a controlled fashion. The latter method of controlled forward movement is the one which will be discussed here.

b. Seat Design

Studies have been made of many configurations for incorporating an energy absorber in the attachment of the Allied Chemical Company safety seat to the floor. Configurations involving motion of the seat on tracks at or near the floor line, similar to current seat adjusting mechanisms, have proven unsatisfactory. The inertia force acting through the center of gravity of the passenger-seat combination produces a heavy moment which gives rise to heavy forces on the blocks or rollers moving along the track.

The only arrangement thought to be satisfactory is the parallelogram arrangement shown in Figure 9.4.1. Two methods of

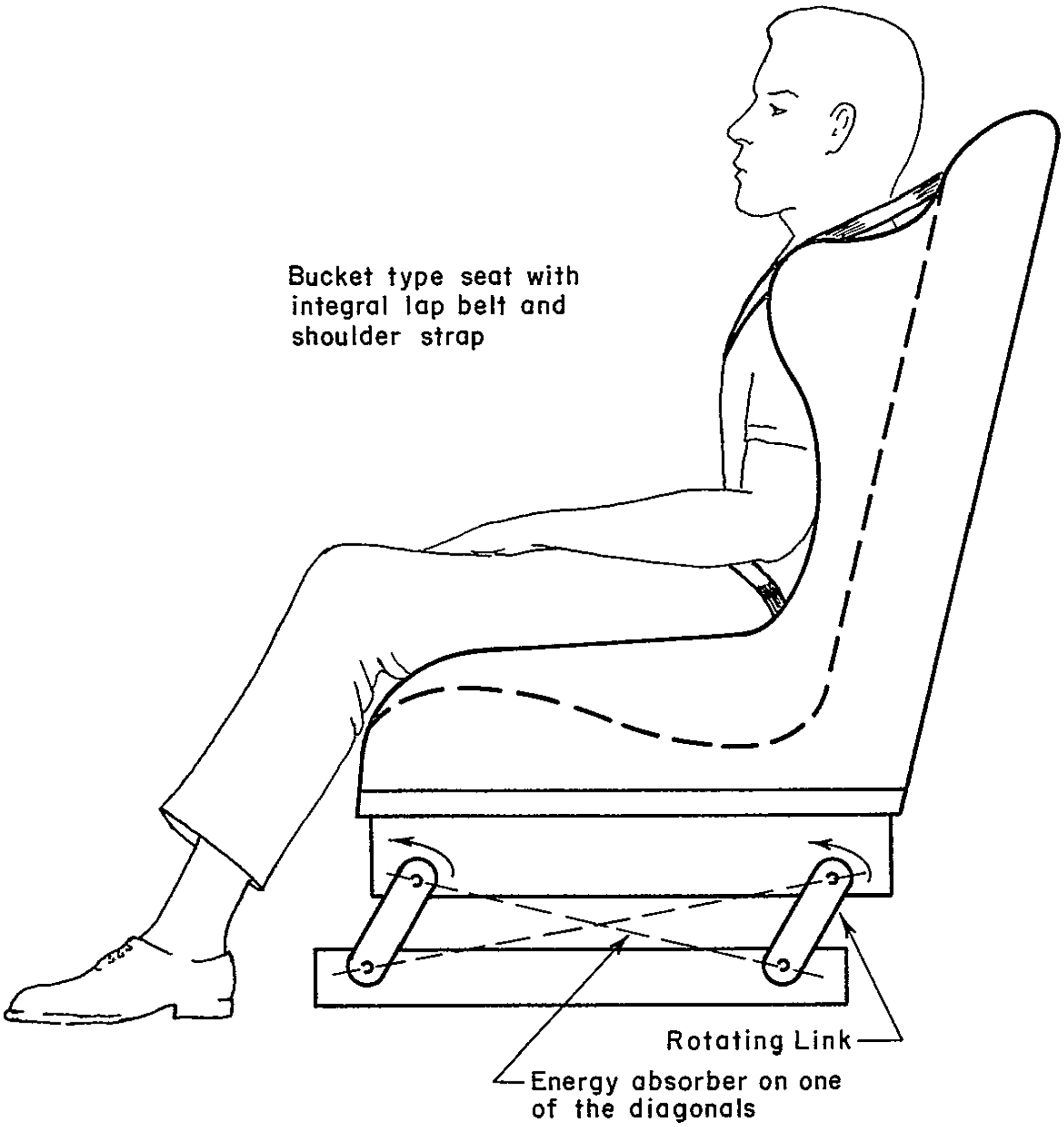


Figure 9.4.1 Sketch of Proposed Energy Absorbing Seat

incorporating energy absorption seem feasible. One would attach an energy absorber in an inclined position between the seat or the top of one of the support bars and the floor. The other method would incorporate a torsion type of absorbers at the pivot points. The energy absorber mounted on the diagonal could be a tube and mandrel. With suitable fittings this device can be made to operate either in tension or compression so it could be used on either diagonal. The torsional energy absorber used at the pivot points could be mild steel pins which are fixed at both ends so that relative rotations between the links of the parallelogram result in plastic rotations of the pins. Both of these energy absorbers are being considered.

c. Mathematical Model

A simplified model of the occupant and seat system are shown in Figure 9.4.2. This is a two dimensional two degree of freedom system involving the occupant, seat, seat belt, and energy absorber. At time t = 0 the auto impacts a fixed object and the frontal structure begins to collapse imparting the acceleration x to the system. The equations of motion for the occupant, M_1, and the seat, M_2 are

$$M_1(\ddot{s}_1 - \ddot{x}) + K(s_1 - s_2) = 0 \qquad (9.4.1)$$

$$M_2(\ddot{s}_2 - \ddot{x}) - K(s_1 - s_2) - L = 0 \qquad (9.4.2)$$

where K is the spring rate of the seat belt and L is the load transmitted through the energy absorber and where K is a function of $s_1 - s_2$ and L is a function of s_2.

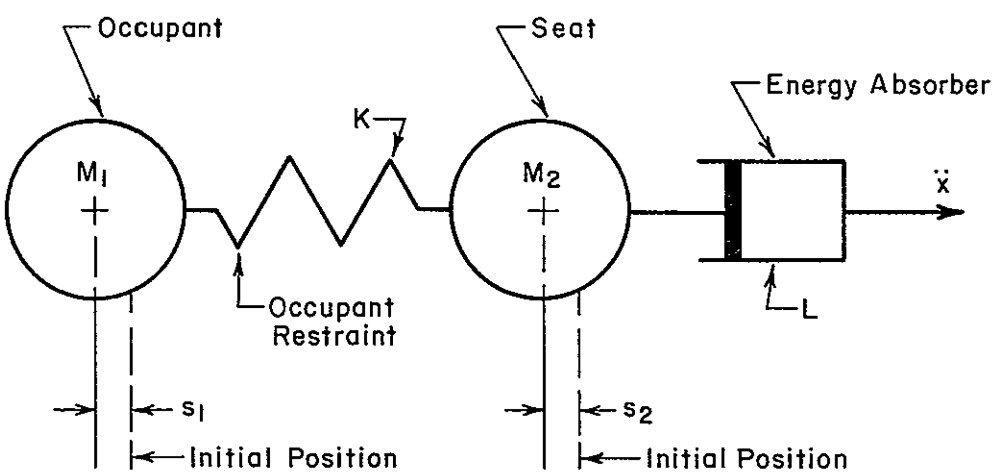


Figure 9.4.2

A computer program has been developed to solve the above equations for the purpose of doing a parameter study. The results of this study will indicate the effect on passenger loadings of using the energy absorbing device and will also help to determine the energy absorber characteristics required.

5. Controlled Collapse Frontal Structures for Autos

a. Introduction

The crashworthiness of automotive vehicles can be improved by designing the structure to absorb the crash energy in a controlled fashion. Recent work by many investigators has demonstrated that large amounts of mechanical energy can be absorbed by collapsing structures. Structural elements have been tested which not only absorb large amounts of energy per pound of material but also make very efficient use of the collapse distance. These structures collapse in a very predictable fashion so they lend themselves to the development of an auto structure which has predictable collapse characteristics. This Section addresses the problem of developing a frontal structure having optimum collapse characteristics.

The present American production auto collapses in the front end with a force of approximately 10 to 12 G's per foot in a barrier impact. To achieve maximum efficiency the force deflection curve must be made as close as possible to a square wave. The amplitude of this square wave must be as high as possible without imposing intolerable loads on the occupants in order to provide protection over the greatest range of speeds. It is believed that properly restrained passengers (lap belt and shoulder strap) can tolerate accelerations in the 40-60 G range for the time period involved in the collapse of the frontal structure of an auto. Therefore, the criteria for efficient frontal collapse should be square wave collapse with an amplitude in the 40 to 60 G range and the greatest possible collapse distance.

The maintenance of the integrity of the passenger compartment during the collision is extremely important to the occupant survival. The collapsing structures of autos should be designed, as much as possible, to prevent intrusion in impacts with pole type structures and other autos. This requires the use of a rigid beam at the exterior of the structure to distribute the loads and promote compatibility between impacting autos. The collapse force in two impacting vehicles must be nearly the same to prevent one vehicle from intruding into the structure of the other. Since autos vary considerably in mass the equal collapse force requirement will result in a difference in the G level of the accelerations experienced by autos having different masses. Thus, the compatibility requirement dictates that the collapse force must be the maximum possible for the compact auto and varied accordingly for autos which are more massive. This leads to the requirement of a 60 G collapse for the compact car, a 40 G collapse for the medium sized car and 30 G for the luxury car.

The response of the passenger compartment of an auto during a frontal collision should be such that acceptable levels of human tolerance are not exceeded. This condition places an upper bound on allowable force levels and therefore on the force-displacement response of the front end structure. The energy dissipated by the frontal structure is the area under its force-displacement curve. To maximize energy absorption, and thus provide protection at the highest possible velocities, the force level must be raised to a value near its upper bound and the maximum displacement (stopping distance) must be increased as much as possible.

Present frontal structure designs of American make autos do not begin to meet these ideals. We, in this section, will indicate how the present structures can be modified, with a minimum of geometry change, so that the frontal structure operates satisfactorily as an energy absorbing system. We will limit ourselves specifically to the redesign of the frontal structures of a typical late model luxury auto with the requirements of a 30 G square wave passenger compartment response.

b. Energy Absorbing Systems and Elements

1. Characteristics

An energy absorbing system is an arrangement of individual energy absorbing elements and tie-down, interconnecting, and load distributing elements. The energy absorbing elements are those structural members which deform plastically and/or fracture under impact. The frontal structures of current automobiles may be regarded as inefficient energy absorbing systems, since the amount of energy they dissipate per weight of deformed material is small. The deformed members of the structure have not been designed to function as energy absorbing elements and therefore there is wide variation in the modes of finite deformation and the force levels associated with them. It is not surprising that the force-displacement characteristics of current frontal structures are not satisfactory for the prevention of injury for other than low speed collisions.

The key to the proper design of an energy absorbing system, we believe, is in the selection of suitable energy absorbing elements whose modes of failure and operating force levels are predictable analytically or at least determinable from a simple experimental program. The development of an accurate analytical model of the system (prerequisite to the development of a computer program) is virtually impossible without the use of such elements.

2. Selection Criteria

Most of the available mechanical energy absorbing devices can be engineered to operate over a wide range of force and energy levels. In order to evaluate a particular energy absorber for use on the frontal structure we must first identify those properties which are important in this particular application. The following characteristics were used as a basis of comparison of the approximately fifty devices we have examined.

(i) Stroke Efficiency

By stroke efficiency we mean the percentage of the original length of the device which can be used for controlled energy absorption. The stroke efficiency of the energy absorbing elements will be the major factor in determining available stopping distance.

(ii) Adaptation for Use as a Structural Member

Prior to impact the energy absorbing element will function as a structural member of the frame. It must be able to transmit, without plastic deformation, the bending and twisting moments and the shear forces which occur under normal driving conditions.

(iii) Material Efficiency

Material efficiency is the energy absorbed per pound of device. For a particular device there can be large variations in material efficiency with changes in materials and geometry. In general, however, when only a small portion of the available material is permanently deformed or fractured, as in devices which absorb energy by the formation of localized plastic hinges, material efficiency is low. High material efficiency is desirable since it minimizes the weight of the energy absorbing elements. The total weight of these elements is thus kept small compared with the total weight of the front end. A vehicle with an efficient energy absorbing system would therefore have the same center of gravity and handle about the same as current vehicles.

(iv) Reliability

An energy absorbing element in the frontal structure of an automobile will be required to operate over a wide range of environmental conditions during a time period measured in years. Therefore, devices which rely on friction and/or close tolerances for proper operation, and devices which contain weather sensitive materials have not been considered.

(v) Directional Load Capability

Those devices whose primary mode of energy absorption was in tension rather than compression were not considered. The compression devices were also required to operate successfully under some degree of eccentricity of the applied load.

In addition to these criteria we also considered the need for a description of the response characteristics of the device in terms of its geometry and material properties.

c. Selection of Energy Absorbing Elements

1. Devices Satisfying Selection Criteria

Of the approximately fifty devices we considered only two are satisfactory in nearly all respects. These are the folding tube and the tube and mandrel.

The folding tube is simply a straight thin walled tube. Under the application of a sufficiently large axial load the tube forms circular or plygonal folds as it shortens and absorbs energy by the plastic work done in forming the folds. The tube will collapse to about 20% of its original length and therefore, has a stroke efficiency of 80%. There is an initial peak force P_{max} followed by a regular fluctuation in the force level about some average value $\overline{P}$. Each cycle corresponds to the formation of a single fold. The force level is not constant because of the change in geometry which occurs during the folding process.

The tube and mandrel device, consists of a thin walled tube and a conical mandrel. With the application of an axial load the tube is forced onto the mandrel and expanded radially. When the axial load reaches a critical value the tube begins to split longitudinally in several places around the circumference. The fractured sections flow out along the mandrel and then curl on themselves. The device absorbs energy by plastic radial expansion, fracture, plastic bending, and friction. The stroke efficiency depends primarily on the original length of tube and the height of the mandrel. Typical values range between 80 and 90 percent.

The force-displacement curve for the tube and mandrel, is flatter than that of the folding tube. The slight variations in the force level are produced by curl breakage and by changes in the surface properties of the mandrel.

We shall now examine both of these devices with regard to various aspects of their use in the front end structure of automobiles.

2. Materials Used for Folding Tube and Tube and Mandrel

(i) Folding Tube

The material must be ductile enough to allow fold formation without splitting. Generally, materials which have an elongation greater than 15% are satisfactory. Thus, mild steel and 6064-T6 aluminum may be used, but 2024-T3 aluminum may not be.

(ii) Tube and Mandrel

The material must have medium ductility. If the material is too brittle the initial cracks propagate longitudinally along the length of the tube, while a very ductile material will fold prior to splitting. Notching the tube tends to encourage fracture and thus allows the use of materials which would otherwise be too ductile. Examples of suitable materials are mild steel and 2024-T3 aluminum. Cast iron and 1100-0 aluminum are not suitable.

Both devices can be made to function properly using common metals. These metals are easily rendered insensitive to the oil, dirt, grease, etc., they will be subjected to.

d. Adaptation of Energy Absorbers as Structural Members

The designs given in this section are intended to serve only as one example of the many ways in which either of the two devices may be adapted for use as a structural member.

1. Folding Tube

The addition of a flange as shown in Figure 9.5.1 to both ends of a tube will enable the tube to transmit both torque, bending moment and end shear. The flange may be either welded or bolted to an adjoining structural element.

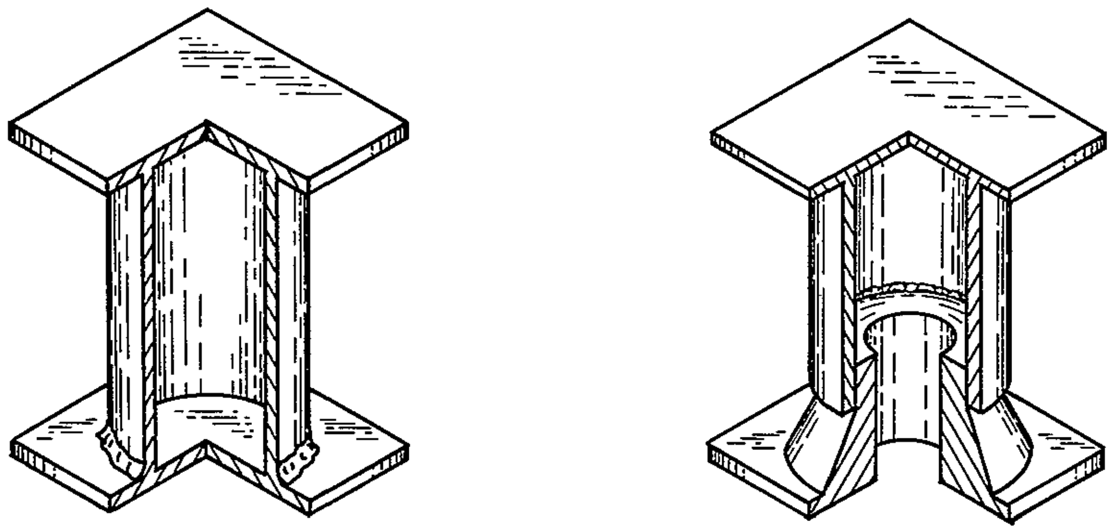


Figure 9.5.1. End Connections for Energy Absorbing Elements

2. Tube and Mandrel

The mandrel can be extended into the interior of the tube, Figure 9.5.1 and the tube can be attached to the extended mandrel by welding, brazing, or by the use of a shear pin. The connection must be sufficiently strong so that normal structural loads can be transmitted, but must fail under impact conditions to allow axial motion of the tube. The upper end of the tube need only have a flange section as discussed above.

e. Evaluation of the Application of the Folding Tube and the Tube and Mandrel in Auto Structures

From the preceding discussion it is clear that both the folding tube and tube and mandrel have high stroke and material efficiencies and can be easily adapted as structural members. There already exists a considerable body of knowledge concerning their behavior under purely axial load, and there appears to be no reason why the use of dimensional and mathematical analyses coupled with experimentation cannot be used to achieve the same level of understanding about the operation of tubes which are subjected to combined end loadings. The folding tube has no reliability problems. The tube and mandrel absorbs part of its energy through friction and thus the operating force level varies somewhat depending on the nature of the sliding surfaces. This problem can be eliminated by the application of a lubricant to the mandrel (or painting it).

Both devices appear to be quite satisfactory for use as energy absorbing elements in the auto frontal energy absorbing structures. Because of its simplicity and high material efficiency, we would probably use the folding tube as our basic energy absorbing element. However, we can foresee certain instances where the tube and mandrel would be preferable to the folding tube. The operating force levels of either device at a given location in the structure would be essentially the same. In view of the differences in material efficiency, the tube and mandrel would require a tube whose wall was perhaps two to three times as thick as that of the folding tube. The tube and mandrel would therefore be less likely to collapse as an Euler column or to develop a plastic hinge. It would also have a higher capacity for carrying the pre-impact operating design loads of the structure. Thus, the tube and mandrel might be used in high load carrying areas of the structure and near the bumper in the event of oblique impact.

We propose, therefore, to retain a high degree of design flexibility by having both the folding tube and tube and mandrel available as basic energy absorbing elements.

f. Prediction of Element Response Collapse Characteristics

The proper design of energy absorbing elements and the prediction of the response characteristics of the system requires a knowledge of the force-displacement relation as a function of material properties and element geometry.

1. Primary Mode of Collapse

Based on our previous work on mechanical energy absorbing devices, approximate relations of the force levels necessary for the primary mode of operation of both the folding tube and tube and mandrel devices have already been developed using the techniques we have just described. (See Sections III and IV.) The behavior of the folding tube and tube and mandrel under more complex loadings (a combination of axial load, bending moment, and end shear) is required for the purpose of the proposed study, however, this will necessitate further analysis and experimental work.

(i) Primary and Secondary Collapse Modes

Accurate prediction of the force-displacement relationship is necessary but not sufficient for design purposes or for the determination of system response. The conditions under which the device will operate in its primary mode must also be specified. Both the folding tube and the tube and mandrel also can fail, i.e., undergo large plastic deformations, by column buckling or by the development of a plastic hinge at the cross section of maximum moment. We will call these secondary modes of energy absorption. In general, operation in a secondary mode is to be avoided because of the decrease in the efficiency of energy absorption. Thus, under axial load P we would design the tube so that its length L and moment of inertia I are such that

$$P < \frac{\pi^2}{4} \frac{E_t I}{L^2} \qquad (9.5.1)$$

in order to avoid column buckling. E_t in equation 9.5.1 is the tangent modulus. Similarly the formation of a plastic hinge is avoided if the maximum bending moment M satisfies the inequality

$$M < \pi Y D t. \qquad (9.5.2)$$

For the general case where the tube is subjected to a combination of axial load, bending moment, and end shear, some modification of these equations must be made.

(ii) Operation in the Primary Mode of Energy Absorption

The prediction of the range of end loading conditions for which the folding tube and tube and mandrel will operate in their primary mode of energy absorption was discussed previously. The range must be sufficiently wide to allow the flexibility needed for proper design. The only indications we have at this time concerning the magnitude of this range are favorable. In a test of a 3" diameter aluminum tube with a .050" wall thickness where the load was applied concentrically but at an angle of 10° with the axis of the tube, the tube folded. The only information we have concerning non-axial loading of the tube and mandrel comes from Lockheed Corporation use of the device in the legs of a proposed lunar landing vehicle. Lockheed conducted parachute drop tests in which the landing load was applied at angles up to 20° with the tube axis. In all cases the tube and mandrel operated in its primary mode.

There are ways, should the need arise, to extend the range of primary mode operation by modification of the basic design. For example, the folding tube can be encouraged toward folding by putting circumferential grooves in the tube, using stiffening rings on the outer diameter, or by using a convoluted tube. The latter method has been used in energy absorbing steering columns, which have been designed to fold under large oblique loads. Other techniques, e.g., the slotting and/or tapering of the mandrel end of the tube can be used for the tube and mandrel.

g. Design of the Controlled Collapse Frontal Structure

The basic design philosophy would be to design all primarily axial load carrying structural members to be energy absorbing elements. All other structural members, such as cross members, should be designed to suffer little permanent deformation during impact, at least until the front end structure has entirely collapsed. The collapse distance of the front end would be made as large as possible. Provisions for downward deflection of the engine during severe impact must therefore be made.

1. Design Force Levels

The force levels of the individual energy absorbing elements must be determined by the square wave force-displacement response requirement at the passenger compartment. The forces acting on the passenger compartment at any time during impact are the sum of the inertia forces, the forces necessary to deform the sheet metal, and the operating forces of the energy absorbing elements which are actively dissipating energy.* The operating force level of a particular element is therefore the difference between the design force on the passenger compartment and the inertia and sheet metal forces,* divided by the number of elements which are functioning at the same time.

Because of the complicated nature of the structure interposed between an individual energy absorbing element and the passenger compartment, it is likely that any variations of an elements force-displacement curve from a square wave will be attenuated before it reaches the passenger compartment. Thus, the initial spike and force oscillations which occur during tube folding will probably not be reproduced in the passenger compartment response curve unless the collapsing tube location is adjacent or nearly adjacent to the firewall. This implies that for the purpose of determining system response under impact, some of the energy absorbing elements may be treated as constant force devices.

2. Controlled Frontal Structure for a Late Model Luxury Auto

We shall now apply our design philosophy to the front end of a late model luxury auto. We shall attempt to place the energy absorbing elements in the same location as the current load carrying members to minimize redesign of the remainder of the front end nonstructural components.

The current frame is shown in Figure 9.5.2, and the proposed energy absorbing frame is shown in Figure 9.5.3. A cross hatched member is an energy absorbing element, either a folding tube or a tube and mandrel device. All other elements have been stiffened so that they suffer little plastic deformation during front end collapse. In particular, the front cross member has been considerably stiffened to prevent intrusion under the application of localized forces. The front cross member acts as a beam supported at the attach points for the energy absorbing elements. Intrusion will be prevented in all impact situations if the front cross member, under a normal load applied at the midpoint of its span, remains elastic until the energy absorbing elements have been activated.

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* If the front wheels are being crushed at this time, then the wheel crushing force must also be included.

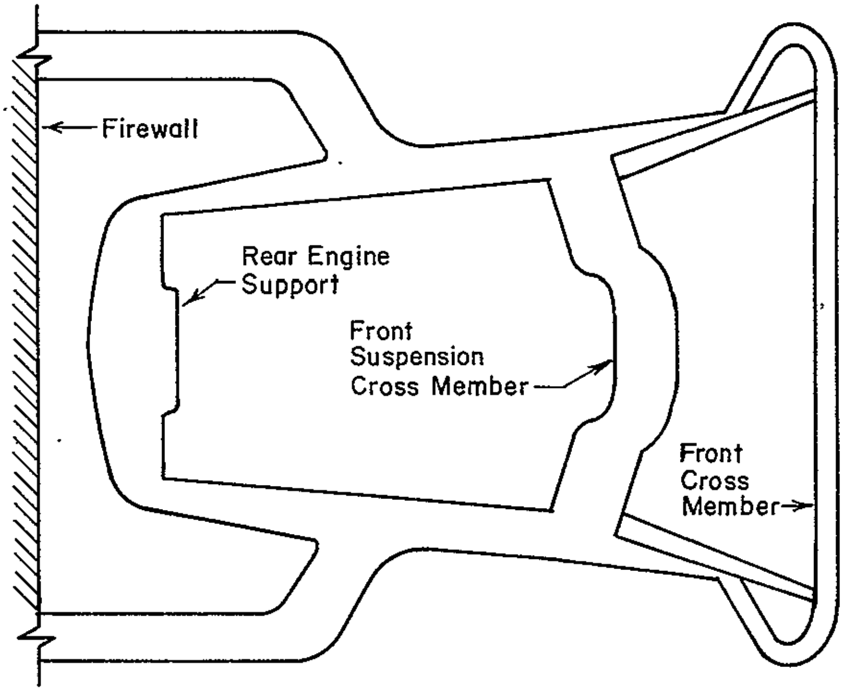


Figure 9. 5. 2 Standard Frame

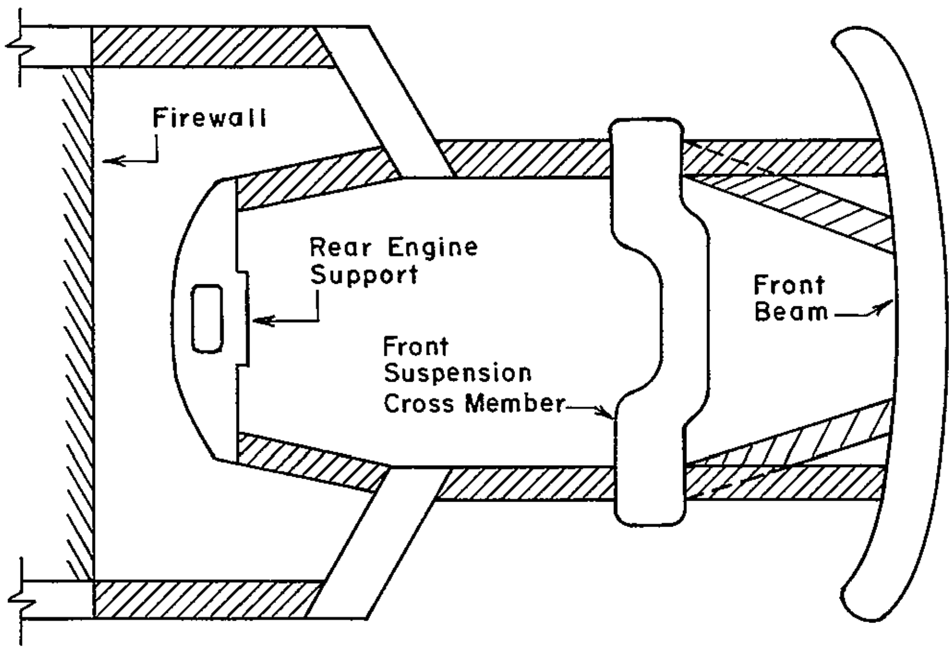


Figure 9. 5. 3 Energy Absorbing Frame (1)

The firewall must also be stiffened to prevent intrusion into the passenger compartment. The firewall must also support structural members whose purpose is to deflect the engine under the passenger compartment during impact so that additional stopping distance can be provided. The details of downward deflection of the engine have been solved in two ways by the Cornell Aeronautical Research Laboratories - using diagonal beams and rails to guide the engine under the passenger compartment. Either of these can be used in our design.

The design of the individual elements must be governed by the results of the analytical and experimental analysis coupled with the system response computer program. Whether or not the simple design configuration we have shown will be satisfactory exactly as we have shown it cannot be answered at this time.

A design configuration which involves some minor geometrical changes, but which appears to be more satisfactory is shown in Figure 9.5.4. In this design there is increased multi-directional load capability near the front bumper. With this type of structural arrangement there are certain to be some energy absorbing elements nearly aligned with the direction of load application in oblique impacts.

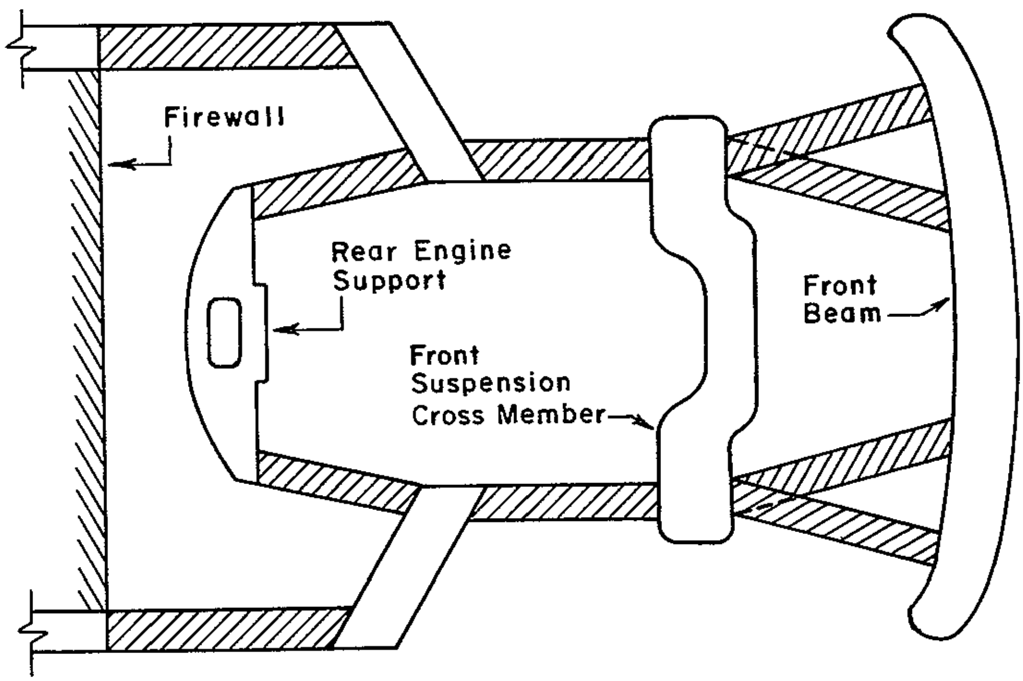


Figure 9.5.4. Energy Absorbing Frame (2)

Since cylindrical tubing is not as efficient a beam section as some of the structural elements now being used in front end design, we believe that there will be some increase in the weight of the front end. Stiffening of the cross members will also result in some additional weight. The total weight increase will probably be small, however, compared to the total front and weight so that the center of gravity and handling characteristics will not be significantly affected.

REFERENCES

1. Haddon, William; Traffic Safety A National Problem, 1967, The Eno Foundation for Highway Traffic Control, Saugatuck, Conn. page 1.

2. Lundstrom, L. C.; Cichowski, W. G.: Field Experience with the Energy Absorbing Steering Column. S.A.E. Paper 690183. January 13-17, 1960.

3. Frost, J. D.; U.S. Patents Nos. 3,250,065 and 3,370,886, also private correspondence and conversations.

4. Foster, David; Address to a public meeting convened by the National Highway Safety Bureau in Washington, August 27, 1969.

5. Frost, J. D.; Demonstration witnessed by W. H. Parks.

6. Campbell, Horace, M.D.; Private conversation with W. H. Parks.

7. New York State Department of Motor Vehicles, The Safety Sedan, Document FHR 3526-1 prepared by Fairchild Hiller, 27 November 1967, Page 6-1.

8. Baracos, N.; Rhodes, A.; Ford "S" Frame, SAE Paper 69004, Page 7.

X. SEVERE EARTHQUAKE PROTECTION APPLICATIONS

High-rise construction in earthquake zones is increasing steadily. Methods for designing multistory frames to resist seismic forces follow code regulations and are fairly well established. However, they are usually based on the most probable severity of quake over a certain period of time only, say 40 to 50 years. No provisions are included for higher lateral loads. It is our opinion that some way of resisting the latter type of loads should be included in the multistory structures if the added costs can be kept low. The result would be a dramatic decrease in structural damage and lost lives as well.

1. Proposed Device #1

Figure 10.1 shows a typical elevation of multistory frame. The dashed line shows the configuration due to lateral sway under horizontal seismic loads.

In a typical bay ADBC, see Figure 10.2, the diagonal AB increases in length from AB to A'B'. This increase is variable and depends mainly on the story level.

The tube and mandrel device described schematically in Figure 10.3 is suggested for installation on the diagonal AB. Under normal seismic loads the device will not work. However, under very severe tremors the lengthening of AB would be such that the device will function and absorb extra energy.

Two arrangements might be used for this purpose as shown in Figures 10.4 and 10.5, depending on the architectural constraints.

2. Proposed Device #2

The recent issue of "Engineering-News Record," dated May 21, 1970, describes a ball-bearing system to eliminate superstructure stress during quakes. The system is designed by the California Consultant Marc S. Caspe and shown in slightly modified form in Figure 10.6. We believe that this system could be greatly improved by the addition of tube and mandrel energy absorbing devices as shown in Figure 10.6. The reason is that the force level for such an energy absorbing device can be predicted with good accuracy.

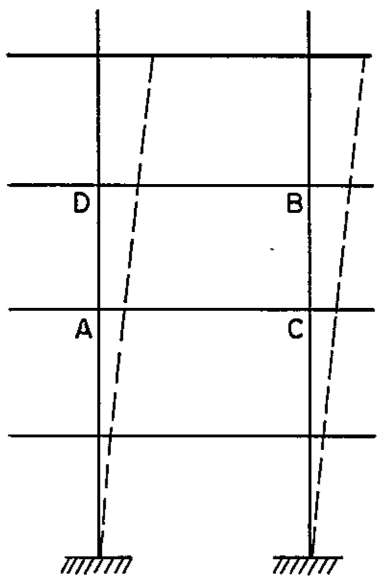


Figure 10.1

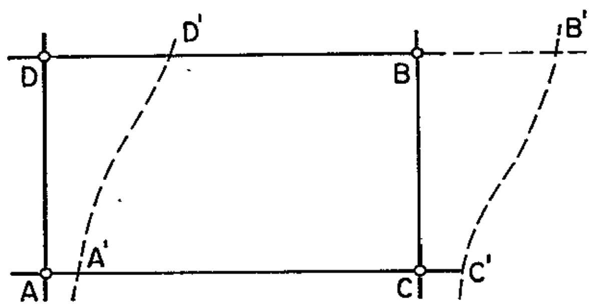


Figure 10.2

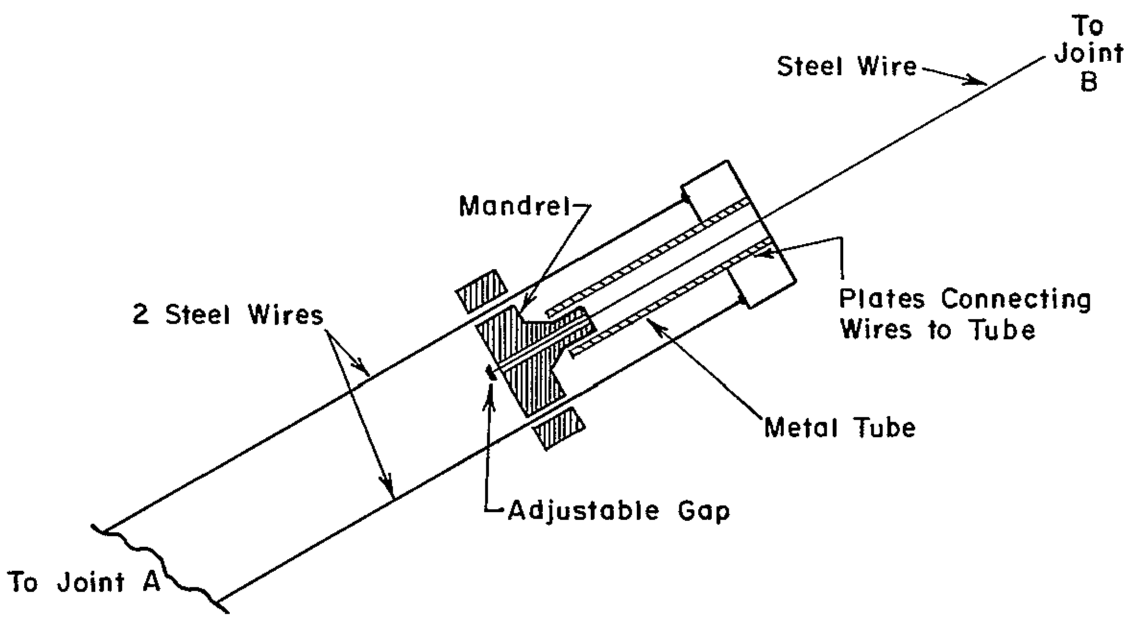


Figure 10.3

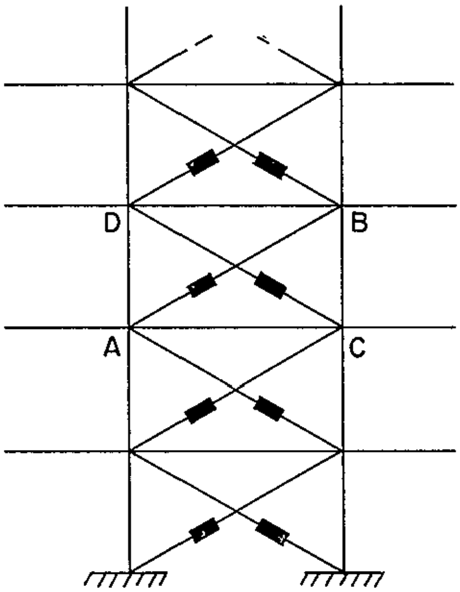


Figure 10.4

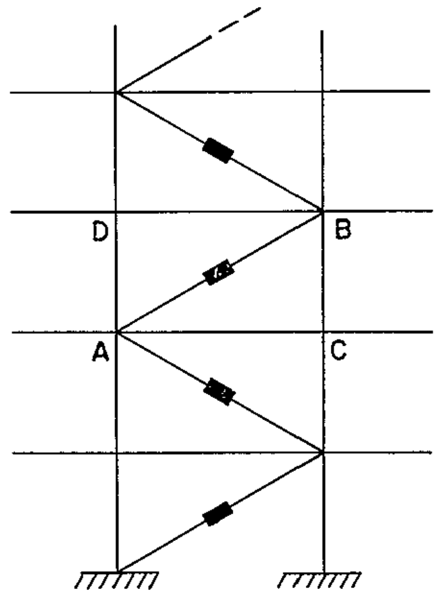


Figure 10.5

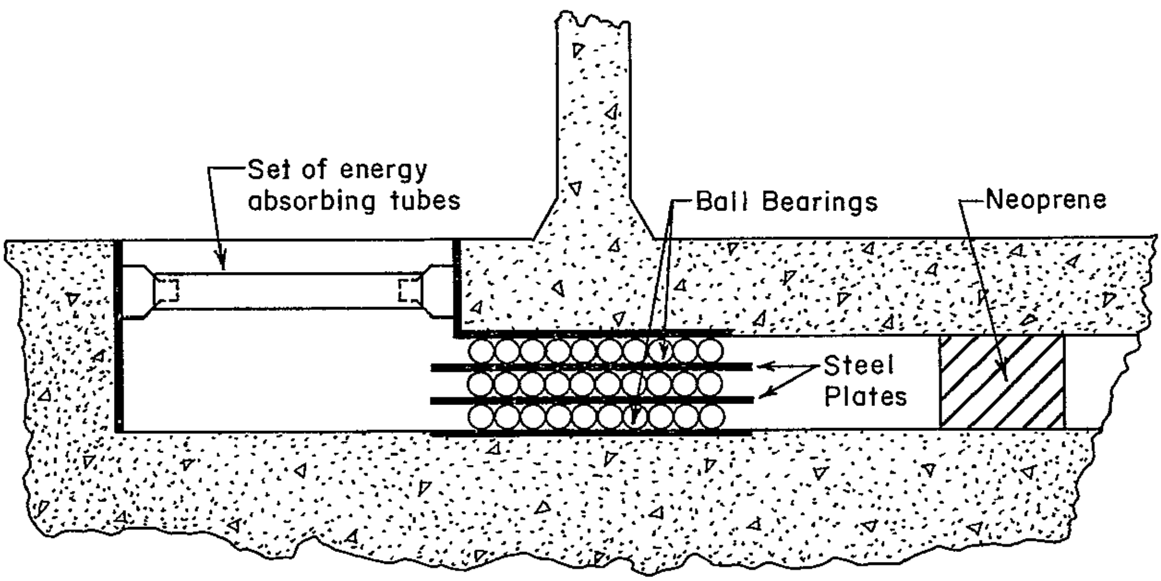


Figure 10.6

3. Conclusions

Further detailed investigation will be carried out to determine the range of application of device #1 in seismic resistant structures. Such research might start by examining existing structures and their structural behavior if such device had been included in the framework. It is our opinion that such investigation when completed will be highly beneficial to the state of art in seismic design.

Furthermore, contacts will also be made with the designer of device #2 to examine the possibility of proposed improvements in the present design.

A common feature for the energy absorbing tubes in both devices is the low cost of replacement after very severe quakes.

Visits will be made to consulting Engineering firms having an interest in the design of large buildings in earthquake zones and to steel manufacturers who may be interested in manufacturing and marketing the device.

XI. TECHNOLOGY TRANSFER VIA STARTING A NEW COMPANY

In some instances the Technology transfer process may be expedited by forming a new company to make and sell a product embodying the Technology. This section presents information relative to the steps required to establish a small manufacturing company to produce a commercially marketable device that has been evolved from an unused NASA patent. Because of the many variables involved in such an activity, the presentation will be made in general terms and guidelines rather than absolute instructions that must be followed. The subject is presented in the form of an example using the energy absorbing device developed for use in the semitrailer support system as the assumed product.

A. Product Description

The product is an energy absorbing device that is intended to be used on semitrailers (See Section VI). It eliminates damages caused by accidental dropping of units that are unhooked from the tractor without first properly lowering the front end support structure. The device is installed as a portion of each lower support column, and collapses in a predetermined manner under the extra loading caused by dropping the trailer. The deformation of the unit absorbs the extra energy thereby protecting the trailer from damage. The nature of the device is such that is essentially destroyed by being used and therefore must be replaced. This apparent disadvantage actually helps because the driver cannot conceal the fact that he has dropped the rig. Not only are the deformed parts visible, but the design is such that the wheels fall off when the dolly is raised above the ground. Replacement of the two devices is accomplished by fastening the dolly wheels and axle to the new units, unbolting the old ones from the shafts and replacing them with the new ones. The total operation can probably be accomplished in less than 30 minutes.

The product itself can be described as a tube and mandrel, plus attaching hardware. The design (See Section VI) is such that when sufficient loading is applied, the welds between the tube and mandrel are broken. This allows the mandrel to be driven up into the tube. The latter is bevelled and prenotched to fracture in eight places under this impact. The tube walls deform into circular rolls because of the mandrel shape. The fracturing and rolling of the tube walls continue until the deforming has absorbed the energy resulting from the dropping of the semitrailer. The device is constructed of mild cold rolled steel.

B. Marketing Information

The Fruehauf Trailer Corporation has indicated an interest in offering this product as an optional device on their semitrailers. They would also probably install them on trailers repaired under warranty where the damage indicated that the rig had likely been dropped. Fruehauf also builds support structures for other trailer manufacturers, and they would therefore be able to offer these devices to the entire market. A company representative estimated their requirement at 20,000 units (10,000 pairs) per year. This is approximately 20 percent of the theoretical available market and appears to be a good conservative estimate to use of projecting requirements for the manufacturing company. The total semitrailer market in the U.S. is very stable and should be at least 100,000 new units per year for quite some time. All indications are that the industry will continue to grow during the coming years.

No other market for this product has been established at this time. Possibilities do exist, but they have been ignored for the purpose of this report. The sales volume indicated above is sufficient to adequately support the formation of the new company. It is assumed that all devices produced would be sold to Fruehauf for resale to their customers. No in-house sales effort will be anticipated.

C. Company Organization

The company will be formed as a separate corporation with no legal ties to either DRI or DU. This will provide liability protection to both institutions and the officers of the company. It is assumed that persons from DRI, DU and possibly Fruehauf would serve on the Board of Directors along with any major private investors and the President of the company.

D. Company Personnel

Responsibility for all operations of the company will be vested in the President, who will report to the Board of Directors. In addition to directing the overall performance of company activities, the President will also be the contact with the customer, Fruehauf Trailer Corporation. He will be assisted by a lawyer and a certified public accountant as required. The Board of Directors will provide general guidance to the President and establish overall company policy, goals and basic methods of operation. The company's record of performance will be periodically reviewed by the Board of Directors.

Day to day direction of the manufacturing operation will be the responsibility of the Plant Foreman, who will report directly to the President. This person must be familiar with the operation of machining equipment, an arc welding facility and a spray painting installation. He will establish detailed operating procedures for these three shops and perform inspections to insure that the proper steps are being performed. A machinist, an arc welder and a painter will be hired by the foreman, plus three laborers to assist throughout the manufacturing operation.

A secretary will be hired to provide clerical and typing support to both the President and the Plant Foreman. In addition, she will make entries into the books for the accountant, answer the telephones, act as the receptionist and coordinate purchasing activities.

E. Plant Operation

The basic device consists of the following three parts: a top plate, the tube and a mandrel. A fourth part to hold the dolly axle would be provided by Fruehauf. The top plate will be stamped out by the supplier and will be ready to use when received. Initially, the tubes will be cut to the proper length by the vendor and then bevelled with a lathe in the company shop. Later, the cutting task may also be performed in-house. The bevelled end of the tubes will also be notched using a small press and a specially designed tool for this task. Mandrels will be purchased as thick-walled tubing cut to the proper length. They will be machined to the proper shape on a lathe. A hydraulic drawbar will be used to hold the piece being machined. This technique will double the output by cutting down on set-up time.

Assembly of the device will be performed next. The tube will be given a light coating of lubricant just inside the bevelled area and then it is mated with the mandrel. Eight filet arc welds will hold the two parts together. Then the top plate is fastened to the tube with a continuous arc weld. The support axle holding piece is attached to the other end of the device in the same manner. All welding will be performed with a hand held automatic welding gun that is designed to provide a much more uniform quality than can be achieved with a manually controlled system.

The final major task to be performed is painting. This will be accomplished with a spray gun in an explosion proof booth. A coat of zinc chromate will be applied and heat lamps will be used to speed up the drying process.

Tools, jigs and fixtures will be used throughout the entire manufacturing operation to simplify the tasks performed, to insure that the parts are identical, and to provide a quick check on the quality of each manufactured part. Detailed inspection of hardware will be performed on random samples selected from the production line. Destructive testing of a few completed assemblies will be performed on a regular basis. Normally, in-process inspections will be performed by the Plant Foreman. Also each operator will check the hardware processed by him. Destructive testing will be performed by DRI personnel.

Completed units will be packaged two to a box and stored for shipment to Fruehauf.

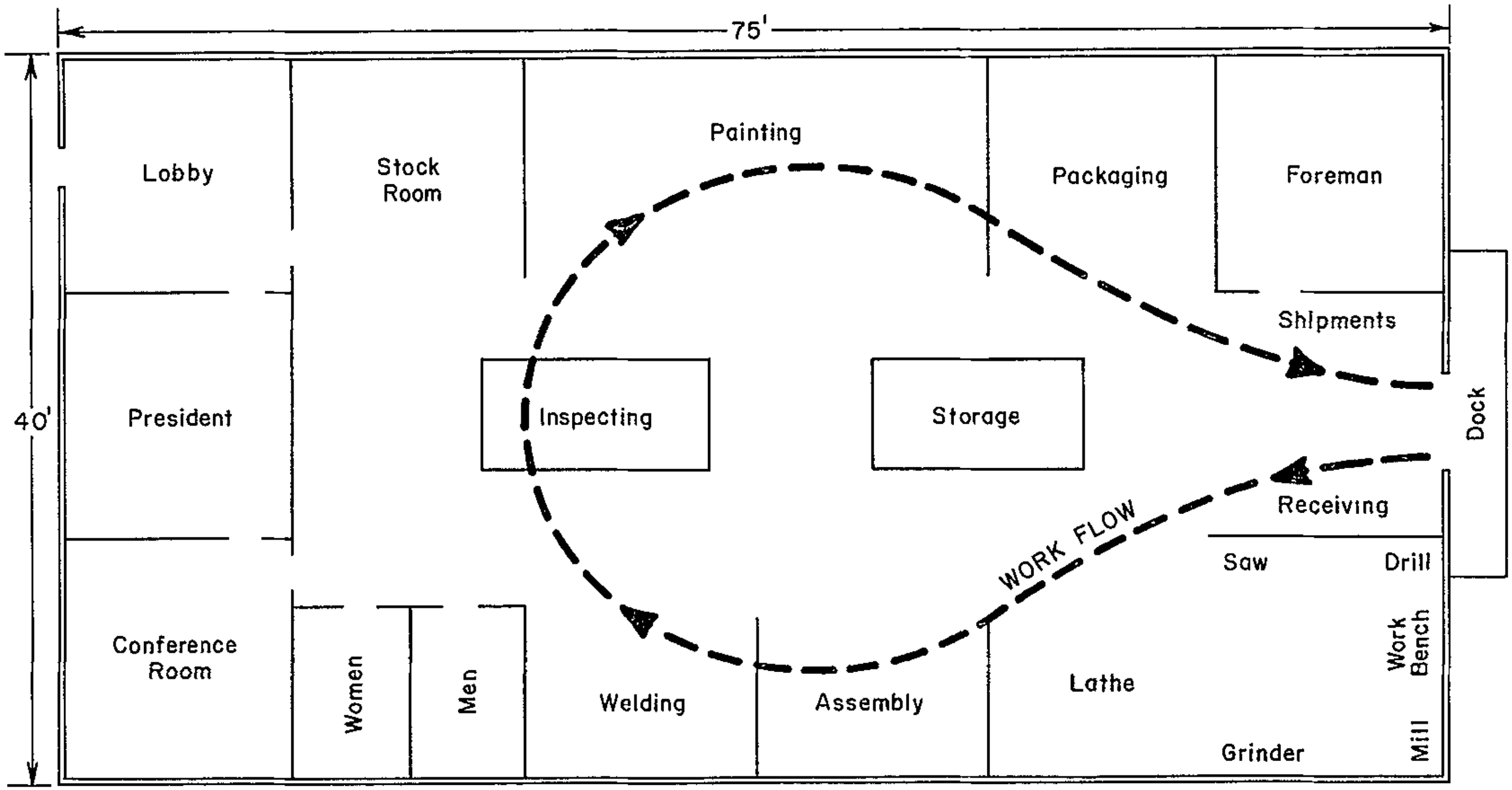
A possible layout of the plant is shown in Figure 11.1. The final arrangement of such a plant will be very dependent on the actual building selected for the company. The circular flow shown is a very efficient work pattern for the facility. However, since the material being processed is relatively light and small, other arrangements could be used without serious reductions in plant efficiency.

F. Production Quantities

As previously stated, the output of this manufacturing company needs to be 20,000 units a year. This means that 10 units an hour must be produced, a rather large amount. To accomplish this, the equipment must be good, so as to minimize breakdowns; the tooling, fixtures and jigs must be well-designed to facilitate quick material handling; shop procedures and layout must also not consume excessive time; and since some operations take different lengths of time, proper coordination of all of the manufacturing efforts is mandatory. The Plant Foreman will be specifically responsible for normally maintaining an output of at least 400 units a week. During vacation periods, overtime may be required to achieve this result. It should be possible to produce 500 units a week when the full labor force is available.

G. Manufacturing Company Start-Up Activities

It is anticipated that a period of about three to four months will be required to set up the company and put it into full operation. During this period a considerable amount of money will be required, but no sales will be realized since the plant will not be in production. To insure that sufficient funds are available, the total cost of start-up, plus three months operation, should be available when the company is formed.



PLANT FLOOR PLAN

Figure 11.1 Plant Floor Plan

The following major items should be accomplished just prior to and during the initial three month period:

1. Agreement to contract executed with Fruehauf
2. Licensing agreement executed between DU and NASA
3. Financing finalized with private parties and MESBIC or SBA
4. Incorporation accomplished
5. Royalty agreement executed between company and DU
6. President, Plant Foreman, Laborer and Secretary hired
7. Accounting system established
8. Purchasing system established
9. Insurance programs executed
10. Engineering field testing accomplished
11. Basic operating procedures established
12. Final design established
13. Final contract with Fruehauf executed
14. Plant machinery ordered
15. Suitable building located and lease executed
16. Necessary plant modifications accomplished
17. Plant furniture and small equipment ordered
18. Manufacturing materials ordered
19. Welder and one more laborer hired
20. Pilot production run conducted
21. Evaluation testing accomplished
22. Revisions to manufacturing processes and procedures made as required
23. All remaining employees hired
24. All preparations completed for full production operation

The items are presented in their approximate order of occurrence. The engineering testing and establishment of final design must be accomplished by the time shown; it will probably happen sooner than that.

H. Start-Up Costs

An estimate has been prepared for the costs needed to establish the Energy Absorber Manufacturing Company. At this time, some of the figures cannot be finalized for a variety of reasons. It is felt, however, that the overall magnitude of the estimate is reasonably accurate and that it can certainly be used for planning corporate financing arrangements.

No attempt will be made to explain each item. The estimates have been made after conversations with the appropriate companies and government organizations. It is assumed that the building and major shop equipment will be leased. All other items will be purchased on a cash basis and immediately expensed in the company records. In estimating the start-up costs, the President's salary is assumed to start one month prior to incorporation. Naturally, the lawyer must also be retained prior to the formal beginning of the company. Other costs have been estimated for the periods when they will happen, i.e., the building will only be leased for two months, not three. DRI support is estimated to be about a three to three and a half man-month effort plus overhead.

START-UP COSTS (3 Months Plus 1 Month Prior to Incorporation)

Payroll

President 4 mos.@ $1,250		$ 5,000
Plant Foreman 3 mos.@ $850		2,550
Welder 1 mo.@ $700		700
Laborer 3 mos.@ $450		1,350
Secretary 3 mos.@ $400		1,200
		$10,800

Payroll Taxes

Unemployment (3.1% of 1st $3,000/person) ($93, 79, 22, 42, 37)		$ 270
Workman's Comp.	Approx.	122
FICA (4.8% of 1st $7,800/person)		520
Denver Head Tax ($2/mos./person)		28
		$ 940

Legal Expenses

Incorporation	$ 1,000
Licensing Agreement	400
Fruehauf Contract	800
General Support	200
Total Legal Cost	$ 2,400

Accounting System Costs

Establishing System	$ 1,000
Two months support @ $125	250
	$ 1,250

Building Lease

(10¢/ft^2/month) 2 mos.@ $300	$ 600
Janitorial Service during start-up efforts	200
Heat, Electricity and Telephone 2 mos. @ $200	400

Insurance

Liability and Property	180
Medical 2 mos. @ $15/mo./employee	120
	$ 300

Shop Equipment (leased)

Lathe	$3,000	(used)		
Mill	1,500	(used)		
Arc Welder	1,500	(new)		
Press	1,000	(new)		
		Total . . .	$7,000	$ 7,000

3 year lease 1 mo. @ $260	$ 260

Shop Equipment (purchased)

Cut-off Saw	$300		
Spray gun and compressor	250		
Drill press	150		
Bench grinder	100		
Total.		$ 800	$ 800

Shop and Office Furniture

Executive Dest 2 @ $160	$ 320
Executive Chair 2 @ $60	120
Table	120
Bookcase 4 @ $30	120
Side Chairs 13 @ $30	390
Steno Desk	200
Steno Chair	40
Conference Table	140
File Cabinets 2 @ $90	180
Storage Cabinet	90
Couch	100
Easy Chair	50
Coffee Table	60
Lamp and Table	70
Work Benches 12 @ $120	1,440
Stools 10 @ $30	300
Storage Shelving 6 @ $30	180
Cabinets 4 @ $80	320
Carts 4 @ $40	160
Total Furniture. . .	$ 4,400

Office Supplies

Typewriter lease 2 mos. @ $25	$ 50
Small office equipment	150
Paper, forms, miscellaneous	300
Copying Equipment lease 2 mos. @ $25	50
	$ 550

Shop Supplies

Misc. small tools	$ 450
Jigs and fixtures	500
Welding supplies	150
Misc. supplies	300
	$ 1,400

Building Improvements

Partitions 100' @ $15	$ 1,500
Power and lights	500
Spray booth and fan	1,000
Hoods and fans 2 @ $500	1,000
Total Improvements	$ 4,000

Production Materials (Initial Stocking)

Plates 5,000 @ $1.00	$ 5,000
Tubes 5,000 @ $ .40	2,000
Mandrels 5,000 @ $ .80	4,000
Paint	100
Lubricant and misc.	100
Shipping boxes 2,000 @ $ .085	170
Packing materials and misc.	30
Total Production Materials	$11,400

START-UP COSTS SUMMARY:

Payroll	$10,800
Payroll taxes	940
Legal expenses	2,400
Accounting system costs	1,250
Building lease	600
Janitorial service	200
Heat, electricity and telephone	400
Insurance	300
Shop equipment (leased)	260
Shop equipment (purchased)	800
Shop and office furniture	4,400
Office supplies	550
Shop supplies	1,400
Building improvements	4,000
Production materials	11,400
Sub Total . . .	$39,700
DRI Support . .	7,300
Total Cost . .	$47,000

I. Production Costs

An estimate has been prepared for the first nine months of operation of the manufacturing plant. This completes the first year of operation of the firm and the books should be closed and analyzed. The start-up costs have been included at the end of the summary, along with a 10 percent contingency allowance, royalty payments to DU of $1.00 per unit and profit. A selling price of $30 per pair is shown for the first 7,500 pairs. For subsequent years, the price drops to $22 per pair. Further reduction can be anticipated if production volume is increased.

Production Effort - 9 Months

President		$11,250
Plant Foreman		7,650
Welder		6,300
Machinist @ $550		4,950
Painter @ $500		4,500
Laborers 3 @ $450		12,150
Secretary		3,600
	Total payroll	$50,400

Payroll Taxes

Unemployment (3.1% of 1st $3,000/person) (--, 14, 72, 93, 93, 237, 56)		$ 565
Workman's Comp. Welding $2.87 × (7,650 + 6,300 + 12,150 = 26,100) ÷ 100		750
Machine $1.29 × 4,950 ÷ 100 =		65
Paint $1.58 × 4,500 ÷ 100 =		70
Office $.05 × (11,250 + 7,650 = 18,900) ÷ 100 =		10
	Total Workmen's Comp.	$ 895
FICA (4.8% of 1st $7,800/person) (135 + 250 + 300 + 240 + 215 + 585 + 175)		1,900
Denver Head Tax ($2/mo./person)		162
	Total Payroll Taxes	$ 3,520

Legal Support	$ 300
Accounting support @ $125/mo.	$ 1,125
Building lease (10¢/ft²/month) 9 mos. @ $300	$ 2,700

Janitorial Service $125/mo.	$ 1, 125
Heat, Electricity, and Telephone @ $200/mo.	1, 800
Insurance	
Liability and property	750
Medical 9 mos. @ $15/mo./person	1, 215
	$ 1, 965
Shop equipment lease cost @ $260/mo.	2, 340
Office supplies	
Typewriter lease @ $25/mo.	225
Paper, etc.	400
Copy equipment @ $25/mo.	225
	$ 850
Shop Supplies	500
Production Materials (15, 000 units)	
Plates 15, 000 @ $1.00	15, 000
Tubes 15, 000 @ $ .40	6, 000
Mandrels 15, 000 @ $ .80	12, 000
Paint	400
Lubricant and misc.	400
Shipping boxes 8, 000 @ $ .085	680
Packing materials and misc.	120
Total Material	$34, 600
Property Taxes Est. 75 mil on 28% of 60K value	$ 1, 350

PRODUCTION EFFORT (9 Months) - SUMMARY

Payroll	$ 50,400
Payroll Taxes	3,520
Legal Support	300
Accounting Support	1,125
Building Lease	2,700
Janitorial Service	1,125
Heat, Electricity and Telephone	1,800
Insurance	1,965
Shop Equipment Lease Cost	2,340
Office Supplies	850
Shop Supplies	500
Production Materials	34,600
Property Taxes	1,350
Sub Total.	$102,575
DRI Consulting	14,000
Sub Total	$116,575
Start-up Cost	47,000
Sub Total	$163,575
Contingency	16,358
Sub Total	$179,933
DU Royalty @ $1.00/ unit	15,000
Total Cost.	$195,000
Profit	30,000
Total Sales	$225,000

for 15,000 units; single price $15
Pair price $30

SUBSEQUENT YEARS (12 Months) - SUMMARY

Annual Cost $116,575 \times \frac{12}{9}$		\$156,000
Contingency		15,600
DU Royalty @ \$1.00/unit		20,000
	Total Cost.	\$191,600
	Profit.	28,400
	Total Sales	\$220,000

for 20,000 units: Single price \$11

Pair price \$22

J. Miscellaneous Comments

Several clarifying comments on a wide variety of subjects need to be made. This subsection presents them.

All shipments are assumed to be made FOB Denver. This means that shipping costs and any insurance will be paid for by Fruehauf. This is standard practice, and it greatly simplifies estimating the costs for shipments when multiple destinations are required.

The Plant Foreman will have to carefully monitor the supply of production materials on hand and initiate procurement action in a timely manner so as to avoid shortages. The quantities and types of items needed are such that significant lead time is required to obtain delivery. Since the company's building will be fairly small, huge investions (like a six month's supply) cannot be maintained.

The nature of the production operation is such that there should be very little stealing of production materials by employees. Tools and supplies will be a different story and careful attention should be paid to safeguarding these items.

Each employee will earn two weeks vacation, six paid holidays and be eligible for five days of sick leave. The latter is a privilege and care must be taken to see that it is not abused. A minimum Blue Cross-Blue Shield group medical policy will be provided by the company for all employees.

K. Operating Capital Requirements

To satisfactorily meet its obligations during the first six months of operation, the company will require a considerable amount of funding. This money amounts to the $47,000 start-up cost plus a third of the $116,000 needed to cover three months of production. The bare minimum amount is $85,000; a figure of $100,000 is recommended.

XII. LIST OF PAPERS GIVEN AND INTERNAL REPORTS WRITTEN

Several papers have been given and internal reports written on different aspects of this project by students, faculty, and research staff.

The following papers were given at meetings and seminars:

1. Patent Development: Its Use in Project-Type Instruction, Wilbur H. Parks, Presented at the American Society for Engineering Education Annual Meeting, Pennsylvania State University, University Park, Pennsylvania, June 23-24, 1969.

2. Evaluation Criteria for Highway Gore Buffering Systems, Michael A. Kaplan. Presented at the Bureau of Public Roads, California Department of Highways Workshop on Energy Absorption Barriers, Sacremento, California, July 15-16, 1969.

3. Space Technology and Auto-Highway Safety, M. A. Kaplan, R. J. Hensen, and R. J. Fay. Presented at the Annual Meeting of the Highway Research Board, Washington, D. C., January 12-16, 1970.

4. Highway Gore Area Buffers, Lance Lindeman. Presented at the American Society of Civil Engineers Student Paper Contest, University of Colorado, Boulder, Colorado, April 24-25, 1970.

5. Scale Model Test of the BPR Modular Crash Cushion Barrier, Richard J. Fay, and Edward P. Wittrock. Presented in the Mechanical Sciences Seminar Series, University of Denver, Denver, Colorado, April 24, 1970.

The following papers on different technical aspects of the project have been prepared by engineering students and faculty in the course of the past years work:

	Title	Author
1.	Tube Collapse	Tom Miller
2.	Shearing of Sheet Metal as a Means of Energy Absorption	Einer Fernholt

	Title	Author
3.	The Second Collision	Leonard Hychalk Gregory Batten
4.	Patent Project Management Control	Edward Binggeli
5.	Automobile Configuration and the Use of Energy Absorbers	Douglas Freiburger Joseph Lemaire
6.	Automobile Frames	Douglas Freiburger Joseph Lemaire
7.	Extrusion Device and Tube and Mandrel	John Freeman Guy Harvey
8.	Patent Group II Progress Report	Jose Reyes
9.	Final Report: Senior Civil Engineering Project - The Patent Project	Larry Buyck
10.	Final Report of Highway Buffer Design Group	Larry Buyck Larry Gallegos Christopher Moore
11.	Planning and Controlling Multidisciplinary Research and Development Projects	Christopher Moore
12.	Test Rig Release Assembly	James Baddaker John Dean Robert Jaedecke
13.	A Statistical Summary of Current Automobile Accident Data	Douglas Freiburger
14.	A Description of the Status Report Forms	Edward Binggeli
15.	Preliminary Comments on Highway Safety Market	J. G. Milliken
16.	How Are We Doing on the Patent Development Project?	W. H. Parks

	Title	Author
17.	Testing Activities, Summer 1969	John Freeman
18.	Energy Absorbers (Report Nos. 1, 2, 3, 4, 5, 6)	Edward Wittrock Roy Olsen
19.	Incorporation of an Energy Absorber into Transport Trailer Support Wheel Structure	Gregory Batten
20.	Model Crash Testing, Interim Report	Joseph De Amicis Barry Gilbert
21.	Energy Absorbing Safety Seat	Leonard Hychalk
22.	Progress Report - Study on Patents in Chemical Engineering Area	Michael King
23.	The Automobile Crash	Clark Lee
24.	Tube and Mandrel Patent Development	Kenneth Cook
25.	Progress Report, Marketing and Manufacturing Study	Joseph Peckis
26.	Engineering for Innovation (13 minute, color and sound 16 mm Motion Picture based on the patent development project)	John Dinkmeyer, Director Harry Spetnagel, Script Writer
27.	Folding Tube Testing	Bruce Burr
28.	The Kaplan Barrier	Lance Lindeman Phillip Overeynder
29.	The Application of Energy Absorbers to the Ski Industry - Market Survey	Joseph De Amicis Barry Gilbert Paul Rexroth
30.	Scale Model Auto Launching System	Joseph De Amicis
31.	Scale Model Auto Launching System - Air Cylinder Design	Barry Gilbert

	Title	Author
32.	GMC "Black Box" Problem	Joseph Lemaire
33.	Tube and Mandrel Testing	Kenneth Cook
34.	Gore Area Buffer Design	Lance Lindeman Phillip Overeynder
35.	Energy Absorbing Seat for Automobile	Leonard Hychalk
36.	Forming a Small Manufacturing Business	Joseph Peckis
37.	The Automobile Crash	Clark Lee
38.	Scale Model Test of the BPR Modular Crash Cushion Barrier	Richard Fay Edward Wittrock
39.	Project 4114 Library Information System	J. G. Milliken

The following papers were prepared by the non-engineering students participating in this project:

	Title	Student	Affiliation
1.	Project 4114 Library Information System	Carol Coltrane Doris Maeser	Graduate School of Librarianship
2.	Evaluation of Information Needs During Research	Doris Maeser	Graduate School of Librarianship
3.	The Budget Summary and an Analysis by Account	Roger Squire	School of Business Administration
4.	Final Report, Marketing Group	Tony Hopper Elnar Hunsager Joseph Peckis Roger Squire	School of Business Administration
5.	Preliminary Market Analysis of Energy Absorption Applications	Roger Squire	School of Business Administration

XIII. PROPOSALS WRITTEN IN RELATED AREAS

Several proposals have been written to governmental agencies during the year in the areas of energy absorber applications, energy absorber performance evaluations, and technology utilization. The proposals written are as follows:

	Title	Agency	Amount
1.	Vehicle Crash Testing Methodology and Facility	Federal Highway Administration, National Highway Safety Bureau	$466,695
2.	Center for Innovation and Technology Transfer	National Science Foundation	237,151
3.	Methods of Predicting Performance of Energy Absorbers in Automobile Collisions	Insurance Institute for Highway Safety	34,914
4.	Human Engineering of Restraint Systems	Federal Highway Administration, National Highway Safety Bureau	99,680
5.	Development of Controlled Collapse Frontal Structures for Representative American Compact and Luxury Automobiles	Federal Highway Administration, National Highway Safety Bureau	315,600
6.	Collapsible Steering Column Testing Device	Federal Highway Administration, National Highway Safety Bureau	32,891

XIV. TECHNICAL INFORMATION SYSTEM

The project's system for gathering, evaluating and retrieving technical information has been developed through the interaction of graduate students in Librarianship and other students and faculty assigned to the project. The information system has been established and certain modifications made to eliminate imperfections, by the end of the second year.

Professional talent for the technical information system has been provided by three students from the Graduate School of Librarianship. Participation in the project permitted these students to complete the research paper (a form of master's thesis) required for graduation. The students, each of whom had an interest in a special (e. g., specialized engineering or science) library field, expressed satisfaction with the opportunity for the interdisciplinary contact provided.

The information system consists of the following:

1. A file of some 250 entries containing specialized books, reports and articles related to project activities. Actually there are substantially more than 250 individual documents in this collection because in many instances several related documents have been placed in a folder and filed as one entry.

2. A collection of 67 reports written by individual students or groups of students reporting their work on the project. Writing these reports is an important part of the students' educational experience. They serve important functions as a record of things which have been accomplished and in the exchange of information between the many individuals working on the project.

3. A research card index containing 2,000 eight-by-five cards, allowing retrieval of information by author(s), title, and subject on some 500 documents located in the project collection or in another library of the University or in the Denver metropolitan area. Each of the 500 documents has been reviewed by a participant in the project and has been classified according to one or more (usually several) subject categories. The categorization system is extensive, involving five major categories and about 60 subcategories within each.

4. A collection of about 800 patents (over 50 in the area of energy absorption) related to the subject areas of the project.

5. A collection of some 50 newsletters issued to date in this project. These give a chronological record of the content of class sessions and related activities and events.

The information system, while complex, permits a project participant to conduct exploratory research in many areas of interest. It is designed to promote widespread awareness of information, a critical evaluation of it, and ready retrieval. Many of the oversize (8" × 5") library cards contain abstracts of the documents, photocopied from abstracting publications, so the researcher can form an immediate opinion of the document's worth. Exhibit 14.1, a description of the information system prepared for the instruction of project participants, contains additional detail on the mechanics of the system.

During the spring and summer quarters of 1970, another graduate student in librarianship has joined the project. She plans to analyze the project technical information system in light of experience to date, and to propose ways to improve it and extend its effectiveness when used in new subject areas.

EXHIBIT 14. 1

LIBRARY INFORMATION SYSTEM

The following description of the project information system has been prepared to promote maximum usefulness of the system to Project 4114 participants. Suggestions or questions should be addressed to Professors Parks or Milliken.

The System Consists of:

1. A 2-part library card index located just outside Professor Parks' office in a 2-drawer gray file box. This index contains 8" × 5" cards with descriptions of all reviewed library materials. In the left drawer, the alphabetical file, you can locate materials by

   a) author;
   b) title;
   c) major subject. (A list of subject headings is posted next to the file box.)

   In the right drawer, the classified file, you can locate materials by functional category (analysis, design, test, fabrication, and marketing) and by about 60 subcategories. (A list of these category classifications also is posted next to the file box.)

2. A file cabinet, located outside Professor Parks' office, containing articles or books owned by the Project. These materials are numerically arranged, and bear a number with a prefix "PD-4114-." These materials are to be checked out by Mrs. Duykers.

3. An "in-process" library card index located outside Professor Parks' office. This index contains 8" × 5" cards with partial information on all unreviewed library materials. These cards are filed alphabetically by author (or, if no author shown, by title).

To Find Information:

1. Search the 2-part card index outside Professor Parks' office. These cards show:

   a) the subject matter of the material;
   b) an evaluation of its usefulness;

c) where the material can be located;
d) where an abstract of the material can be found.

2. Sometimes the material is located in a library other than at DU (e.g., CU Medical Library, Denver Public Library). If you want to review the material, but cannot conveniently visit the library where it is located, fill out a "Request to Buy or Borrow Library Materials" form (see below).

If You Learn of New Material:

If you encounter a reference to any library material which might be useful to our project, please describe it as fully as possible on a new 8" X 5" library card (i.e., include complete bibliographic citation, if possible, and indicate where it was referenced or where it might be located). These cards are available outside Professor Parks' office. Please give the completed card to Mrs. Duykers. If you wish the material purchased or borrowed and sent to you, fill out a "Request to Buy or Borrow Library Materials" form (see below).

When you obtain a copy of any new document, please alert the rest of the staff by giving the document to Mrs. Duykers for recording and reference in the weekly newsletter. The document can be immediately checked out to you once it is recorded.

To Order Library Materials:

To order any books or articles which are not available at a DU library, complete one copy of a "Request to Buy or Borrow Library Materials" form, found outside Professor Parks' office. The form should contain as complete a bibliographic citation as possible (or be clipped to a new 8" X 5" library card--see previous section). Requests to borrow materials are processed directly. Requests to buy materials are first approved by a faculty member on the Project. Normally, when requested materials are received, they will be sent to you for review. (See next Section).

When you Review Materials:

As participants in the NASA Patent Project you are the best qualified to review documents for usefulness on the project and your assistance is appreciated. Clerical personnel will perform reviews when no participant is available. You are requested to follow these instructions when reviewing literature, to assure that our library records are as useful as possible to other (and future) participants:

1. Each time a report, journal article, or book is reviewed, please complete an 8" X 5" library card. Copies of cards may be obtained from Mrs. Duykers. Please check to see if a card already exists in the card index, and add revisions or personal comments to make it more useful or complete. Otherwise, prepare a new card and it will be added to the file.

2. In the top section of the card, enter a bibliographic citation (author, then title of item).

3. Under "Subjects" enter as many subject headings as are appropriate to indicate the scope of the item reviewed. Use subject names that seem meaningful to you, which may or may not correspond with the printed subject heading list prepared by the Library group.

4. Circle the appropriate term showing your evaluation of the usefulness of the item to Project 4114.

5. Enter your name as reviewer.

6. Indicate in "Location" the library or file where the item can be found.

7. If you encounter an abstract (such as in Engineering Index), indicate the volume and page number where the abstract can be found.

8. Under "Comments or Summary," add any significant comments which you think are helpful to other project participants. This may include a brief summary of content, important data included, etc.

9. Place the completed card on top of the card index file.

XV. PROPOSED PROGRAM ACTIVITIES FOR THE THIRD YEAR

Since the funding level will start decreasing in June 1970, and since considerable momentum has been generated over the past two years in the exploitation of energy absorbing patents, it is planned to spend the remainder of this effort in this technical area. Work will continue in the application areas which have been defined including the semi-trailer application, the elevator application, auto safety applications, and auto-highway safety applications. Other potential applications will be sought, especially those in which the results of previous work can be directly applied.