AD

USAAVLABS TECHNICAL REPORT 66-50

AIRCRAFT CARGO RESTRAINT SYSTEM

By

A. Russo

September 1966

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-68(T)

ALL AMERICAN ENGINEERING COMPANY

WILMINGTON, DELAWARE

Distribution of this document is unlimited

Disclaimers

The findings in this report are not to be construed as an official Department of the Army position unless so designated by other authorized documents.

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the Government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission, to manufacture, use, or sell any patented invention that may in any way be related thereto.

Trade names cited in this report do not constitute an official endorsement or approval of the use of such commercial hardware or software.

Disposition Instructions

Destroy this report when no longer needed. Do not return it to originator.

DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

The need for an improved cargo restraint system for use in Army aircraft forms the basis of this study. This activity concurs with the approach used and conclusions drawn in this report. It is not felt, however, that the proposed restraint device represents the optimum design of a load-limited system.

This activity is continuing its investigation of air cargo restraint requirements and will proceed with the development of a 5,000-pound and a 10,000-pound load-limited system in the near future. Programs to provide a better definition of the acceleration/time relationship for Army fixed-wing aircraft accidents are also planned.

Project 1F121401A254
Contract DA44-177-AMC-68(T)
USAAVLABS Technical Report 66-50
September 1966

AIRCRAFT CARGO RESTRAINT SYSTEM

by

A. Russo

Prepared by

All American Engineering Company
Wilmington, Delaware

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

Distribution of this
document is unlimited

SUMMARY

This report is written as a fulfillment of contract DA44-177-AMC-68(T), which pertains to a cargo restraint system.

The purpose of the report was to investigate the restraint requirements for Army cargo in Army aircraft and to present realistic criteria for cargo restraint systems with the formulation of feasible restraint concepts. This was accomplished in the following manner:

1. By defining the restraint problems that exist in methods presently adapted to restraining cargo.

2. By formulating design criteria consisting of a crash pulse duration envelope, design cargo weights, and design considerations.

3. By conducting a dynamic analysis relating the crash pulse criterion to restraint system deflection.

4. By formulating applicable load limiting type energy absorber concepts.

5. By comparing the Army cargo restraint methods as presently employed, the same existing methods applied in a correct manner, and the proposed restraint system properly applied.

The existing restraint methods are found to be insufficient in light of the discrepancy now apparent between static and dynamic crash pulse criteria.

It is determined that a strap device incorporating a load limiting type of energy absorber can be economically adapted to restrain cargo in Army aircraft correctly. The preferred type of energy absorber utilizes the self-storing, wire-bending principle.

CONTENTS

	Page
SUMMARY	iii
LIST OF ILLUSTRATIONS	vi
LIST OF TABLES	xii
INTRODUCTION	1
DEFINITION OF RESTRAINT PROBLEMS	5
DESIGN CRITERIA	19
DYNAMIC ANALYSIS OF RESTRAINT SYSTEM	22
DESIGN CONCEPTS	29
ANALYSIS OF RESTRAINT PROBLEM	46
CONCLUSIONS	79
BIBLIOGRAPHY	81
DISTRIBUTION	84
APPENDIXES	
I. Pull Test Results	86
II. Energy Absorber Configurations	97
III. Validation Tests of Self-Storing, Wire-Bending Energy Absorbers	102
IV. Study of Test Methods	118
V. Itemized Cost Estimate of Self-Storing, Wire-Bending Energy Absorber	171

LIST OF ILLUSTRATIONS

Figure		Page
1	Example of Restraint Methods Defined by Aircraft Manual	5
2	Package with 1,250-Pound Force	6
3	Free Body of Box	6
4	Free Body of Tie-Down Device	6
5	Forces at Left Lower Corner of Tie-Down Device	7
6	Free Body of Package With Applied Forces	7
7	Simulated Aircraft, Cargo, and Restraints	8
8	Free Body of Cargo and Restraint Forces	9
9	Simulated Half-Sine-Wave Pulse Curve	10
10	Maximum Amplitude of Vibration (Restraint Deflection) Versus ωT Compatible to Existing Army Restraint Devices	13
11	Cargo Restrained by Tie-Down Devices of Different Percent Elongations and Rated Strengths	15
12	Cargo Restrained by Different Length Tie-Down Devices Having Identical Properties	16
13	Optimum Restraint	17
14	Pulse-Duration Curve	19
15	Restraint Load-Deflection Curve	22
16	Equilateral Triangular Pulse Curve	23
17	Free Body of Cargo and Restraints	23

18	Existing Army Nylon Tie-Down Devices in Series with Load Limiter Based on Pulse-Duration-Envelope Design Criterion and ϵ = 25% and ℓ = 4'	27
19	Dacron Tie-Down Devices in Series with Load Limiter Based on Pulse-Duration-Envelope Design Criterion and ϵ = 2.5% and ℓ = 6'	28
20	Government-Supplied Plastic Range Counterflexing Load Limiter	29
21	Wire and Plate Combination Showing Points of Flexure Locations.	30
22	Element of Wire Showing Internal Moments	30
23	Geometry of Counterflexing Load Limiter	32
24	Geometry of Wire Threaded Through Hole in Platen	33
25	Geometry of Wire Diameter Versus Hole Size in Platen	34
26	Single Wire Threaded Through Platen	35
27	Two Wires Threaded Through Platen	35
28	Strap Device with Self-Storing, Wire-Bending Energy Absorbers (5,000-Pound Rated Capacity)	37
29	Strap Device With Self-Storing, Wire-Bending Energy Absorbers (10,000-Pound Rated Capacity)	39
30	Strap Device With Tube-Ball Energy Absorbers	41
31	Strap Device With Self-Storing, Wire-Bending Energy Absorbers and Pulleys	42
32	Strap Device With Tube-Ball Energy Absorbers and Pulleys	43
33	One End of Strap Device With Energy Absorbers and Pulleys Fastened to Two Floor Tie-Down Fittings	44
34	Aircraft Floor Space Occupied by Cargo and Tie-Down Devices	48
35	Restraint of 1/4-Ton Utility Truck Using Existing Army Methods (Cargo Weight 2,665 Pounds; Air Transported)	50

36	Restraint of 1/4-Ton Utility Truck Using Existing Army Methods (Cargo Weight, 5,350 Pounds; Air Transported)	51
37	Restraint of Seven 55-Gallon Drums Using Existing Army Methods (Cargo Weight, 2,513 Pounds; Air Transported)	52
38	Restraint of 1/4-Ton Utility Truck With Pallet Using Existing Army Methods (Cargo Weight, 2,665 Pounds; Air Dropped)	53
39	Restraint of Sealdbin "70" Container Using Existing Army Methods (Cargo Weight, 3,500 Pounds; Air Dropped)	54
40	Restraint of 1/4-Ton Utility Truck Using Existing Methods With Correct Restraint (Cargo Weight, 2,665 Pounds; Air Transported)	55
41	Restraint of Six 55-Gallon Drums Using Existing Methods With Correct Restraint (Cargo Weight, 2,154 Pounds; Air Transported)	56
42	Restraint of Thirty-Six 81-mm Cartridge Boxes Using Existing Methods With Correct Restraint (Cargo Weight, 1,800 Pounds; Air Transported)	57
43	Restraint of 1/4-Ton Utility Truck Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,665 Pounds; Air Transported)	58
44	Restraint of Two 1/4-Ton Utility Trucks Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 5,330 Pounds; Air Transported)	59
45	Restraint of Two 1/4-Ton Utility Trucks Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 7,500 Pounds; Air Transported)	60
46	Restraint of 1/4-Ton Utility Truck With Rocket and Trailer Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 4,054 Pounds; Air Transported)	61
47	Restraint of Six 55-Gallon Drums Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,154 Pounds; Air Transported)	62
48	Restraint of Twelve 55-Gallon Drums Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 4,308 Pounds; Air Transported)	63

49 Restraint of Thirty-Six 81-mm Cartridge Boxes Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 1,800 Pounds; Air Transported) 64

50 Restraint of One Hundred and Eight 81-mm Cartridge Boxes Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 5,400 Pounds; Air Transported) 65

51 Restraint of 1/4-Ton Utility Truck With Pallet Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,665 Pounds; Air Dropped) 66

52 Restraint of Sealdbin "70" Container With Pallet Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 3,500 Pounds; Air Dropped) 67

53 Single-Platen Unit; Two Spools, One Above The Other 98

54 Single-Platen Unit; Two Spools, Side by Side 99

55 Two-Spool, Double-Platen Unit 100

56 Canister Storing Unit 101

57 Platen, Guides, Filler, and Wire 111

58 Assembly of Platen, Guides, Filler, and Wires; Plan View 111

59 Assembly of Platen, Guides, Filler, and Wires; Side View 112

60 Assembly of Platen, Guides, Filler, Wires, and Tongue; Plan View 112

61 Assembly of Platen, Guides, Filler, Wires, and Tongue; Side View 113

62 Configuration of Platen, Guides, Filler, Wires, and Tongue Assembly After Pull Tests 113

63 Assembly of Platen, Guides, Filler, Wires, Tongue, and Shields 114

64 Assembly of Platen, Guides, Filler, Wires, Tongue, Shields, and Mylar Polyester Film 114

65 Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity) 115

66 Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (10,000-Pound Rated Capacity) 115

67	Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity) Attached in Test Machine Prior to Pull Test	116
68	Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity) Attached in Test Machine After Pull Test	117
69	Equivalent Square Pulse Curve	119
70	Approximated Characteristics of Restraint System	119
71	Simulated Aircraft, Cargo, and Restraints	119
72	Free Body of Cargo and Restraint Forces	120
73	Vehicle and Cargo Velocity Versus Time	122
74	Equivalent Pulse Envelope Corresponding to Triangular Pulse Curve of -25 g's and 0.25-Second Duration, Which Results in 2 Feet of Restraint Deflection	123
75	Honeycomb Decelerator	126
76	Force-Time Response Curve	127
77	Shock Struts	130
78	Shock Struts Decelerator	131
79	Equivalent Square Acceleration Pulse Limitations for Honeycomb Decelerator	132
80	Spring Decelerator	133
81	Test Vehicle Impacting Stationary Dolly	134
82	Maximum Pulse Time Limitations for Spring Decelerator	136
83	Maximum Stroke Limitations for Spring Decelerator	137
84	Pendulum	139
85	Limitation of Equivalent Square Acceleration Pulse Envelope for Guillotine Test Method	141
86	Guillotine	142
87	Limitations of Equivalent Acceleration Pulse Envelope for Pendulum Test Method	143

88	Piccolo Tubes as Dynamic Water Brake	145
89	Limitations of Equivalent Square Acceleration Pulse Envelope for Piccolo Test Method	146
90	Stainless Steel Decelerator	148
91	Stress Strain Curve for 302 Annealed Stainless Steel Sheet	149
92	Limitations of Equivalent Square Acceleration Pulse Envelope for Stainless Steel Decelerator	151
93	A Typical Acceleration Pulse Curve for Stainless Steel Decelerator	152
94	Friction Brakes	154
95	Friction Coefficient Versus Vehicle Velocity	155
96	Pulse Duration Envelope Limitations for Friction Brakes Neglecting Weight of Brakes	157
97	Pulse Duration Envelope Limitations for Friction Brakes Including Weight of Brakes and Test Vehicle	158
98	Dead Load Assembly Proposal	161
99	AAE Cost Analysis	167

LIST OF TABLES

Table		Page
I	Operational Feasibility	69
II	Payload Comparison	71
III	Cost and Weight of Existing and Proposed Restraint Systems	72
IV	Pull Test Results (0.062-Inch-Diameter Wire, 6061-T6 Aluminum Platen, Hole Edges Broken)	86
V	Pull Test Results (0.072-Inch-Diameter Wire, 2024-T3 Platen Material, Hole Edges Broken)	87
VI	Pull Test Results (0.080-Inch-Diameter Wire, 2024-T3 Aluminum Platen Material, Hole Edge Broken)	88
VII	Pull Test Results (0.092-Inch-Diameter Wire, 2024-T3 Aluminum Platen, Hole Edges Broken)	89
VIII	Pull Test Results (0.105-Inch-Diameter Wire, 2024-T3 Aluminum Platen, Hole Edges Rounded by Polishing)	90
IX	Pull Test Results (0.120-Inch-Diameter Wire, 304 Stainless Steel Platen, Hole Edges Broken)	91
X	Wire Ultimate Tensile Values	91
XI	Test Plan	92
XII	Laboratory Test Results of the Self-Storing, Wire-Bending Energy Absorbers	105
XIII	Prototype Versus Model Parameters	127
XIV	Accuracy of Model Versus Prototype Dynamic Testing	166

INTRODUCTION

This report is an investigation of the restraint problems and solutions for Army cargo in Army aircraft.

An initial study was inaugurated by All American Engineering Company (AAE) in 1963 to determine design concepts predicated on static load criteria as applied to air-transported and air-dropped cargo presently being restrained in Army aircraft. As a result of the existing static load criteria, the design concepts that could be formulated are impractical when the factors of weight, cost, and system effectiveness are considered. An effort in 1964 by the Government, through Aviation Safety Engineering Research (AvSER) of the Flight Safety Foundation (FSF), resulted in a dynamic pulse-duration-envelope criterion for forward restraint of cargo in fixed-wing aircraft. A comprehensive study was then undertaken by AAE to determine the effects of the Government-supplied pulse envelope and the suggested material and design techniques on the design criteria.

The problems that exist in the methods presently used to restrain cargo in military airplanes are:

1. Restraint Methods Defined by Aircraft Manuals: The aircraft manuals give an incorrect impression of the proper way to restrain cargo. It is explained that tie-down devices that lead over the cargo and in the forward and aft directions will provide full restraint in the vertical, aft, and forward directions. An analysis shows that little or no restraint is afforded in the forward and aft directions, with double strength in the vertical direction.

2. Forward Restraint Direction: Under landing and takeoff crashes, the forward restraint direction is pertinent to affording full crew protection.

3. Improper Forward Restraint Resulting from Existing Load Factor Criteria: A dynamic analysis was undertaken which relates the crash pulse criteria and spring rate of the existing tie-down devices to restraint deflection. It was found that an equivalent 39-g static load factor for forward restraint is representative of the dynamic response of the cargo restraint system. Therefore, the imposed load is five times the static load factor utilized as existing Army criterion. For explanation of "equivalent static load", see page 25.

4. Elasticity Problems: Basic fundamental principles are disregarded

when chains and webbing devices are mixed or when variant length devices for restraint in one direction are used. The mixing of different material devices results in an uneven load in the devices because of their elasticity properties. For straps of the same material but of different lengths, the capacity of the smaller length straps will be realized first. The problem is compounded when chains and straps of different lengths are used.

5. Optimum Loading Conditions: Ideal loading is to have a minimum number of restraint devices in relation to the cargo weight. The cargo would have to have built-in tie-down attachments and would have to be sized to permit optimum proportioning of the restraint vectors in the three directions of loading. Most Army cargoes capable of being restrained in Army aircraft do not meet these requirements.

6. Partial Utilization of Cargo Compartment: Investigation of the types of cargo transported by Army aircraft indicates that most cargoes are of relatively high density. Therefore, the cargo weight limit is reached with relatively low cargo compartment volume utilization. The densest cargo results in low cargo volume utilization and in relatively large aircraft floor space coverage.

7. Restraint Devices: The number of devices required to restrain cargo depends upon the shape, size, and weight of the cargo. Cargo shape determines the availability for direct tie-down from cargo to floor fittings. Size of cargo determines the number of available fittings to be utilized with tie-down devices, and restraining of heavy weight cargo requires a multiplicity of restraint devices. Also, the forward restraint direction requires a larger proportion of devices than do the other restraint directions.

8. Tie-Down Fittings: Tie-down fittings available for restraint depend upon the type and size of cargo and upon whether the cargo is air-transported or air-dropped. Pallet-type cargo, roller conveyors, and package-type cargo, by covering up the floor fittings, reduce the number of existing floor tie-down fittings available for restraint.

9. Cargo Tie-Down and Release Time: The time to rig and derig cargo depends on the size and shape of cargo being restrained. Crawling under vehicles to tie to axles, being able to move around cargo to get to floor fittings, passing straps around cargo for restraining purposes, and attaching two devices to make a longer device are problems that become time consuming when an excessive number of devices are required for correct restraint.

Design criteria are presented which include a pulse-duration envelope, design cargo weights, and design considerations. The pulse-duration envelope, which is predicated on fixed-wing crash test data, is defined as an equilateral triangular shaped curve with a maximum peak intensity of 25 g's and a duration of 0.25 second. This criterion is considered pertinent to the longitudinal axis in the forward direction of fixed-wing aircraft. It is assumed that the existing crash static load factors utilized in restraining cargo in Army aircraft is considered to be prevalent for restraint in the other aircraft axes. The section "Design

Cargo Weights" presents plausible assumptions warranted to design a cargo restraint system, while the section "Design Considerations" comprises design objectives formulated from the previous investigation of the problems inherent in existing restraint methods.

A dynamic analysis was undertaken, predicated upon a maximum energy restraint concept and the pulse-duration-envelope criterion, which results in the restraint system deflection. The maximum energy restraint concept is basically a constant load device capable of dissipating energy by constant loading over a 2 foot stroke. An equivalent static load factor corresponding to the dynamic response of the cargo restraint system, for forward restraint, was attained at 15 g's.

Two types of energy absorbers, which act as constant load devices or load limiters, are presented with the purpose of properly restraining cargo in Army cargo airplanes. They are self-storing, wire-bending energy absorbers and tube-ball energy absorbers. The wire-bending energy absorber concept is preferred for compactness, irrespective of stroke requirements. From Government-supplied exploratory data pertinent to counterflexing load limiters, an analytical/experimental program was undertaken which resulted in the design and development of the self-storing, wire-bending concept. See the section "Design Concepts". For cargo restraint, these concepts are formulated into a strap device with energy absorbers, with or without pulleys. The energy absorbers are joined to a Dacron webbing device which is of high strength and low elongation. When utilized, the pulley is placed between the strap device and energy absorbers to provide double-strength capacity.

For fixed-wing aircraft, a design stroke of 2 to 3 feet is required to restrain cargo properly with the use of constant load energy absorbers. However, the Government has requested that the stroke be attenuated to 8 inches for the development of the self-storing, wire-bending energy absorbers to be utilized in rotary-wing aircraft. The testing phase of these devices in prototype form can be found in Appendix I. The report is mainly concerned with the larger design stroke and its application to fixed-wing aircraft.

A comparison is made of the following Army cargo restraint methods: those as presently employed; the same existing methods applied in a correct manner; and the proposed strap device with energy absorbers and pulleys concept correctly restrained. The factors of technical feasibility and operational feasibility are considered and evaluated on a common basis. It is recognized that the proposed design concept can be installed economically as part of the existing restraint system. It was found that the existing Army methods permit insufficient restraint for the considered cargo weights. The existing method with correct restraint was limited in its payload capabilities, while the strap device with energy absorbers and pulleys provides total forward load protection to the aircraft crew even when the aircraft is loaded to its full cargo capacity.

A study of model and full-scale dynamic testing was conducted to conform to the triangular acceleration pulse envelope criterion of 25 g's and a time duration of 0.25 second. See Appendix IV.

For model testing, it was found necessary to determine by dynamic analysis an "equivalent" square pulse curve that will simulate the design pulse criterion. In addition, a dimensionless analysis was undertaken to obtain dynamic similarity for the scaled test.

The model test methods considered were of momentum exchange, while the full-scale test studies considered both momentum exchange and arrestment types. Of all the methods studied, the deformed tube-type shock struts decelerator (see Appendix IV) utilized for model testing appears to be the most adaptable method available for the general requirements.

An itemized cost estimate of testing a cargo restraint system is available; basically, it includes test preparation, test operations, and documentation.

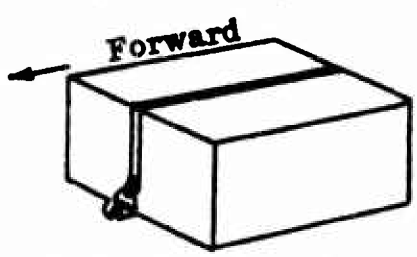
The investigation covered by this report spans the period 1 July 1963 through 8 February 1966.

DEFINITION OF RESTRAINT PROBLEMS

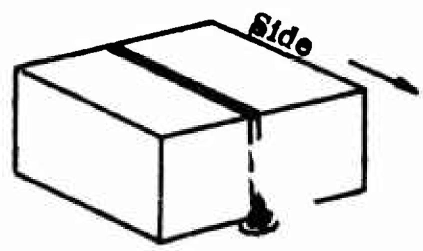
The purpose here is to define the problems that exist in the methods presently adapted to restraining cargo in both rotary- and fixed-wing airplanes.

RESTRAINT METHODS DEFINED BY AIRCRAFT MANUALS

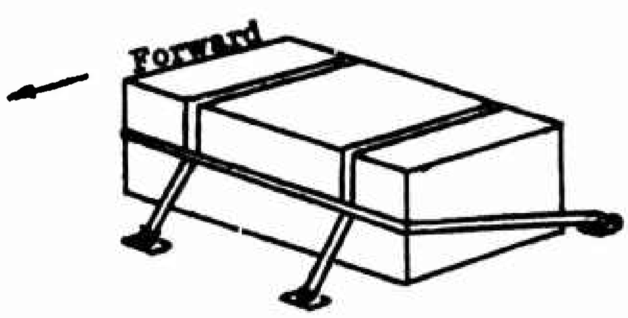
The aircraft technical manuals, as utilized by the pertinent Army personnel, indicate falsely the proper method of restraining cargo. An illustrative example is excerpted from the handbook for the CH-34 (TM-55-1520-202-10, Chapter 4) and is depicted in Figure 1 below.



This A-1A 1,250-pound tie-down device will provide 1,250 pounds of restraint in the forward, vertical and aft directions.



This A-1A 1,250-pound tie-down device will provide 1,250 pounds of restraint in the lateral and vertical directions.



Each A-1A cargo tie-down device will provide 1,250 pounds of forward restraint.

Figure 1. Example of Restraint Methods Defined by Aircraft Manual

Notice that the top two diagrams give full strength credit to a tie-down device in three directions (forward, aft, and vertical for one strap; lateral and vertical for the other strap) when, in fact, the following analysis will show that, with the exception of the presence of the tie-down ring in the way of the box, there is no forward, aft, or lateral restraint. For vertical restraint, there is double strength from each strap. Below is shown a package (solid lines) with a 1,250-pound force in the direction as indicated by the arrow. Assume that the tie-down ring is below the box, as may be realized when the cargo is placed on an air-drop roller conveyor system.

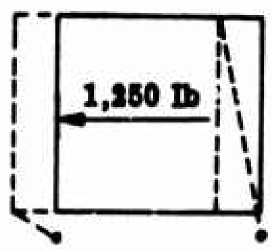
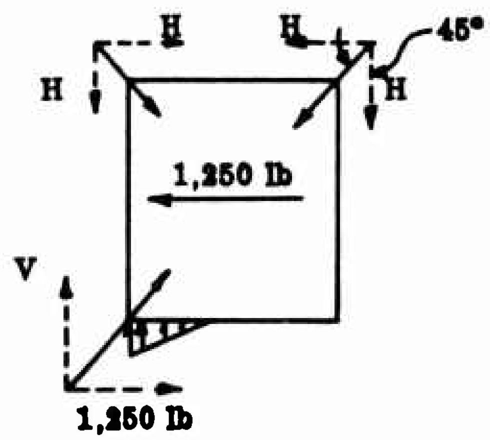


Figure 2. Package With 1,250-Pound Force

When the load is applied to the box, it is obvious that there is very little restraint available, and the box will shift as shown by the dashed lines. For a small amount of shifting, the forces on the package (shown below) are reacted at three corners by the straps and the overturning moment at the bottom of the package. Also shown is a free body of the tie-down device.



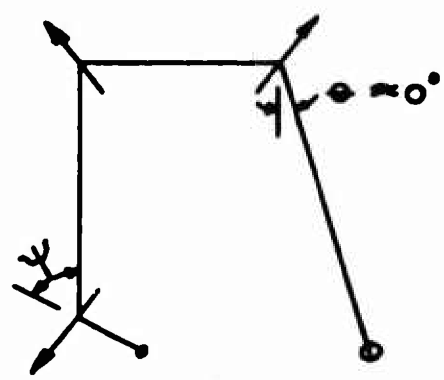


Figure 3. Free Body of Box Figure 4. Free Body of Tie-Down Device

The angles θ and ψ made between the devices and the box are of different magnitude. Angle θ is much smaller than ψ and can be assumed to be practically equal to zero degrees for small shifting of cargo. Therefore, the vector forces at the two upper corners can be assumed to be equal in magnitude and direction. The two horizontal forces from these vectors are opposite in direction and cancel each other. The two vertical forces are of the same magnitude, since the vectors are at 45-degree angles, and the forces do not cancel but act in the same direction. By summing all the horizontal forces on the package, the horizontal vector at the left bottom corner is equal to 1,250 pounds. If a free body of the strap at the lower corner (see Figure 5) is considered, it is easy to see that the strap tension is greater than the restraint force of 1,250 pounds. In order for the forces to try to balance, the package will have to shift a large amount, so that the angle θ will be large enough to change the magnitude of the vector at that corner a sufficient amount. The forces on the package are then

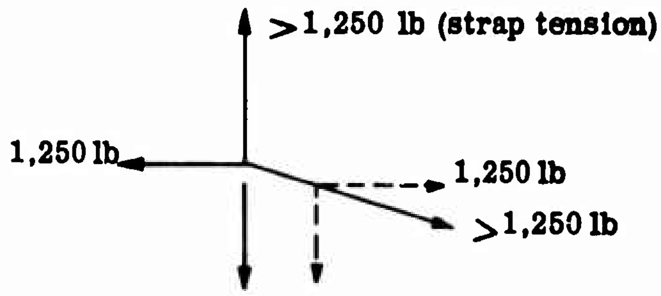


Figure 5. Forces at Left Lower Corner of Tie-Down Device

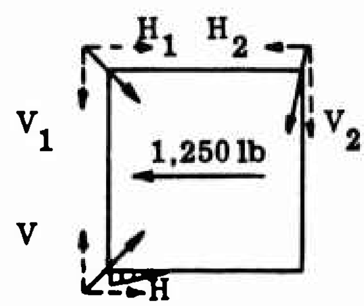


Figure 6. Free Body of Package With Applied Forces

as shown in Figure 6. The problem is then redundant and there are many variables that have to be taken into account, such as the angles the devices make with the box and the package dimensions. In order for the package to shift a great amount, the device would have to be of a material with a larger percentage of elongation, such as a nylon-type webbing. Chain would be ineffective, as it would fail. If a large amount of shifting would occur, then probable crushing of the box would result and, because of the variables involved, there may still be failure of the strap. In the above discussion, friction was neglected, as it is felt to be of a small magnitude. There will still be some measure of restraint, but far below that of the full strength of the device.

The third illustration in Figure 1 is reproduced only to show a repetition of the principle stated in the other two. Here, forward restraint is claimed for two straps which actually lead forward of the cargo and thereby, in actuality, provide no restraint whatsoever. Again, the strap which leads around the package is given only half the credit it is due, since it is in a position to provide twice its strength in restraint.

It is obvious from the above that the manuals give a false impression of the proper way to restrain cargo.

FORWARD RESTRAINT DIRECTION

The forward restraint direction is vulnerable to large acceleration forces as a result of fixed-wing aircraft takeoff and landing crashes. Definitive evidence of the high acceleration forces is the result of air transport crash tests conducted by AvSER.

During a crash event, a pulse is imparted to the aircraft floor while the airplane's speed is decreasing. The cargo, unlike the aircraft, tends to keep its momentum, and consequently the cargo pulse is dependent on the restraint system. It is then evident that the forward restraint direction is of prime concern in affording full crew protection. Other restraint directions are not considered to be significant for crew protection, but restraints are necessary in order to react inflight loads.

IMPROPER FORWARD RESTRAINT RESULTING FROM EXISTING LOAD FACTOR CRITERIA.

A major fault with the existing Army restraint system is associated with the existing forward restraint load factor criterion. This criterion is predicated upon a static g intensity which is not pertinent to restraining cargo in Army aircraft correctly.

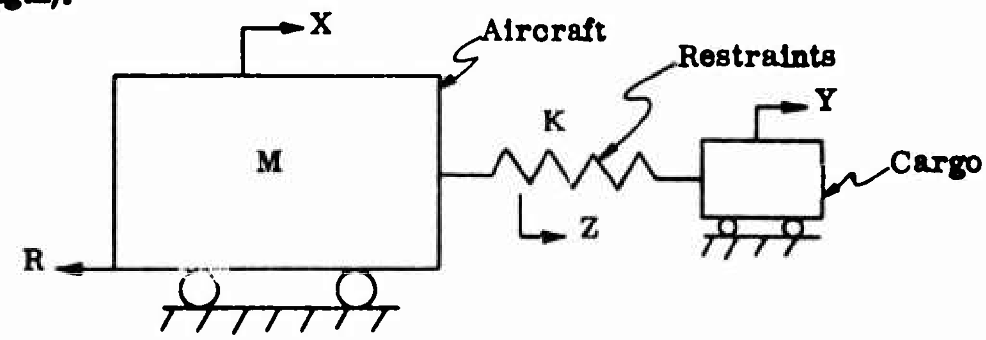
The cargo restraint system has to be capable of reacting dynamic loads which are results of acceleration pulses imparted to the aircraft under landing and takeoff crash conditions. Therefore, it is recognized that restraint deflection plays an important role in defining an affordable restraint system.

The following dynamic analysis will show the inadequacy of the existing Army criteria to restrain cargo correctly in the forward restraint direction. To simplify the analysis, the following assumptions are warranted:

1. The restraint devices react as a linear spring.

2. Friction forces are neglected between cargo and aircraft floor (as exist when utilizing roller conveyors).

3. Restraint pre-tensioning is neglected (in existing Army restraint methods, pre-tensioning loads are small in percentage of restraint breaking strength).



R = simulated crash acceleration pulse
K = spring rate of restraint system
X = aircraft stroke
Z = deflection in restraint
Y = cargo stroke

Figure 7. Simulated Aircraft, Cargo, and Restraints

The aircraft, cargo, and restraints as a dynamic system can best be represented as delineated in Figure 7. If the cargo is isolated from the system (Figure 8) and the forces are summed, then

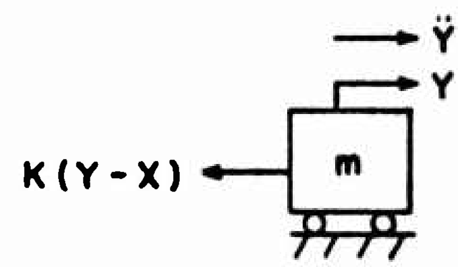


Figure 8. Free Body of Cargo and Restraint Forces

$$K(Y-X) + m\ddot{Y} = 0$$
$$K(Y-X) = -m\ddot{Y}$$

and if $+m\ddot{X}$ and $-m\ddot{X}$ are added, then

$$K(Y-X) = -m\ddot{Y} + m\ddot{X} - m\ddot{X}$$
$$K(Y-X) = -m(\ddot{Y}-\ddot{X}) - m\ddot{X}$$

Let

$Z = Y - X$; $\ddot{Z} = \ddot{Y} - \ddot{X}$

Then

$$KZ = -m\ddot{Z} - m\ddot{X} \tag{1}$$
$$\ddot{Z} + w^2 Z = -\ddot{X}$$

where

$w^2 = \frac{K}{m}$

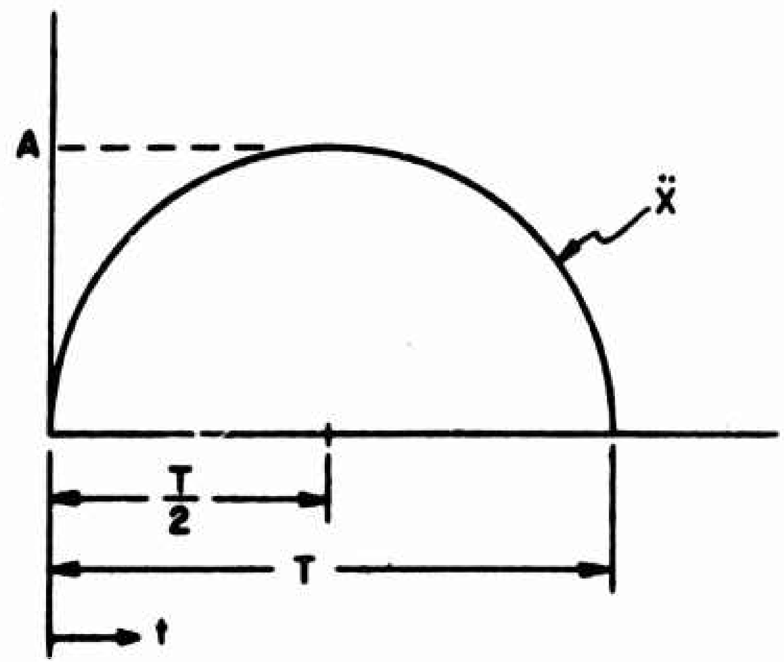
$\ddot{X}$ is a function of t and depends upon the pulse imparted to the aircraft

$\ddot{Y}$ is acceleration of cargo

$\ddot{Z}$ is acceleration of restraint

The solution to equation (1) is $Z = B \cos wt + C \sin wt + Z_{i(t)}$ (2)

Consider a half-sine-wave pulse curve (Figure 9) to replace the equilateral triangular pulse curve as defined under the section "Pulse-Duration Envelope", page 19. This will greatly reduce the complexity of the equations with little sacrifice to the accuracy that would be obtained if an equilateral triangular pulse curve were used.



A is acceleration in g's

T is pulse duration in seconds

t is time in seconds

Figure 9. Simulated Half-Sine-Wave Pulse Curve

The aircraft acceleration equations are

$$\ddot{X} = A \sin \frac{\pi}{T} t \qquad 0 \leq t \leq T \qquad (3)$$

$$\ddot{X} = 0 \qquad t \geq T \qquad (4)$$

From equations (1), (2), (3), (4), and pertinent boundary conditions, the restraint stroke is

$$Z = \frac{-A}{w^2 - \left(\frac{\pi}{T}\right)^2} \left[\sin \frac{\pi}{T} t - \frac{\pi}{wt} \sin wt \right] \tag{5}$$

for $o \leq t \leq T$

$$Z = \frac{A\left(\frac{\pi}{T}\right)}{w^3 - \left(\frac{\pi}{T}\right)^2 w} \left[(1 + \cos wt) \sin wt - \sin wt \cos wt \right] \tag{6}$$

for $t \geq T$

The amplitude of vibration and its time occurence (t_m) for equation (5) is obtained in the following manner: first, differentiate equation (5); second, equate this equation to zero, solving for t_m ; and third, place the values for into equation (5) and obtain the amplitude of vibration. Similarly for equation (6), t_m can be obtained. However, the amplitude of vibration is obtained by separately squaring all the terms in front of the sin wT and cos wT and then by taking the square root. The results are as follows:

$$\text{Amp} = \frac{-A}{w^2 - \left(\frac{\pi}{T}\right)^2} \left[\sin\left(\frac{2\pi^2}{\pi + wT}\right) - \frac{\pi}{wT} \sin\left(\frac{2\pi}{\pi + wT} wT\right) \right] \tag{7}$$

$$t_m = \left(\frac{2\pi}{\pi + wT}\right) T \tag{8}$$

for $o \leq t \leq T$

$$\text{Amp} = \pm \frac{A\left(\frac{\pi}{T}\right)}{w^3 - \left(\frac{\pi}{T}\right)^2 w} \sqrt{2(1 + \cos wT)} \tag{9}$$

$$t_m = \frac{1}{w} \text{arc tan} - \frac{(1 + \cos wT)}{\sin wT} \tag{10}$$

for $t \geq T$

Figure 10, utilizing equations (7) and (9), depicts the maximum amplitude of vibration (restraint deflection) for a specified wT. That portion of the curve to the right of wT equal to π is the restraint deflection for responses felt during the time the pulse is imparted to the aircraft. To the left of $wT = \pi$, the curve represents the restraint deflection for responses felt after the pulse has been imparted to the aircraft.

The existing Army tie-down devices are constructed of nylon material. The shape of its stress-strain relationship takes the form of a cubic equation, or as a nonlinear spring. An equivalent linear relationship will be utilized in equations (7) and (9) to make a restraint comparison of existing Army criteria and the pulse-duration-envelope criteria. From All American's experience with a 10-foot length of nylon-type tie-down device, an equivalent linear spring rate of 5,000 pounds per foot is considered to be conservative. This means that the cargo and its restraints can ultimately deflect 1 foot without failure if the aircraft tie-down fittings are limited to 5,000-pound capacity.

It is appropriate at this time to determine the equivalent static g load factor for forward restraint that is corresponding to the dynamic response of the cargo restraint system. From Figure 10, the following relationships are realized:

$$Amp = C_1 \frac{A}{w^2} \tag{11}$$

$$wT = C\pi \tag{12}$$

where

C_1 is depicted from the ordinate

C is depicted from the absolssa

with $w^2 = \frac{K}{m}$

Thus,

$$K = \frac{C_1 mA}{Amp} \tag{13}$$

$$K = \frac{C^2 \pi^2 m}{T^2} \tag{14}$$

Equating the two equations,

$$C_1 = \frac{\pi^2}{T^2} \frac{Amp}{A} C^2 \tag{15}$$

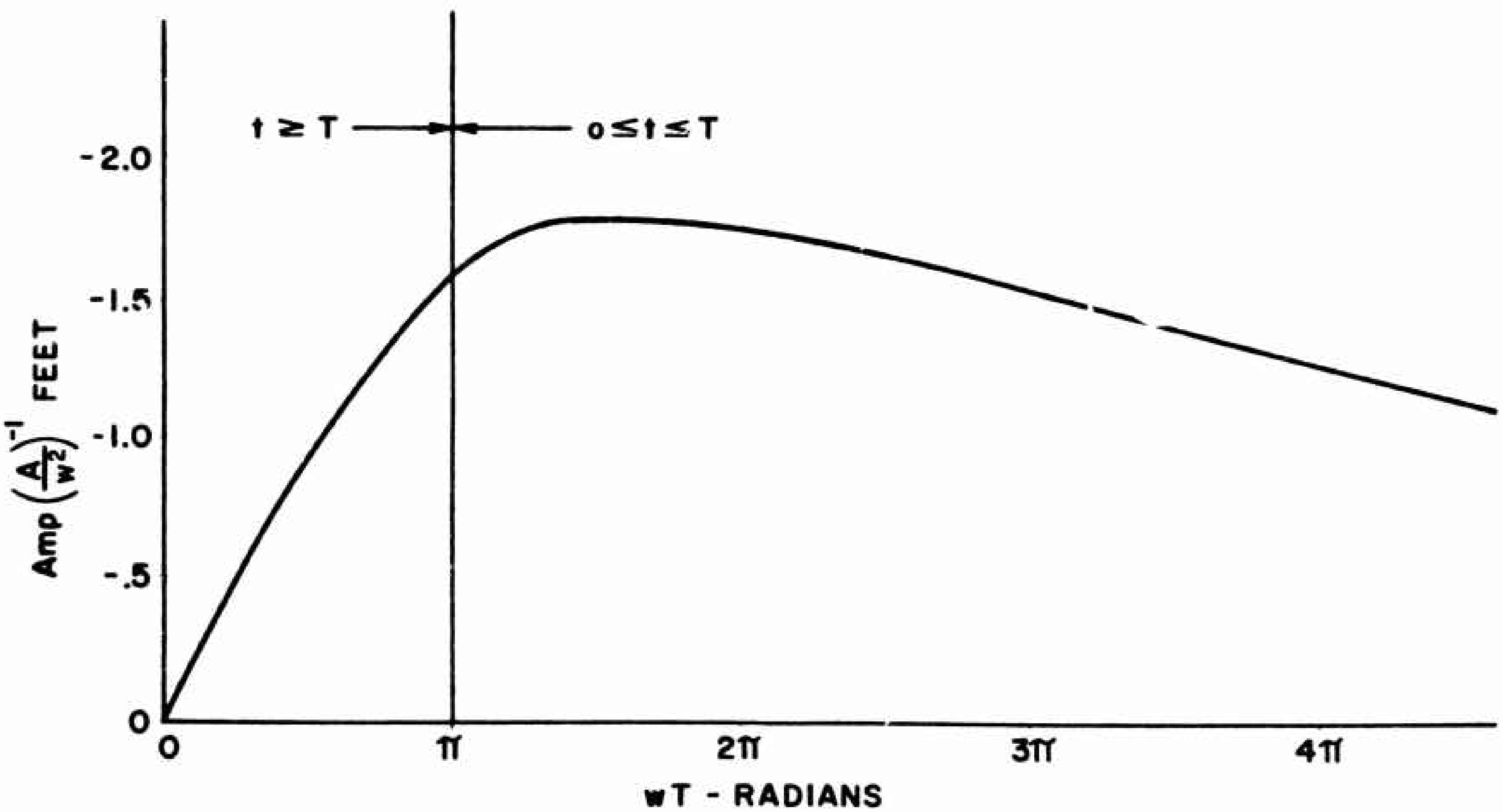


Figure 10. Maximum Amplitude of Vibration (Restraint Deflection) Versus wT Compatible to Existing Army Restraint Devices

with

A = 25 g's = -805 feet per second^2 (maximum pulse intensity)

T = 0.25 second (pulse duration)

Amp = 1 foot (ultimate deflection of existing restraint system)

Then

$$C_1 = 0.192C^2 \tag{16}$$

A trial and error solution results in $C = 2.86$ and $C_1 = 1.56$, which can be verified from Figure 10. From equation (13) and $m = W/g$, where W is equal to cargo weight, then

$$K = 39W \tag{17}$$

The equivalent static load factor is 39 g's.

The existing Army restraint criterion for forward restraint is an 8-g static load factor. Therefore, it is apparent that the existing Army restraint criterion is being erroneously utilized. As an example, consider a piece of cargo weighing 2,000 pounds and being capable of restraint in the aircraft longitudinal axis in the forward direction. Predicated on the 8-g load factor, 5,000-pound-capacity devices, and the assumption that all restraint devices used are of equal length, four tie-down devices are considered to be sufficient restraint according to Army procedures. Utilizing the equivalent 39-g static load factor, sixteen tie-down devices are required for correct restraint. Four times as many forward restraints as utilized with existing methods are required to afford full crew protection during a maximum survivable crash condition.

The foregoing analysis demonstrates the need of a restraint system capable of reacting the crash pulse responses with a minimum of restraint devices. When a large number of restraints is required, the elasticity problem (see page 16) of variant length devices becomes apparent and must be included in computing the required number of devices for a given weight. This could result in a multiple of restraint devices and is dependent on the location of the aircraft floor tie-down fittings.

ELASTICITY PROBLEMS

One of the problems that exists with tying down cargo is caused by the mixing of web and chain devices. Assume that a piece of cargo is restrained to the aircraft by an MC-1 nylon web with a rated strength of 5,000 pounds and by an MB-1 chain with a rated strength of 10,000 pounds. The load being applied to the package is 15,000 pounds (see Figure 11). The percent elongations of the MC-1 and MB-1 devices are assumed to be 50 percent and 3 percent respectively. It is felt that 50 percent is lower than might be expected, but for illustration purposes it will

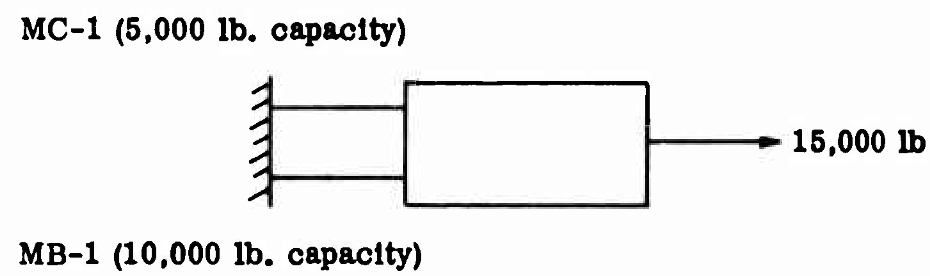


Figure 11. Cargo Restrained by Tie-Down Devices of Different Percent Elongations and Rated Strengths

be sufficient. Assume that the lengths of these devices are the same; therefore, their unit strains are also equal. The loads felt by the devices are directly proportional to their percent elongation or

$$P \sim \epsilon$$

Then

$$P_E = \frac{\epsilon_c}{\epsilon_w} P_w \tag{18}$$

where

P_E is equivalent load in web device

P_w is capacity of web device

ϵ_w is unit strain for web device

ϵ_c is unit strain for chain device

When the steel chain has reached its maximum elongation, the device is loaded to 10,000 pounds. The equivalent load in the nylon webbing from equation (18) is

$$P_E = \frac{.03}{.50} \times 5000 = 300 \text{ pounds}$$

The total load reacted by the devices is 10,300 pounds. The aircraft operating manuals make it seem apparent, from the above figure, that the 15,000 pounds is the sum of the devices. In reality, the system is overloaded, and a load in excess of 10,300 pounds will probably overload the MB-1 device, causing failure and dumping all the load into the MC-1 device.

The above example applies to devices that have a large variation in their elongation properties. For those restraints that have a small variation, the elasticity of the package will probably realign the devices to enable the applied load to be reacted.

On the other hand, if the two devices are of equal elasticity and length and are attached to a 5,000- and a 10,000-pound rated fitting, the load will obviously divide into 7,500 pounds in each, causing failure in the 5,000-pound fitting, and then overloading the higher rated one.

Consider a package with devices identical in all respects except for different lengths (see Figure 12). The deflection equation is shown as follows:

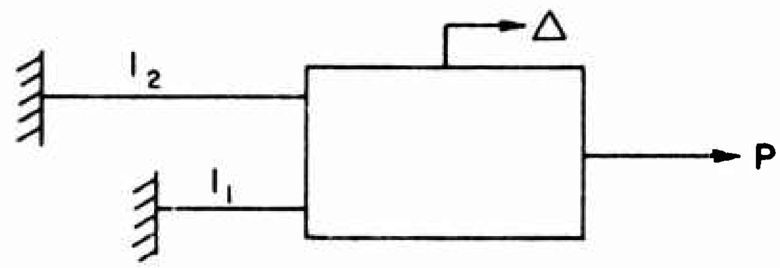


Figure 12. Cargo Restrained by Different Length Tie-Down Devices Having Identical Properties

$$\Delta = \frac{Pl}{AE} \tag{19}$$

where
- l is length of restraint
- A is cross-sectional area of restraint
- E is modulus of elasticity of restraint

With $Pl = AE$,

let $c = AE$;

then $Pl = c$ where P and l are the only variables and

$AE\Delta$ are the same for each device.

It can then be shown that

$P_1 l_1 = P_2 l_2$ where P_1 is load in shorter device

P_2 is load in longer device

$$P_1 = \frac{l_2}{l_1} P_2 \tag{20}$$

It is obvious from the above equation that the shorter device will reach its maximum capacity first. As l_2 becomes excessively longer than l_1, the rating of the longer device becomes negligible.

In general, care must be taken not to use devices for restraint in one direction where one is excessively longer than the other. It is to be noted that the package elasticity will have some effect in realigning the devices, and, therefore, those devices that are slightly longer than others will have essentially the same loads.

OPTIMUM LOADING CONDITIONS

Ideal loading is best accomplished by utilizing a minimum number of restraint devices in relation to the cargo weight. An ideal case would be simply to tie four restraints to a piece of cargo with the capability of proper restraint in the required directions (see Figure 13). For practical application, the cargo would have to have tie-down attachments and to be sized so that the devices may be installed

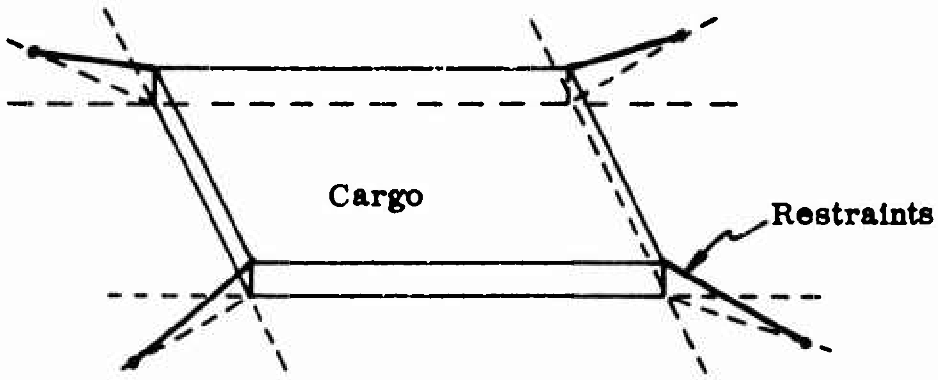


Figure 13. Optimum Restraint

at angles which permit optimum proportioning of the restraint vectors in the three directions of loading. The preponderance of Army cargo capable of being restrained in Army aircraft does not meet these requirements.

PARTIAL UTILIZATION OF CARGO COMPARTMENT

Partial utilization of a cargo compartment is a result of package density. Dense cargo, such as cartridge boxes, use up the restrainable weight-lifting capacity of the aircraft before maximum cargo compartment volume is reached. Investigation of the types of cargo transported by Army aircraft indicates that most cargo is of relatively high density (see Figures 35 through 52). Therefore, the cargo weight limit is reached with relatively low cargo compartment volume utilization. The densest cargo results in low cargo volume utilization and relatively large aircraft floor space coverage. If the crash pulse-duration envelope (reference page 19) is rigorously applied, the existing restraint techniques limit the cargo payload.

RESTRAINT DEVICES

The number of devices required to restrain cargo depends upon the shape, size, and weight of the cargo. To restrain heavyweight cargo requires a multiple of restraint devices. The size of cargo determines the number of available fittings to be utilized with tie-down devices, while the shape of the cargo determines the availability for direct tie-down from cargo to floor fittings. For a rectangularshaped cargo, tie-down devices are passed around the cargo for restraint, where for a vehicle, the axles are used as points of tie-down locations. The forward restraint direction requires a larger proportion of devices than the other restraint directions. Mixing of strap- and chain-type devices and using different length devices for restraining cargo result in elasticity problems, which are discussed under the section "Elasticity Problems", page 14. In addition, the chain devices are considered to be unfit as a restraint method because of their relatively high stiffness characteristics. The devices may become responsive to high oscillations that occur during the crash, whereas flexible devices are relatively less responsive.

TIE-DOWN FITTINGS

The number of floor tie-down fittings available for restraint depends on the type and size of cargo and whether the cargo is to be air transported or air dropped. A package-type cargo reduces the number of existing floor tie-down fittings available for restraint by covering up floor fittings under the package. A vehicle, on the other hand, will not obscure as many tie-down fittings, but it is difficult to make use of these fittings. For heavy packages, roller conveyors are used to facilitate the loading and unloading of cargo. These are examples of air transported cargo. For air delivery, pallets and roller conveyors are used; this results in an additional two rows of aircraft floor fittings being obscured. An insufficient number of tie-down fittings is then available, so that the weight of cargo to be restrained is therefore limited to the desired load factors.

CARGO TIE-DOWN AND RELEASE TIME

It is important to consider the time required to rig and derig cargo for air transport and air drop operations. The time considered depends on the size and shape of cargo being restrained. For cargo such as vehicles, the problems include crawling under vehicles to tie restraining devices to axles; moving around cargo to get to floor tie-down fittings; passing straps around cargo for restraining purposes; and restraining palletized cargo by tying from the cargo to the floor fittings. For cargo such as cartridge boxes, the problems include utilizing devices for each layer and column of boxes for complete restraint and attaching two devices together to make a longer device to restrain the cargo.

DESIGN CRITERIA

The criteria applied to the design of a cargo restraint system are discussed in this section.

PULSE-DURATION ENVELOPE

The pulse-duration envelope is defined as an equilateral triangular-shaped curve with maximum peak intensity of 25 g's and duration of 0.25 second (see Figure 14 and Bibliography reference 1a). The pulse intensity is considered to be pertinent to takeoff and landing crashes and is related to the longitudinal axis in the forward direction of fixed-wing aircraft.

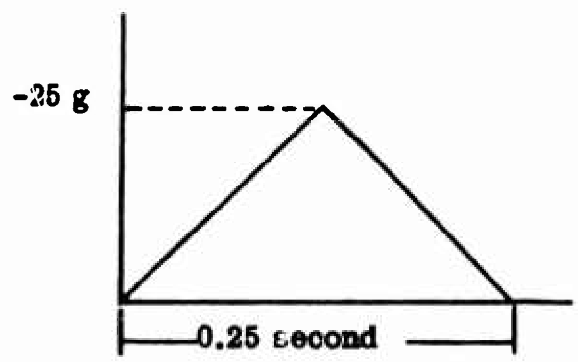


Figure 14. Pulse-Duration Curve

The existing crash static load factors that are utilized in restraining cargo in Army aircraft will be considered to be adequate for restraint in the other aircraft axes. They are as follows:

Restraint Direction	Fixed-Wing
Aft	1.5 g's
Down	Based on floor structure
Up	2.25 g's
Side	1.5 g's

The report "Cargo Restraint Concepts For Crash Resistance" (Bibliography reference 1a) indicates that a pulse envelope for rotary-wing aircraft is less severe than for fixed-wing airplanes. It is then assumed that the above criterion is more than adequate for rotary-wing aircraft, and a greately reduced criterion would be appropriate.

DESIGN CARGO WEIGHTS

The following assumptions are warranted in order to design a cargo restraint system:

1. The cargo C.G. is to be in a centered, vertical, longitudinal plane of the cargo compartment. The vertical C.G. limits are to be 5 inches above the floor to 6 inches above the cargo compartment C.G.

2. All Army cargo is capable of reacting its own g loadings.

3. Factors of safety for designing any proposed concept are included in the crash pulse criteria.

4. Maximum payload carried in the cargo compartments is considered to be maximum payload at zero fuel minus minimum usable fuel.

5. Maximum cargo displacement of 2 to 3 feet is feasible, predicated on the pulse-duration-envelope criteria.

DESIGN CONSIDERATIONS

The following design objectives are:

1. Elasticity Problems: Avoid differential elongation between chains and straps, and refrain from using variant length devices.

2. Proper Restraint: Achieve total restraint without attendant problems.

3. Optimum Loading Conditions: Completely restrain maximum cargo weights.

4. Forward Restraint Direction: React the forward load with a minimum of restraint devices.

5. Utilization of Cargo Compartment: Permit additional floor space and cargo weight utilization.

6. Restraint Devices: Permit a reduction of the number and types of devices for cargo restraint.

7. Tie-Down Fittings: Minimize the number of fittings used in the installation, and do not modify the airframe in any case.

8. Cargo Tie-Down and Release Time: Provide rapid and simple installation along with a quick method of derigging cargo for air transport and air drop operations.

9. Construction: Provide a compact system with rugged construction, capable

of reacting cargo loads based on the pulse-duration-envelope criterion. Minimize use of exotic material.

10. Component Parts: Design components to have a minimum number of parts to reduce the number of fittings to be carried as aircraft equipment.

11. Design Concept Weight: Consider weight to be of primary importance and design component weights to be kept to a minimum.

12. C.G. Requirements: Consider C.G. requirements in design of cargo restraint system.

13. Cargo Handling Environment: Design for sturdiness to withstand abuse inherent in cargo handling environment.

14. Universal Use: Design to be universally applicable to all Army aircraft.

15. General Specification: Comply with general specification for design of "Aeronautical Support Equipment".

DYNAMIC ANALYSIS OF RESTRAINT SYSTEM

MAXIMUM ENERGY RESTRAINT CONCEPT

The maximum energy restraint concept is predicated upon utilizing the maximum possible area under the restraint load-deflection curve (Figure 15). A constant load curve would best fulfill this requirement. However, it is necessary that a webbing tie-down device be used in conjunction with a constant load device to make up a satisfactory restraint tie-down device. Therefore, based on the webbing characteristics, an onset rate will be realized during operation. The web device should be of high strength and low elongation, optimizing the energy under the load-deflection curve. The buildup in energy in the webbing portion of the restraint device is additive to the energy absorbed by the constant load device, resulting in total energy required to restrain cargo. This is apparent since the webbing device will first elongate to the restraint capacity, followed by the reaction of the constant load device. The total cargo displacement will be the sum of the webbing deflection and the load device stroke.

A dynamic analysis, pertinent to the above discussion, relating restraint stroke to its corresponding parameter is to follow. It is assumed that restraint pretensioning and friction between cargo and floor are negligible.

The restraint load-deflection curve and aircraft crash pulse curve (reference Figure 14) are depicted in Figures 15 and 16, respectively.

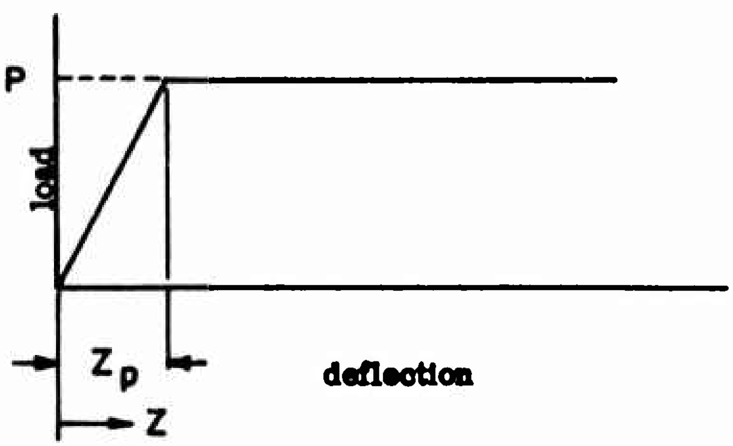


Figure 15. Restraint Load-Deflection Curve

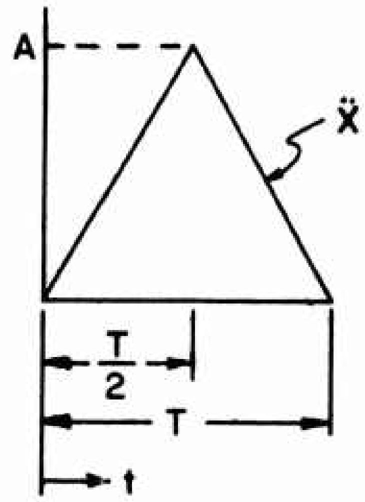


Figure 16 Equilateral Triangular Pulse Curve

Equation (1) is applicable and will be rewritten as equation (21) for convenience.

$$\ddot{Z} + w^2 Z = -\ddot{X} \tag{21}$$

for $0 \leq t \leq T_p$

For the equations for $t > T_p$, consider the free body of cargo depicted in Figure 17. This free body is similar to the one shown in Figure 8 except that P is replacing

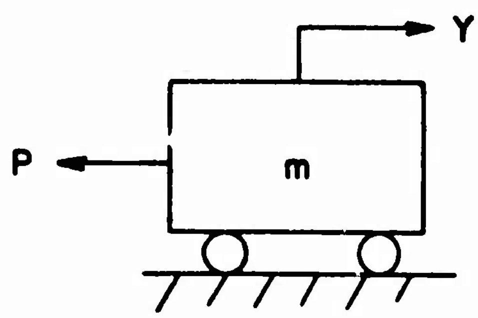


Figure 17. Free Body of Cargo and Restraints

K (Y-X). Let $Q^2 = \frac{P}{m}$ and replace w^2 in equation (1), thus,

$$\ddot{Z} + Q^2 = -\ddot{X}$$

for $t \geq T_p$ (22)

The aircraft acceleration equations are

$$\ddot{X} = \frac{2A}{T}t \tag{23}$$

for $0 \leq t \leq \frac{T}{2}$

$$\ddot{X} = 2A - \frac{2A}{T}t \tag{24}$$

for $\frac{T}{2} \leq t \leq T$

$$\ddot{X} = 0 \tag{25}$$

for $t \geq T$

Placing equation (23) into (21) with the pertinent boundary conditions and solving the resulting differential equation, the restraint stroke is

$$Z = \frac{2A}{w^3 T} \sin wt - \frac{2At}{w^2 T} \tag{26}$$

for $0 \leq t \leq T_p$

It is assumed that Z_p (see figure 15) occurs between $0 \leq t \leq \frac{T}{2}$ and at time $t = T_p$. By placing $Z = Z_p = \epsilon \ell$ and $t = T_p$ into (26), then

$$\sin wT_p - wT_p = \frac{\epsilon \ell w^3 T}{2A} \tag{27}$$

This equation is used to verify that $T_p \leq \frac{T}{2}$ and is the basis upon which the final pertinent equation is based. It so happens that the forward maximum restraint deflection always occurs after the pulse imparted to the aircraft has ceased. Therefore, only this equation will be shown; it is obtained by solving the differential equation (22) with the pertinent boundary conditions and the aircraft acceleration equations (23) through (25). The resulting equation is

$$Z = \frac{-Q^2 t^2}{2} + \left(C_1 - \frac{AT}{2}\right)t + \frac{AT^2}{4} + C_2 \tag{28}$$

for $t \geq T$

where

$$C_1 = \frac{2A}{w^2T}\left(\cos wT_p - 1\right) + Q^2 T_p + \frac{AT_p^2}{T} \tag{29}$$

$$C_2 = \frac{2A}{w^2T}\left(\frac{\sin wT_p}{w} - T_p \cos wT_p\right) - \frac{Q^2 T_p^2}{2} - \frac{2AT_p^3}{3T} \tag{30}$$

The amplitude of vibration or maximum restraint stroke and its time occurrence (t_m) is obtained, first, by differentiating equation (28) and equating to zero; second, by solving for t_m; and third, by placing t_m into (28). Then

$$\text{Amp} = \frac{1}{2Q^2}\left(C_1 - \frac{AT}{2}\right)^2 + \frac{AT^2}{4} + C_2 \tag{31}$$

$$t_m = \frac{1}{Q^2}\left(C_1 - \frac{AT}{2}\right) \tag{32}$$

for $t \geq T$

The above analysis is applicable only when an elastic tie-down device is utilized in series with the constant load device. Eliminating the web device would result in a relatively stiff restraint system. Such a rigid system is responsive to high-frequency oscillations that occur during the interval when the acceleration is imparted to the aircraft. These high-frequency oscillations are not included in the dynamic analysis because of their unpredictable nature. However, flexible systems are relatively less responsive to these oscillations. The above equations and their applications to existing Army tie-down devices and a high-strength, low-elongation web device are discussed under the following heading, "Equivalent Static Load Factor". A constant load device, as described above, is called a load limiter and is discussed under the heading "Design Concepts".

EQUIVALENT STATIC LOAD FACTOR

For forward restraint, an equivalent static load factor, corresponding to the dynamic response of the cargo restraint system, is attainable. The equivalent factor will greatly simplify the calculation required to ensure correct restraint.

The equivalent load factor will depend on the characteristics of the web portion of the restraint system. The existing Army tie-down devices are of nylon material with a 25-percent elongation at 5,000-pound capacity. This type of restraint combined with a load limiter unduly loses the effectiveness of the maximum energy concept. During crash conditions, the nylon web device will absorb little energy with a large stroke. The stroke is dependent upon the length of web device used.

It is felt that 6 feet of strap length is used most of the time in restraining cargo. This, then, results in 1-1/2 feet of stroke during operation prior to reaction of the load limiter. Consequently, the load limiter, which is capable of absorbing more energy per foot of stroke than the web device, has to reduce its design capabilities because of the total restraint stroke design criterion of 2 to 3 feet (see page 20). This results in excessive restraints in tying cargo securely. The optimum restraint system can be utilized by considering a Dacron material to replace the nylon. The Dacron tie-down devices have approximately a 2.5-percent elongation at 5,000-pound capacity under dynamic conditions. During operation, the Dacron webbing will deflect a small percentage of the total allowable restraint stroke.

The curves of Figures 18 and 19 have been delineated from equations (27), (29), (30), and (31) to show the restraint stroke versus equivalent static load factor for both nylon webbing and Dacron webbing in series with a load limiter. A 4-foot nylon webbing length was used in calculating the curve to obtain the best possible combination of webbing and load limiter devices. A 6-foot length was used for the Dacron device. From Figure 19, the equivalent static load factor for about a 2.2-foot stroke is approximately 15 g's, using Dacron straps. A 2.2-foot stroke is predicated on a 2-foot stroke load limiter and on approximately 0.2 foot of Dacron strap elongation. For comparison purposes, from Figure 18, a 24-g equivalent static load factor is obtained for a 2.2-foot restraint stroke. To obtain the number of forward restraint devices required for a given cargo weight, the load factor is multiplied by the cargo weight, and the result is divided by the rated capacity of the tie-down device. As an example, reverting to the previous cited example (page 14), if a 2,000-pound package is restrained in the forward aircraft direction with 5,000-pound-capacity devices, then 10 nylon and load limiter combination devices and 6 Dacron and load limiter combination devices are required. It is to be remembered that these equivalent static load factors apply only for a pulse-duration envelope of 25-g intensity and 0.25-second duration.

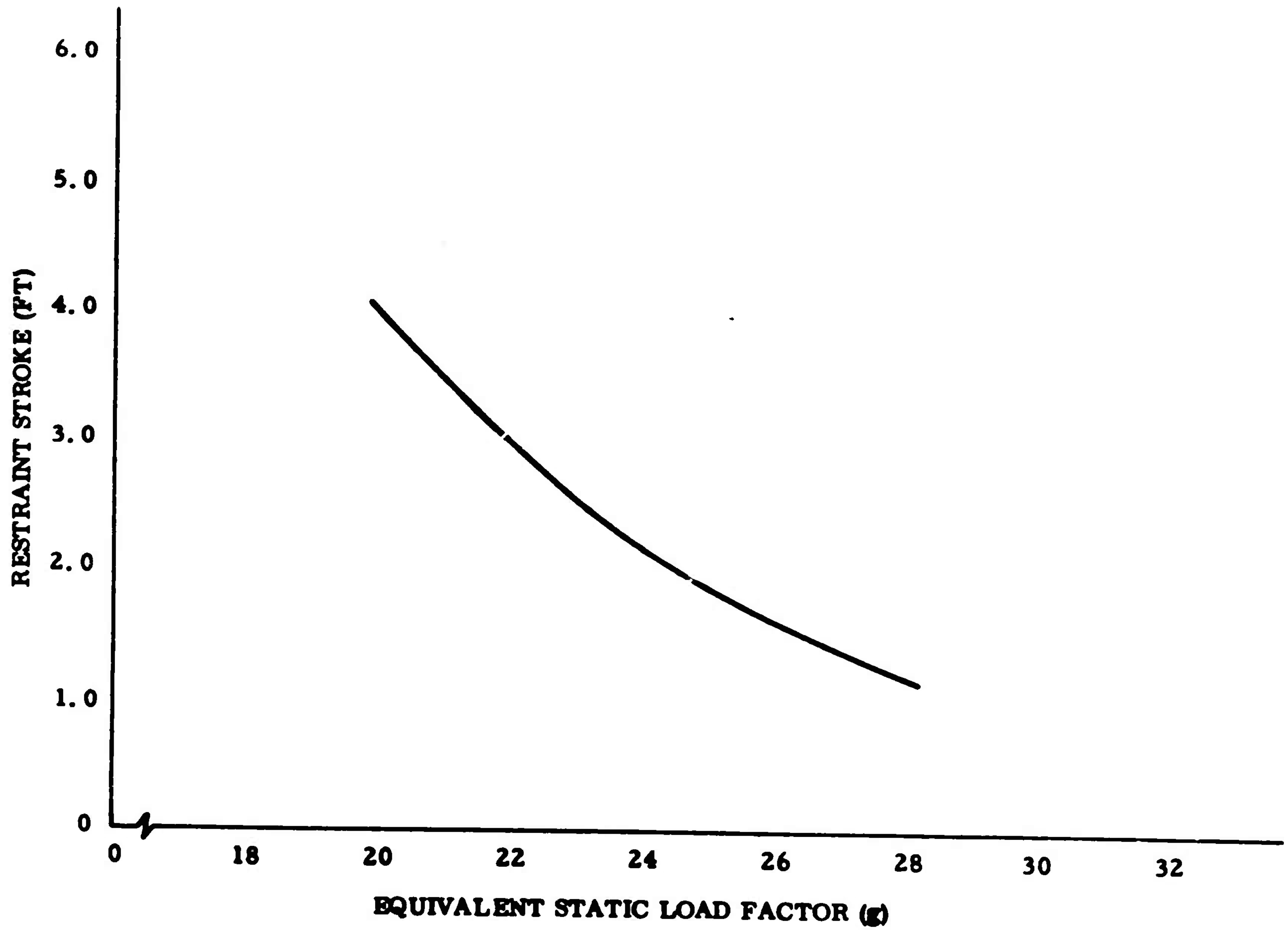


Figure 18. Existing Army Nylon Tie-Down Devices in Series with Load Limiter Based on Pulse-Duration-Envelope Design Criterion and $\epsilon = 25\%$ and $\ell = 4'$

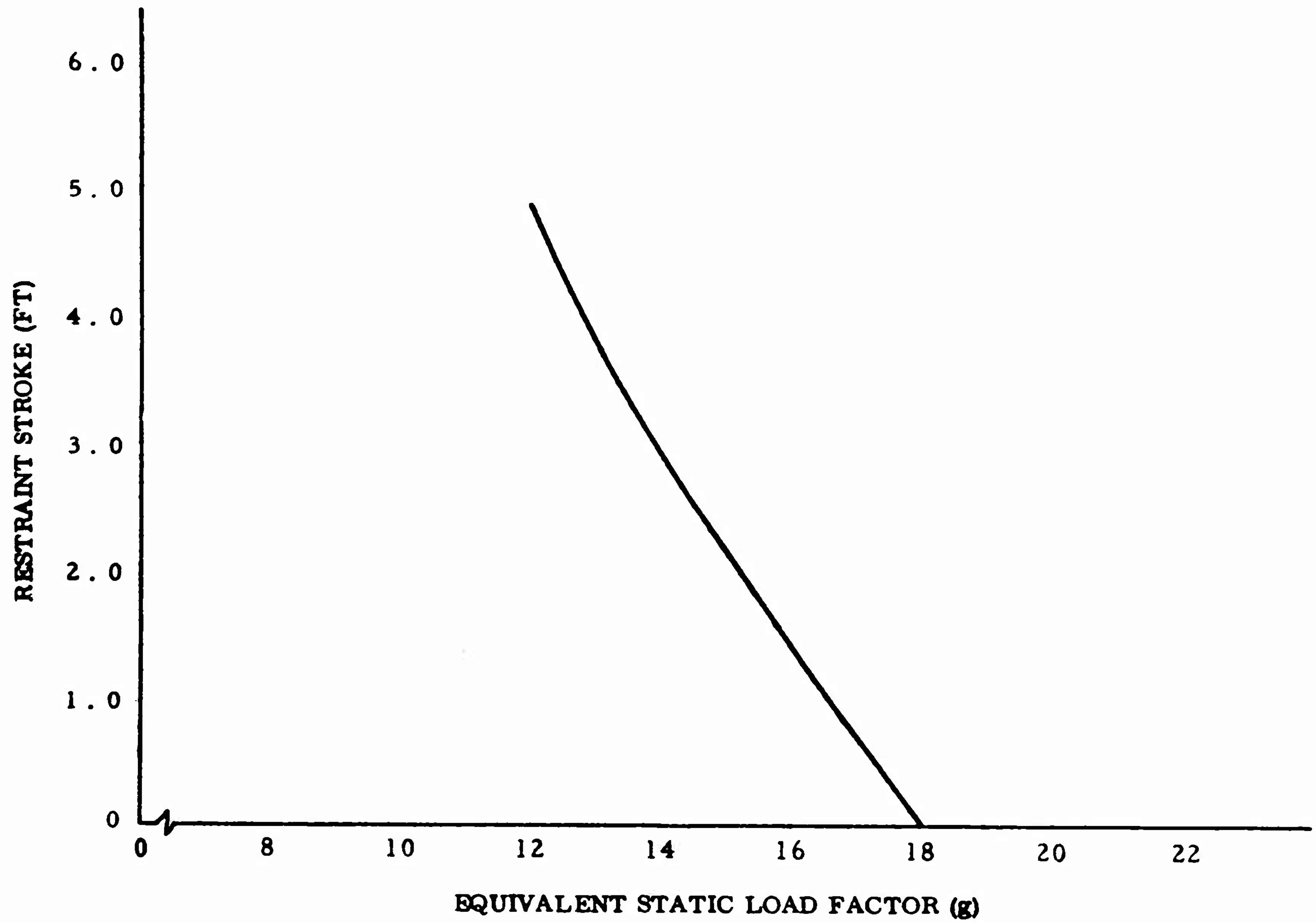


Figure 19. Dacron Tie-Down Devices in Series with Load Limiter Based on Pulse-Duration-Envelope Design Criterion and $\epsilon = 2.5\%$ and $\ell = 6'$

DESIGN CONCEPTS

The design concepts to be discussed are basically web-type tie-down devices attached to load limiters. The web devices, as previously indicated, are of Dacron material with high strength and long elongation properties, and the load limiters are energy absorbers designed to react to a predetermined stroke. Two basic types of load limiters are to be considered. They are the counterflexing and the tube-ball types.

The counterflexing load limiter is a concept preferred by the Government and was initiated from Government-supplied data, which then resulted in an analytical/experimental program. This load limiter concept consists of stainless steel wires woven through a platen with a given thickness, number of holes, hole diameter, and hole spacing.

REVIEW OF GOVERNMENT-SUPPLIED DATA

This data, which was supplied by the Government for evaluation and review on the plastic range counterflexing-type load limiter, was obtained by using an aluminum plate with three oblong 1/4-inch-diameter holes drilled on 3/4-inch centers. Passing through the three holes were four 0.091-inch-diameter stainless steel (annealed) wires woven in and out of the holes as shown.

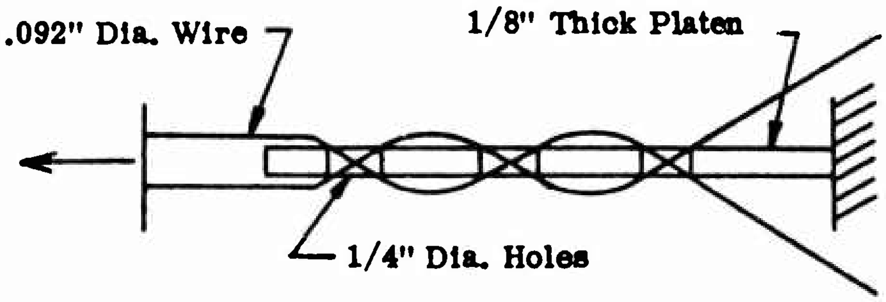


Figure 20. Government-Supplied Plastic Range Counterflexing Load Limiter

The counterflexing load limiter can be easily defined as the means of pulling wire through a stationary platen at a constant load. The load developed basically depends on hole size, hole spacing, and plate thickness. The radius of bend of the wire after weaving is dependent on the above variables. The greater the bend, the greater the pull load that can be accomplished.

Included in the supplied data is an equation of pull load as a function of coefficient of friction, wire-bending moment, and wire curvature of radius. The following delineates the development of this equation:

It is assumed that plastic flexure takes place at points 1 through 6. See Figure 21.

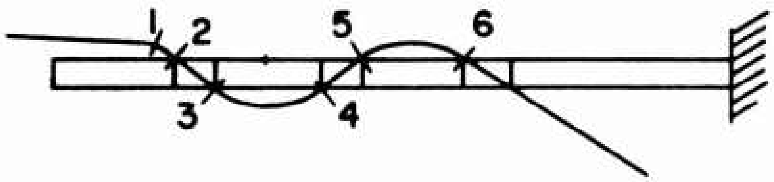


Figure 21. Wire and Plate Combination Showing Points of Flexure Locations

When the wire is pulled from a straight portion to a curved portion, energy is expended in the form of plastic deformation. This is evident from Figure 21 when the wire passes points 2, 4, and 6. The work done is $M^* d\theta$.

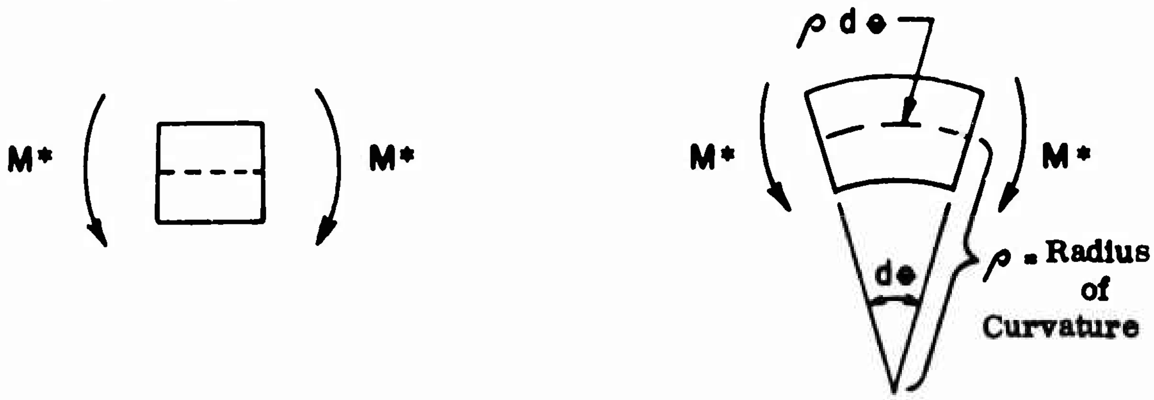


Figure 22. Element of Wire Showing Internal Moments

Therefore, work per unit length, $\bar{\mu} = \dfrac{M^* d\theta}{\rho\, d\theta} = \dfrac{M^*}{\rho}$ (33)

where

M^* is plastic moment
ρ is radius of curvature of wire

Similarly, work done per unit length, in going from a curve to a straight section, when the wire passes points 1, 3, and 5 (Figure 21), is $\frac{M^*}{\rho}$. The external work displacing a wire a unit distance is P_x unity. The internal work or plastic work per unit length, $\overline{W}$, referring to Figure 21, is

$$\overline{W} = \frac{2M^*}{\rho_1} + \frac{4M^*}{\rho_2} \quad (34)$$

The factor 2, corresponding to ρ_1 in the equation, is obtained from points 1 and 2 (Figure 21), and the factor 4, corresponding to ρ_2, is obtained from the remaining points.

If external work is equated to internal work, the following equation is obtained:

$$P \sim M^* \left[\frac{2}{\rho_1} + \frac{4}{\rho_2} \right] \tag{35}$$

If a coefficient of friction factor of 1.5 is considered, equation (35) becomes

$$P = 1.5\, M^* \left[\frac{2}{\rho_1} + \frac{4}{\rho_2} \right] \tag{36}$$

Equation (36) represents the extent of the Government-supplied data. A further investigation of this equation reveals the following:

1. If all holes and hole spacings are considered to be the same for a given platen, ρ_1 is approximately $1/2\,\rho_2$. Equation (36) becomes

$$P = \frac{12\, M^*}{\rho_2}$$

or

$$P = \frac{6\, M^*}{\rho_1} \tag{37}$$

Figure 4 supports this observation.

2. A general equation is written considering an infinite number of holes with the same diameter and the same hole spacings. The equation is

$$P = 1.5\, M^* \left[\frac{2}{\rho_1} + \frac{2(n-1)}{\rho_2} \right] \tag{38}$$

where

n is the number of holes in the platen,

Or with $\rho_1 = \frac{1}{2}\rho_2$ from Equation (38)

then

$$P = 3.0\,(n+1) \frac{M^*}{\rho_2}$$

$$P = 1.5\,(n+1) \frac{M^*}{\rho_1} \tag{39}$$

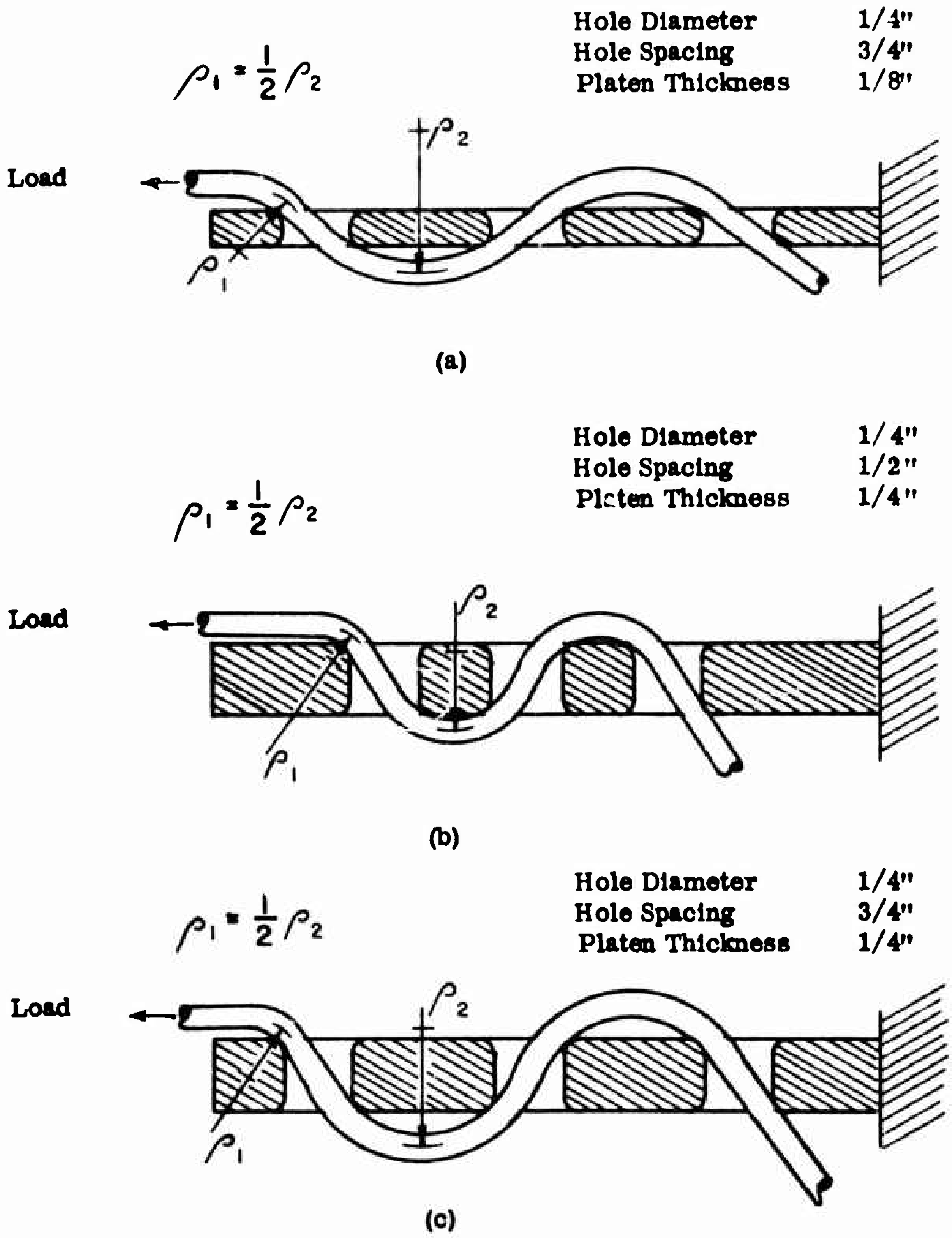


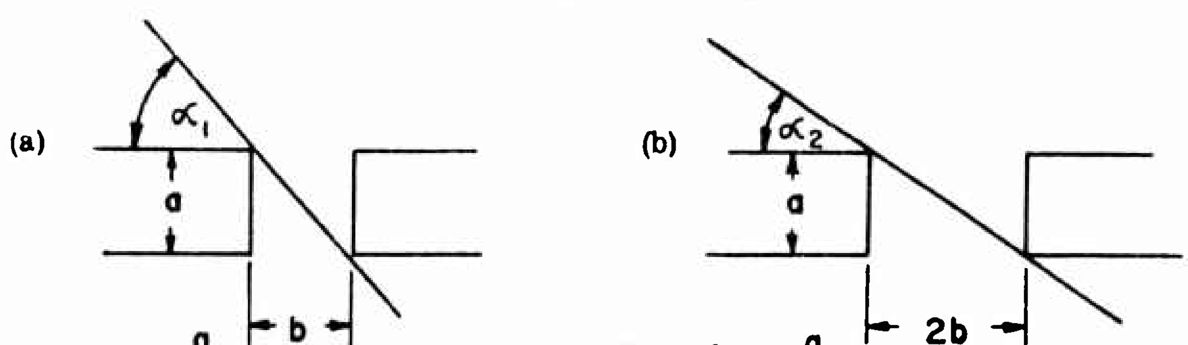
Figure 23. Geometry of Counterflexing Load Limiter

It is evident from the above equations that the maximum bending moment, combinded with the minimum radius of curvature, will give optimum pull load. Bending moment of the wire will depend on its diameter. The radius of curvature of the wire will depend on hole diameter, hole spacing, and platen thickness. In addition, platen material will affect the coefficient of friction, and weaving arrangements, such as a staggered hole pattern, will more than likely augment the pull load over and above a straight hole pattern.

ANALYTICAL STUDY

An analytical study was undertaken to predict the effects of the variables associated with the platen and wire in order to conduct an experimental study. The findings regarding the variables investigated--hole diameter, hole spacing, platen thickness, wire diameter, platen and wire materials, and weaving arrangements--are as follows:

1. Hole Diameter: The angle that the wire will make when threaded through a hole in a plate will depend on hole diameter in addition to some of the other variables. Figure 24, below, is indicative of this. It is obvious from the figure that, as the hole diameter increases, the pertinent angle decreases.



(a) $TAN\, \alpha_1 = \frac{a}{b}$ where $a = b$ and $\alpha_1 = 45°$

(b) $TAN\, \alpha_2 = \frac{a}{2b}$ where $2b$ is twice a and $\alpha_2 = 26.5°$

Figure 24. Geometry of Wire Threaded Through Hole in Platen

The converse is also true. Therefore, the smaller the hole diameter, the smaller the radius of curvature of the wire woven through the platen. From equation (35), the radius of curvature is an inverse function of the pull load, leading to the conclusion that the smaller the hole diameter, the greater the pull load.

2. Hole Spacing: The effect that hole spacing has on pull load is easily realized when wire bending is considered. As the holes in a platen are spaced closer together, the bend in the wire, as woven, would increase, or the wire radius of curvature would decrease. Referring to equation (35) again, the pull load is of greater intensity at the smallest hole spacings.

3. Platen Thickness: Platen thickness is associated in a direct relationship with wire bending when the wire is threaded through the platen. An increase in platen thickness will result in greater bending of the woven wire; therefore

platen thickness is universally proportional to the wire radius of curvature. Subsequently, referring to equation (35), a higher intensity in pull load is evident with increased platen thickness.

4. Wire Diameter: As the wire diameter increases, the bend in the threaded wire increases. This can be illustrated if two identical platens with two different diameter wires woven through the platens are considered. See Figure 25. The

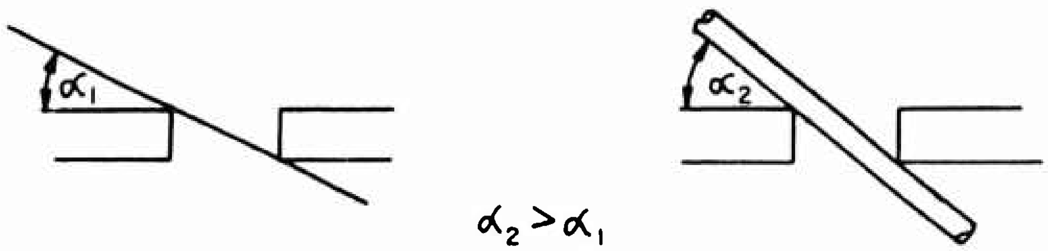


Figure 25. Geometry of Wire Diameter Versus Hole Size in Platen

angle between platen and wire is increased with increased wire diameter. Therefore, the radius of curvature is smaller for the larger wire diameter. This results in a greater wire pull load as defined by equation (35). However, the storage area for 2 feet of wire will be a deciding factor on the size of wire that is feasible.

5. Platen and Wire Materials: Three types of platen material seem to be satisfactory for counterflexing load limiter design. They are 6061-T6 and 2024-T3 aluminum and 304 annealed stainless steel. The 6061-T6 aluminum is soft material as compared to the 2024-T3 aluminum. On the other hand, the 304 annealed stainless steel is the hardest material of the three under consideration. During loading, the soft material may easily be gouged where the wire is in contact with the plate. This could result in a mean pull load that is excessively low. However, during loading, a hard material such as stainless steel may cause excessive resistance with the wire. The 2024-T3 aluminum material falls in a hardness category between the 6061-T6 aluminum and the stainless steel material. These three materials will be sufficient to give a comparison of the effect that platen hardness will have on the wire utilized. The most desirable wire material is annealed stainless steel because of its high strength and its bending capabilities. This is necessary so that a high pull load, easy storing, and weaving of the wire can be accomplished.

6. Weaving Arrangements: It is desirable to test a platen with straight-line holes and staggered holes in order to determine the optimum pull load condition. A minimum of three straight holes in the platen is sufficient to inaugurate the test program. This is verified from the Government-supplied data (page 28). The weaving arrangements will be determined as the test events are concluded. This will help to minimize the number of tests required.

From the above, it is evident that the pertinent factors required to achieve the optimum pull load can be accomplished by utilization of the maximum feasible diameter wire woven through a plate with the smallest hole diameter and hole spacing and the largest reasonable thickness.

EXPERIMENTAL STUDY

To initiate the test program, a preliminary test plan was written which specified annealed stainless steel wire from 0.62 to 0.120 inch in diameter in various combinations with the following:

1. Number of wires utilized (1, 2, and 4)
2. Plate material (6061-T6 aluminum, 2023-T3 aluminum, and 304 annealed stainless steel)
3. Plate thickness (1/8, 1/4, 5/16, and 3/8 inch)
4. Hole diameter (1/4, 5/16, 3/8, 7/16, and 1/2 inch)
5. Hole spacing (1/2, 5/8, 3/4, and 1 inch)
6. Weaving arrangement (straight and staggered)

As the tests were conducted, the test plan was revised as warranted by the test results. Table XII, Appendix I, is the final plan as utilized, and Tables V through X depict the compiled data which includes both mean and peak pull load. The columns of Tables V through X were arranged in an orderly manner in order to correlate the data.

A Baldwin-Emery SR-4, Model F. G. T., test machine was utilized. The test machine has four scales: 0-1,000 pounds, 0-2,500 pounds, 0-10,000 pounds, and 0-50,000 pounds. In the case of this test program, the two scales, which can be read with the most accuracy, 0-1,000 and 0-2,500 pounds were used. The test specimens were attached to the machine in two different manners. See Figures 26 and 27. The wire of the test specimen in Figure 26 was looped at one end. The looped and platen ends were connected to a pin and clevis combination, which was attached rigidly to the jaws of the machine. For the other specimen, one free end of each wire and the platen end with a pin and clevis combination were attached rigidly to the jaws of the machine (see Figure 27). The looped wire test specimens

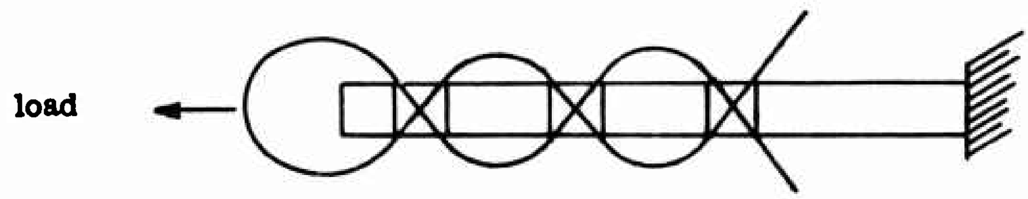


Figure 26. Single Wire Threaded Through Platen

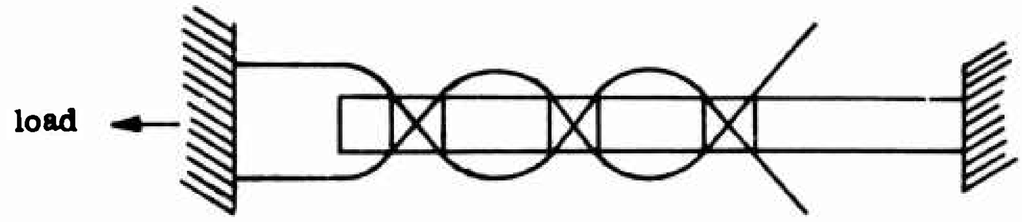


Figure 27. Two Wires Threaded Through Platen

were attained by threading a single wire through the platen. The first 60 tests were conducted in this manner. A change to the other type test specimen (Figure 27) was then undertaken because it became apparent that the pull load was being adversely affected by inherent kinks and local bends in the free end of the wire as a result of the threading operation. This operation consisted of folding the wire in half and threading each end as individual wires. The free end of the wire is first threaded through the platen from the looped side. The threading operation of the second method utilized two single wires which are woven through the platen from the free end side. Upon completion of the threading operation, the free end wire has not been touched, and no kinks or local bends are apparent.

The tests, in general, were conducted with the holes in the platen either broken at edges or chamfered, or the edges were polished to a rounded contour. These hole variations were an attempt to eliminate or reduce the peak load that occurs during the initial stage of each test. The peak load intensity using these methods was not reduced sufficiently. Another method was utilized with excellent results. This method consists of pre-pulling the wire for approximately 1 inch and then backing off to zero load. This procedure established a peak and mean load. The test was restarted, and only the mean load was realized.

The first two tests, as depicted in Table VIII, were conducted to duplicate the Government-supplied data. For the first test, the holes in the platen were edge-broken, and for the second test the holes were chamfered. All other variables were the same. The mean pull load for the first test was approximately 150 pounds above the pull load for the second test. The pull load utilizing the platen with just the hole edges broken reproduced the Government data within a reasonable tolerance.

Table XI (page 91) was constructed for the purpose of delineating the wire ultimate tensile strength for the particular wire sizes under consideration. The various diameter wires were pulled to destruction; their ultimate intensities are tabulated in column 2 of the table. The ultimate loads depicted in column 3, titled "80 percent of Ultimate Load", are considered to be pertinent load limiter working loads that will provide a reasonable factor of safety. This is attributed to the fact that the failure load of a wire-platen combination has been found to result in a slightly lower intensity than the ultimate strength of the wire.

STRAP DEVICE WITH SELF-STORING, WIRE-BENDING ENERGY ABSORBERS (LOAD LIMITERS)

From the above analytical/experimental study, a self-storing, wire-bending energy absorber was adopted. This concept has provisions for a storage area capable of containing the number of wires required to pull through the platen at an intensity of 5,000 pounds for the design stroke of 2 feet (see "Design Cargo Weights", page 20).

To achieve simplicity and to minimize weight and production problems, the most promising design utilizes 0.105-inch-diameter annealed stainless steel wires threaded through a 2024-T3 aluminum platen. See Figure 28. The runout wires

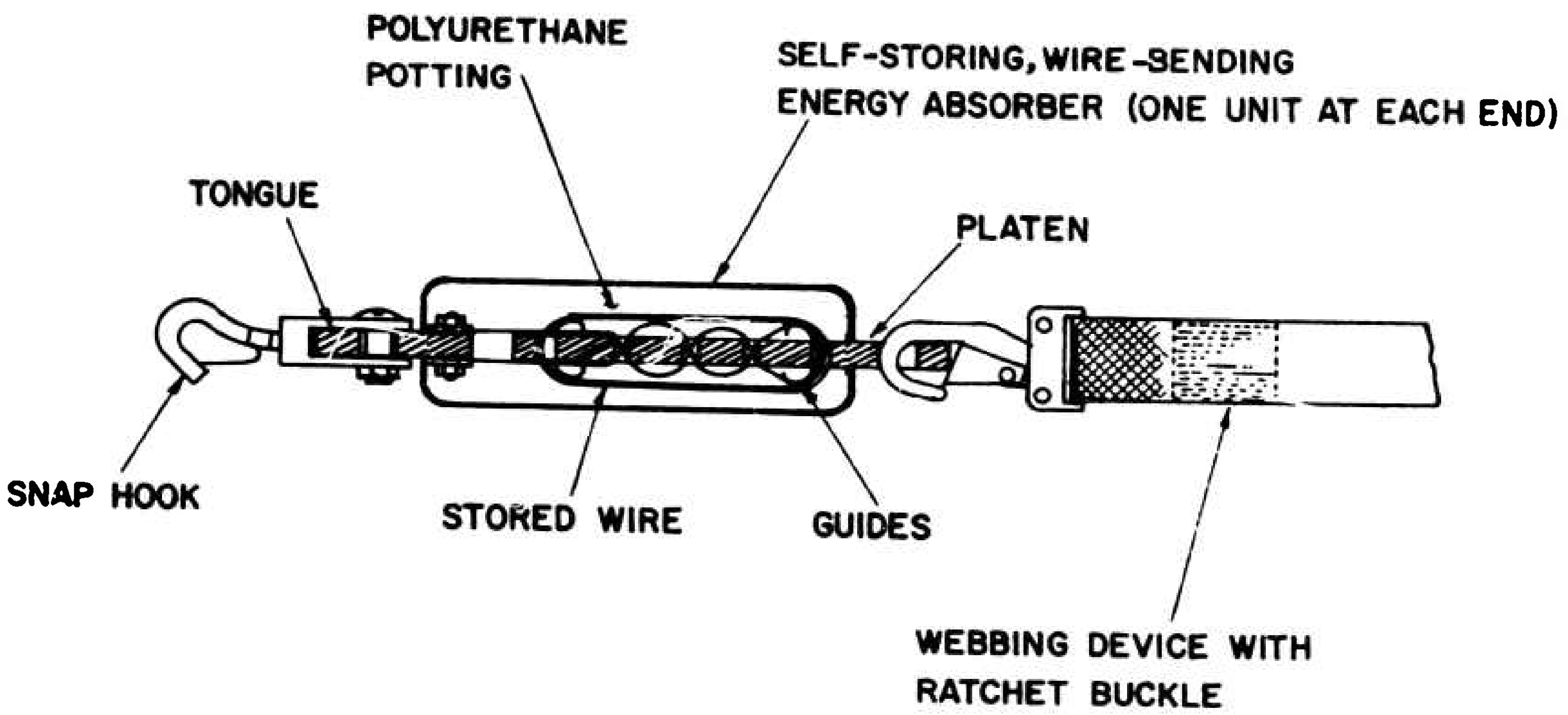


Figure 28. Strap Device With Self-Storing, Wire-Bending Energy Absorbers (5,000-Pound Rated Capacity)

are stored around the platen and around two guides (each end) which keep the wires from binding as they unwind. The above-mentioned wire in combination with the platen can be optimized at 500 pounds; therefore, ten wires would be required to achieve a unit of 5,000-pound capability. The hole diameter will be replaced with a 1/4-inch slot elongated in the width direction of the platen for weaving ten wires, with five wires woven in one direction and five in the opposite direction as depicted in the figure. The hole spacing is 3/4 inch and the platen thickness is 1/4 inch. The first working hole edge nearest the tongue, or leading hole edge, has its edge rounded (contoured to a 1/8-inch radius). The other edges are rounded to a 5/64-inch radius. These radii are a result of testing units capable of storing 8 inches of stroke as requested by the Government. The tested units require one set of guides at one end of the platen only. It is felt that back tension on the guides has contributed to the total load in excess of the desired intensity. Consequently, the use of guides at both ends of the platen for a 24-inch storing stroke will require adjustments in hole spacing, hole size, number of holes, and wire size. Such changes deemed necessary will be accomplished in order that the load effect due to the storage area is a small percentage of the design load.

To protect the wires and platen from inadvertent damage due to handling, the load limiter is potted in polyurethane. A mylar polyester film in sheet form is wrapped over the unit prior to potting, which will allow the wire to move freely under load. In turn, the polyurethane cover will keep the wires in its formed position as it unwraps around the guides. No appreciable load will be realized when the wires are contained in the above described manner.

Pre-pulling of the device approximately 1 inch prior to installation of the tongue and potting will tighten the wire into place and eliminate any peak load that may be realized during operation.

A Dacron web tie-down device will be attached in series with two self-storing, wire-bending energy absorbers. Already a production item in tie-down devices, its elongation characteristic is approximately 2.5 percent at 5,000-pound capacity when reacting to dynamic impact conditions. Under static loading, Dacron has about a 5.2-percent elongation at 5,000-pound capacity. The capacity of the web device with fittings should be capable of 5,000 pounds, which is compatible with the preponderance of Army aircraft floor tie-down fittings. The relatively few floor fittings available at 10,000-pound capacity do not warrant 10,000-pound-capacity restraint devices because of the inherent elasticity problem (page 14), which is a result of mixing variant strength restraint devices. These devices are pre-tensioned to a small percentage of the device capacity by use of a standard ratchet device.

The above discussed self-storing, wire-bending energy absorber can be easily augmented to a 10,000-pound rated capacity. Two platens can be mounted, one above the other, and the runout wire will be wrapped around both platens instead of one. See Figure 29. All other aspects, both physical and operational, are the same as the single platen unit. However, a 10,000-pound-capacity Dacron tie-down device, including the end fittings, is required to complete the restraint system.

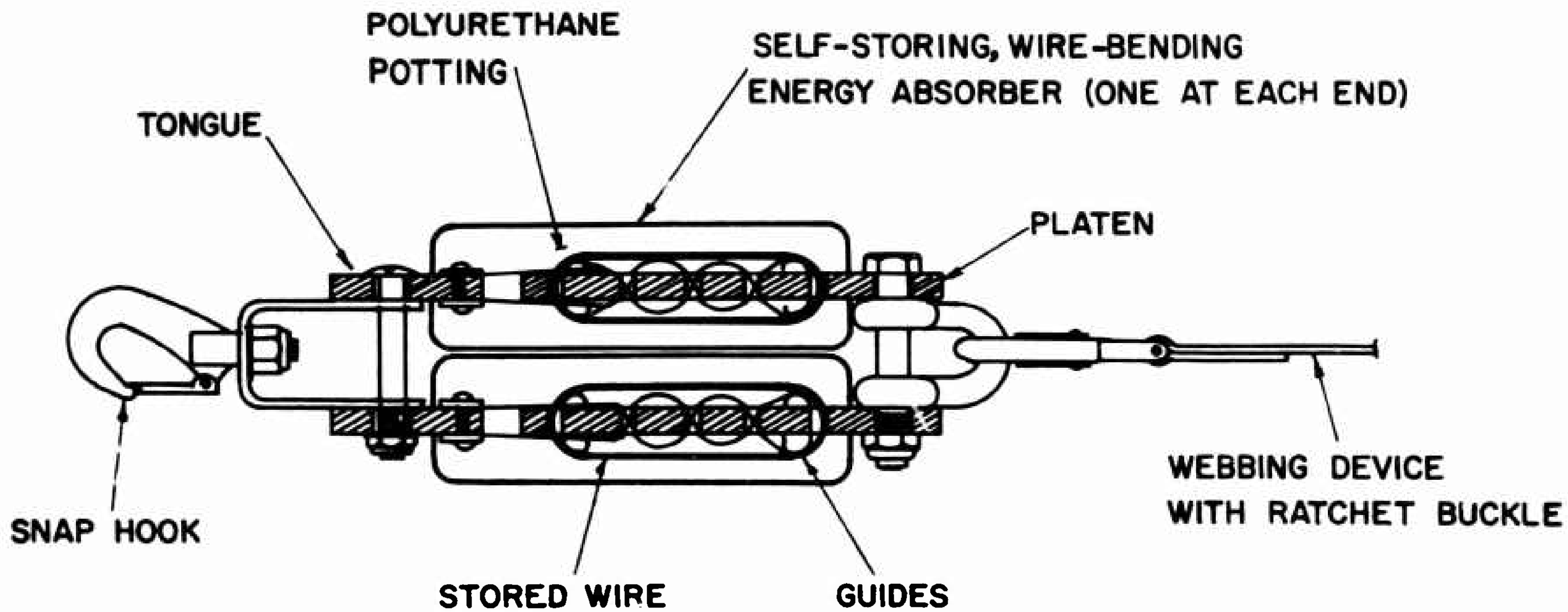


Figure 29. Strap Device With Self-Storing, Wire-Bending Energy Absorbers (10,000-Pound Rated Capacity)

The Dacron tie-down device is detachable from the load limiters. A need for this may arise when one end of the restraint device is wrapped around the axle of a jeep and the other end attached to a floor fitting. One load limiter is used, which is at the floor end. In addition, replacement of any of the separate parts can be accomplished without replacing the complete restraint device.

Other configurations of the wire-bending energy absorber that have been considered to have some merit are discussed in Appendix II.

Five- and ten-thousand-pound units were designed, fabricated, and static tested. These tested units were the basic energy absorbers without the strap device and end snap hooks. In addition, they were designed for an 8-inch stroke as requested by the Government. The results of the tests, with photographic coverage, are depicted in Appendix III.

STRAP DEVICE WITH TUBE-BALL ENERGY ABSORBERS (LOAD LIMITERS)

The strap device to be utilized with the tube-ball load limiters is the same as the one discussed in the previous section. The tube-ball energy absorbers can be used in place of the self-storing, wire-bending energy absorbers.

The tube-ball load limiter is basically a tube with an inserted ball. See Figure 30. When the ball and tube are loaded to 5,000 pounds, the ball is pulled through the tube, deforming the wall of the cylinder. The design stroke will be limited to 2 feet. The total cargo stroke will be the combined load limiter stroke and webbing deflection. The ball will be swaged to a 1/4-inch-diameter steel cord, which will extend through the end of the tube. This end of the steel cord is swaged to a shackle or eye hook for attachment to the end fitting of the webbing device. At the same end of the tube where the ball is located, a fitting is designed to facilitate a snap hook to tie to floor fittings.

A 10,000-pound tube-ball energy absorber can be obtained by redesigning the ball, tube, and all pertinent parts that have to be augmented from 5,000- to 10,000-pound capacity. The load limiter would take the same physical shape and be as operable as the 5,000-pound unit. The strap device would also be of 10,000-pound rated capacity.

STRAP DEVICE WITH ENERGY ABSORBERS (LOAD LIMITERS) AND PULLEYS

A means of obtaining a 10,000-pound-capacity restraint system with the use of 5,000-pound-capacity energy absorbers is accomplished by utilizing a Dacron type tie-down strap attached to pulleys at each end. Each pulley is linked to two energy absorbers by a steel cord with snap hooks. See Figures 31 and 32.

Since each load limiter device will be of 5,000-pound capacity, the capacity of the webbing with fittings will have to be capable of 10,000 pounds. The Dacron web device previously discussed is applicable.

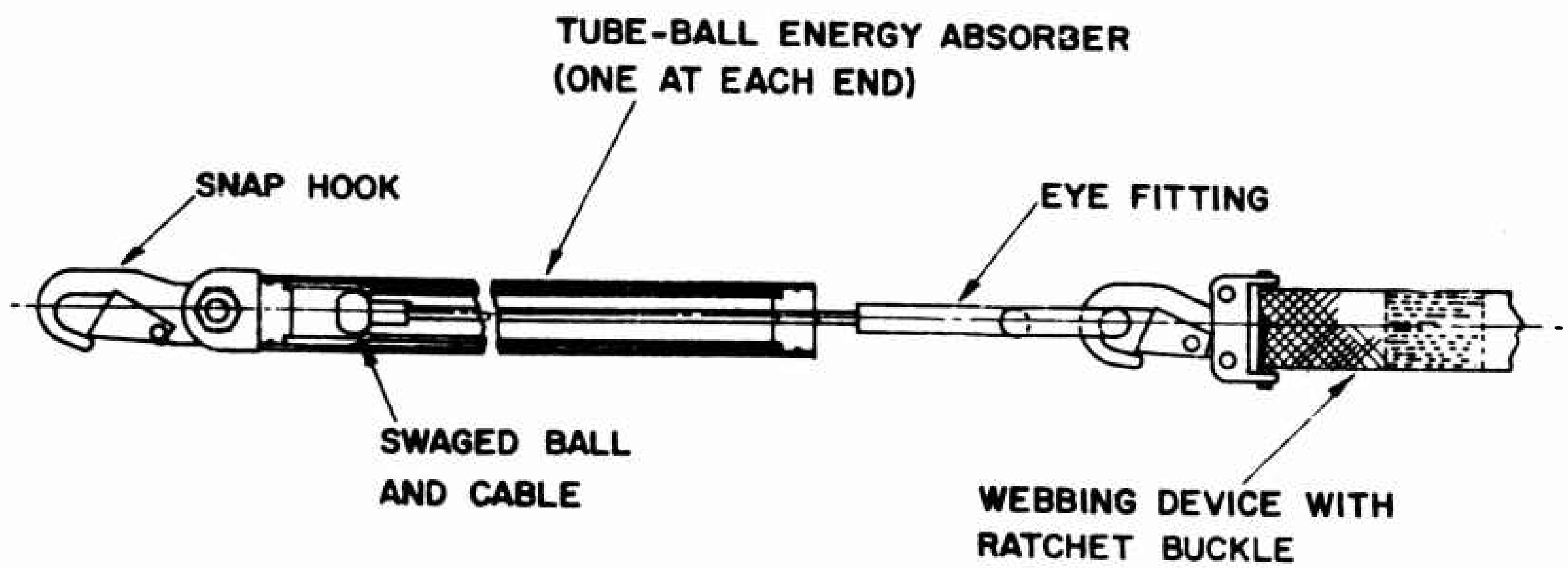


Figure 30. Strap Device With Tube-Ball Energy Absorbers

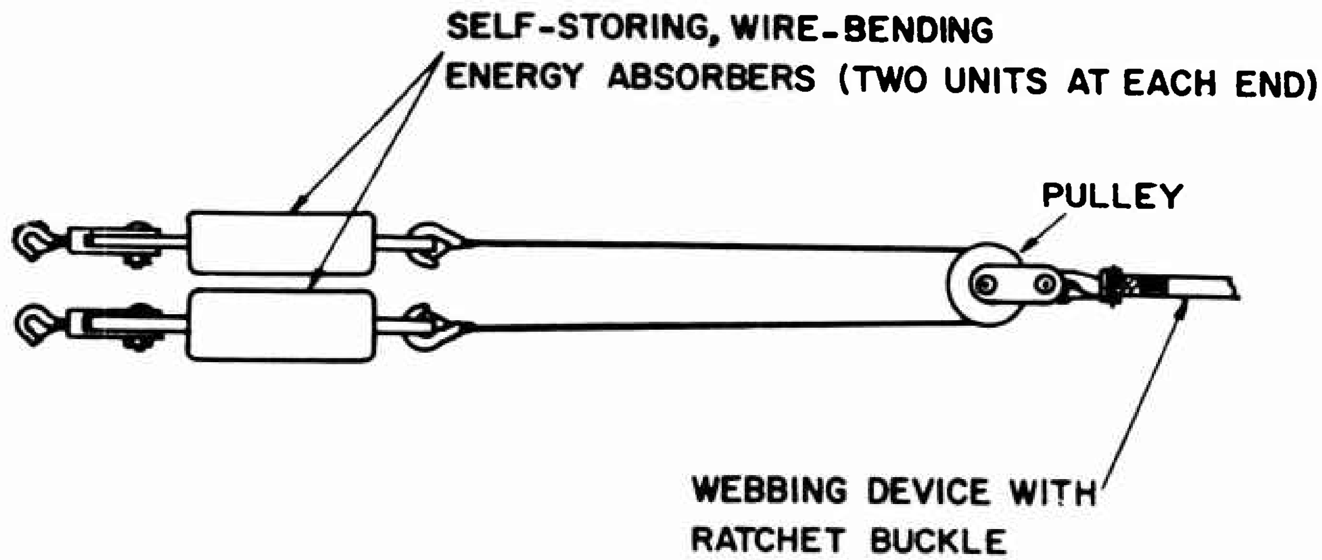


Figure 31. Strap Device With Self-Storing, Wire-Bending Energy Absorbers and Pulleys

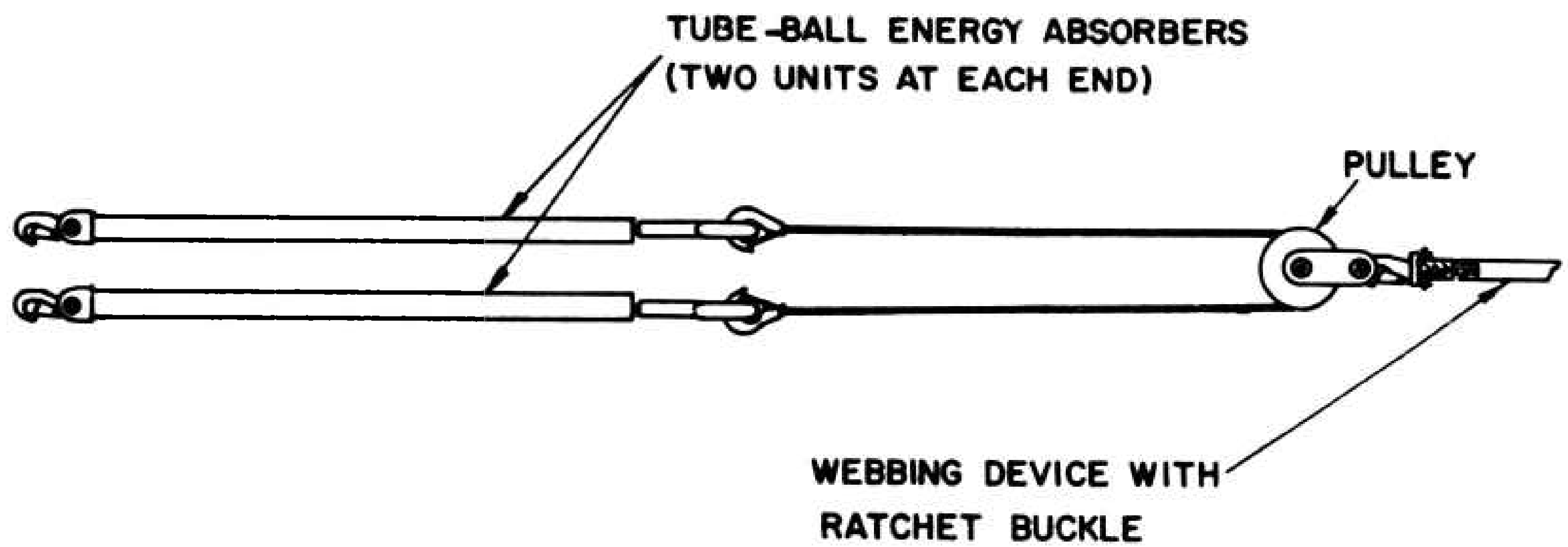


Figure 32. Strap Device With Tube-Ball Energy Absorbers and Pulleys

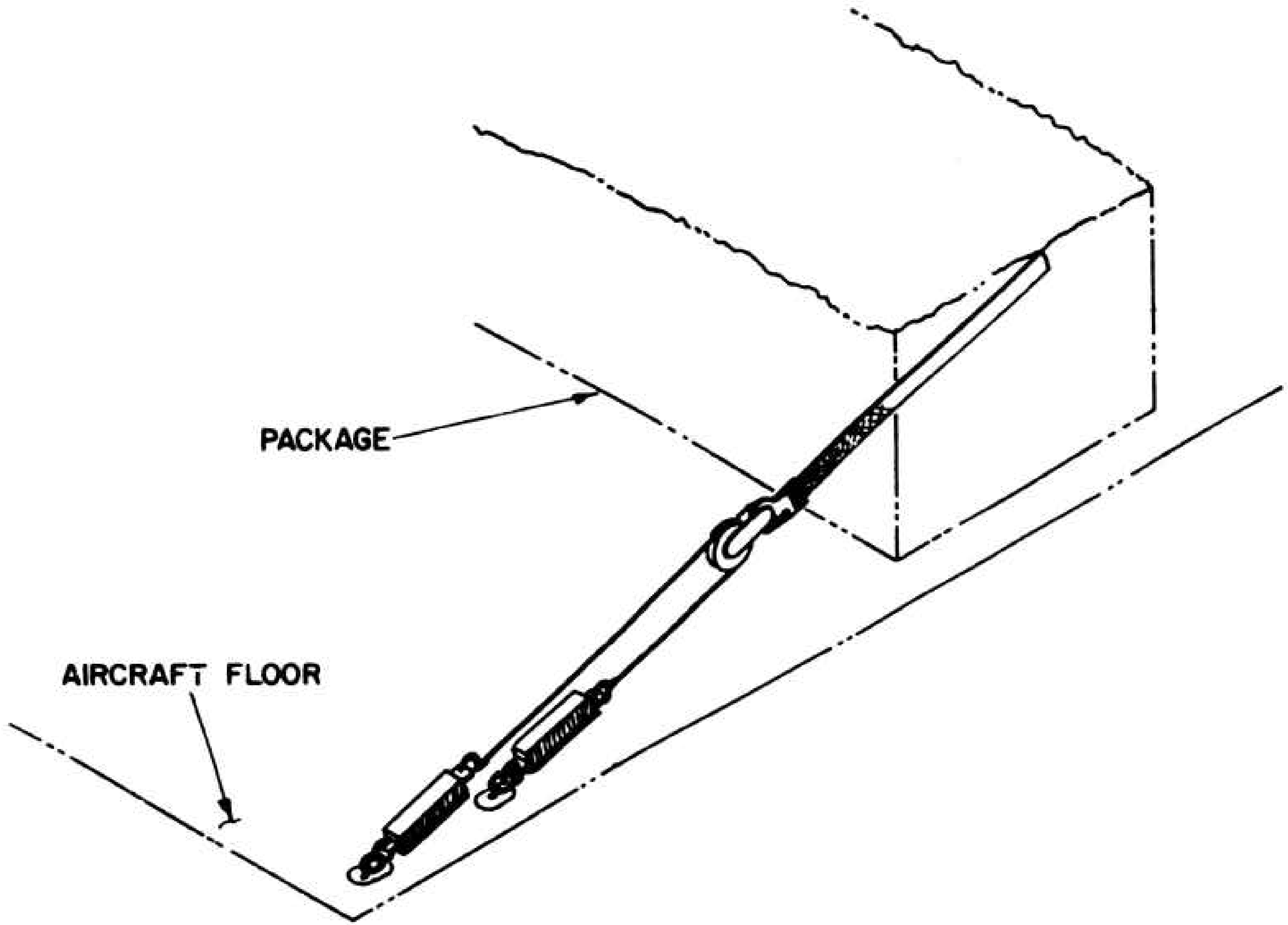


Figure 33. One End of Strap Device With Energy Absorbers and Pulleys Fastened to Two Floor Tie-Down Fittings

The pulley distributes the dynamic load from the cargo to the energy absorbers. Figure 33 depicts one end of the restraint device tied to two adjacent tie-down fittings. Neglecting the fact that the total restraint will advantageously rotate downward as the cargo moves forward, the maximum load reacted by the webbing restraint (10,000 pounds) will have distributed evenly to each energy absorber (5,000 pounds) through the steel cord. This is attributed to the fact that the cord leading from each load limiter will be of equal strain. This is accomplished by rotation of the pulley as the webbing restraint deflects. When the restraint device rotates downward with forward cargo movement, the device tends to approach the longitudinal direction; consequently, it affords greater forward restraint.

The webbing device, energy absorbers, and pulley system are detachable items. The purpose is to be able to connect the webbing and energy absorbers as a restraint device without the pulley components. In addition, replacement of any of the separate parts can be accomplished without replacing the complete restraint device.

The pulley distributes the dynamic load from the cargo to the energy absorbers. Figure 33 depicts one end of the restraint device tied to two adjacent tie-down fittings. Neglecting the fact that the total restraint will advantageously rotate downward as the cargo moves forward, the maximum load reacted by the webbing restraint (10,000 pounds) will have distributed evenly to each energy absorber (5,000 pounds) through the steel cord. This is attributed to the fact that the cord leading from each load limiter will be of equal strain. This is accomplished by rotation of the pulley as the webbing restraint deflects. When the restraint device rotates downward with forward cargo movement, the device tends to approach the longitudinal direction; consequently, it affords greater forward restraint.

The webbing device, energy absorbers, and pulley system are detachable items. The purpose is to be able to connect the webbing and energy absorbers as a restraint device without the pulley components. In addition, replacement of any of the separate parts can be accomplished without replacing the complete restraint device.

remaining cases, restraint was inadequate. The existing static load factors criteria (page 19) still apply to these aircraft directions. Figure 36 delineates the tie-down of two jeeps. The forward vehicle is restrained in the forward direction by means of two straps and two chains, and the aft jeep is restrained only by chains. As discussed under the section "Elasticity Problems", mixing of straps and chains is unfavorable because of the variance in their elongation characteristics. Also, chain is undesirable for restraint because the device may be responsive to high-frequency oscillations that may occur under crash conditions.

2. Existing Methods with Correct Restraint: It is obvious from Figures 40 through 42 that the Army is very much limited in the cargo weight that can be correctly restrained with existing methods. The inability to restrain cargo correctly with existing methods can be traced to the excessive number of forward restraints required, predicated on the equivalent static load factor of 39 g's (page 14).

3. Proposed Strap Device with Energy Absorbers and Pulleys: Figures 43 through 52 delineate the proposed strap device with energy absorbers and pulleys. The preferred energy absorbers are the self-storing, wire-bending type. This design concept is used to restrain cargo correctly and is predicated upon the pulse duration envelope criteria. The equivalent static load factor of 15 g's (page 26) is used to determine the forward restraints required.

This proposed design concept is capable of restraining maximum cargo payload with a tremendous saving in the number of forward restraints used.

OPERATIONAL FEASIBILITY

Operational feasibility demonstrates the techniques as used and proposed for a cargo restraint system. Included herein are restraint methods, estimated time required for restraint, restraint devices, and floor space utilization.

The column headings of Table I, "Operational Feasibility", are defined as follows:

Report Page: This defines the location of the figures depicting the subject matter.

Cargo Weight: Information obtained from the report page listed under column 2.

Number of Restraint Devices Used: Information obtained from the figures listed on the report page shown under column 2.

Equivalent Cargo Weight for Devices Used: The weights listed for the existing Army method are based on the restraint capabilities of the devices. The weights listed for the existing method with correct restraint and for the proposed strap device with load limiters and pulleys are based on the actual cargo weights. In some cases, the actual cargo weights are lower than the restraint system capabilities as calculated, but the nominal value was used for comparison purposes.

Equalizing Factor: An equalizing factor is a method of rationalizing the values in columns 7, 10, 13, 15, and 17. The equalizing factor is derived by dividing the values of column 5 by the values of column 3. This is an attempt to align the Army restraint method results with those achieved by the other restraint methods so that an equable comparison can be made.

Method Effectiveness: Method effectiveness is a measure of the relative restraint capabilities of the three cargo restraint techniques. This value is computed by dividing the values in column 6 by those in column 4 and then by multiplying by a rationalizing factor. This factor is used solely to present the values of the method effectiveness column as nonfractional numbers.

Space Occupied, Cargo and Tie-Down Devices: The column represents that portion of the cargo compartment floor space obscured by the cargo and its tie-down devices. See Figure 34.

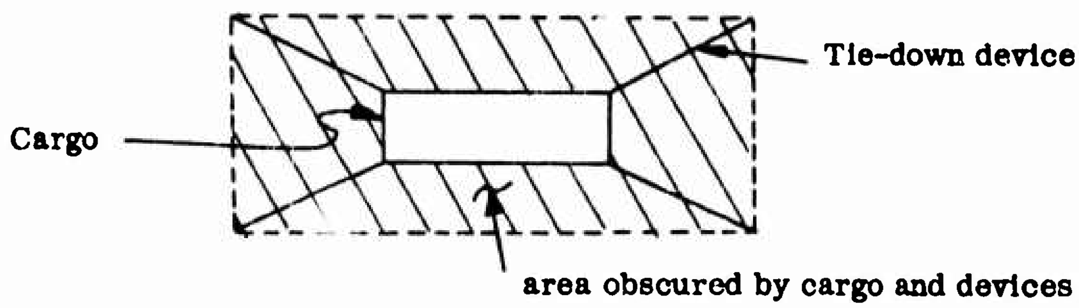


Figure 34. Aircraft Floor Space Occupied by Cargo and Tie-Down Devices

Cargo Space Occupied: This is the space actually obscured by the cargo (see Figure 34).

Space Effectiveness: This is the ratio of the cargo and restraint area to the cargo floor area. The values are obtained by dividing the values of column 9 by the values of column 8 and then by multiplying the result by the values of column 6 and a rationalizing factor.

Tie-Down Fittings Unused After Restraint: Found by examination of the figures on the pages listed under the column "Report Page." When pallets are used, the number of floor fittings covered by the pallets must be counted among those used.

Floor Fittings Before Restraint: These are the basic floor fittings provided for cargo tie-down. The number of floor fittings will vary, depending on whether the roller conveyors are installed.

Tie-Down Fitting Effectiveness: The values of this column are determined by dividing the values of column 11 by the values of column 12 and then by multiplying this result by the values of column 6 and a rationalizing factor.

Number of Unused Devices: The total number of devices available appears on page 46. The total number of devices used is shown in the tables accompanying Figures 35 through 52.

Device Effectiveness: Device effectiveness is a means of evaluating the restraint technique used. The values of this column are computed by dividing the values of column 14 by the values of column 4 and then by multiplying the result by the values of column 6 and a rationalizing factor.

Estimated Restraint Time: This is the time taken to restrain cargo. The times for the existing Army restraint methods were obtained from field observation and discussion with qualified personnel. The times for the other restraint techniques were estimated, based on the Army restraint times and techniques.

Estimated Time Effectiveness: This is a method of evaluating the effectiveness of all three cargo restraint techniques. The values of this column are obtained by dividing the values of column 6 by the values of column 16 and then by multiplying by a rationalizing factor.

PAYLOAD COMPARISON

The payload comparison table (Table II) evaluates the effectiveness of the strap device with load limiters and pulleys concept as compared with the existing method with correct restraint. The Army restraint method was not delineated herein because it does not represent a complete restraint system.

The column headings of the "Payload Comparison" table are defined as follows:

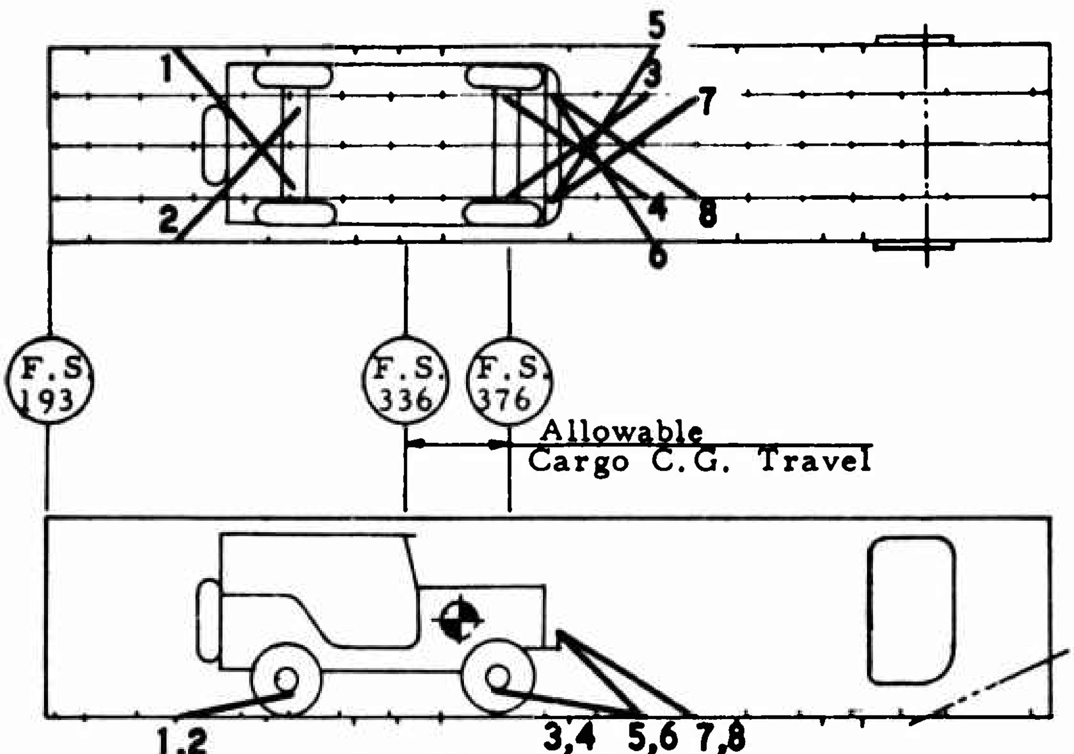
Cargo Weight: Information obtained from the figures on pages 50 through 67.

Total Number of Restraint Devices Used: This heading covers two columns entitled "Existing Method with Correct Restraint" and "Proposed Strap Device with Load Limiters and Pulleys." The values for these two columns are found in the figures on pages 50 through 67.

Number of Devices for Forward Restraint: This heading covers two columns entitled "Existing Method with Correct Restraint" and "Proposed Strap Device with Load Limiters and Pulleys." The values for these two columns are found in the figures on pages 50 through 67. These two columns are similar to the columns mentioned above.

Omnidirectional Restraint Comparison: This is a column providing an index of the effectiveness of the correct restraint techniques. The values for this column are computed by dividing the values of column 1 by the values of column 2.

Forward Restraint Comparison: This is a column providing an index of the effectiveness of the forward restraint as defined by the correct restraint techniques. The values for this column are determined by dividing the values of column 3 by the values of column 4.

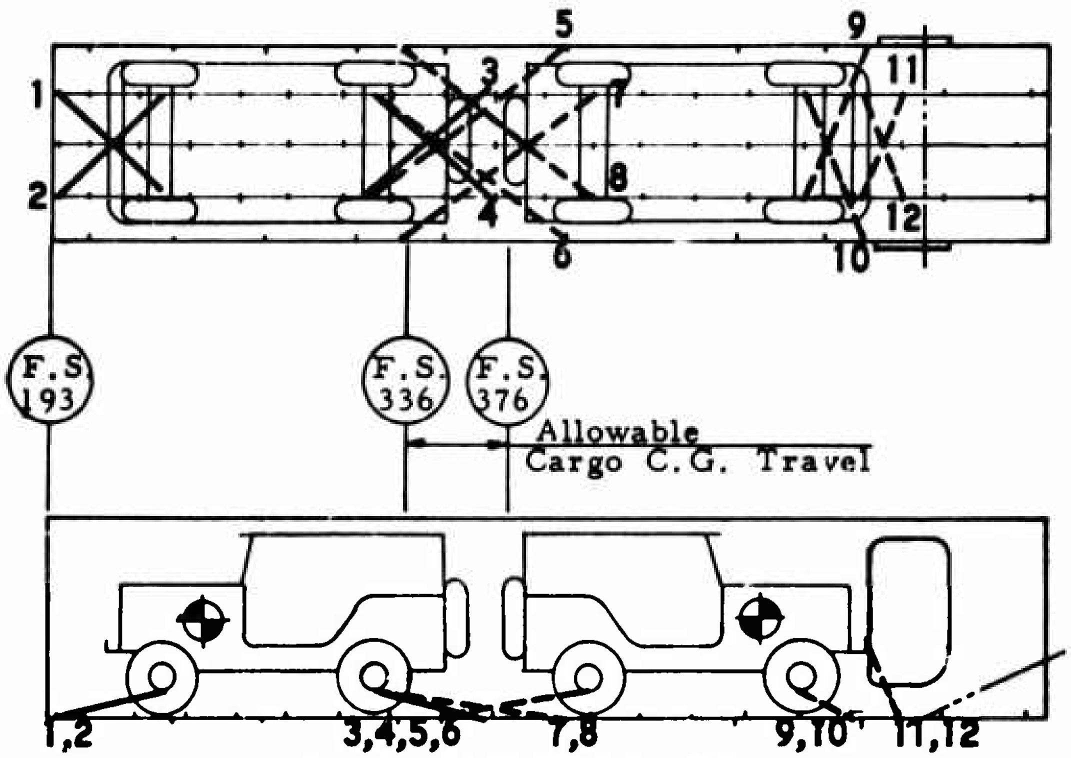


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt. ____				
Midship Group Devices Used { 8 Straps Cargo Wt. 2,665 lb	3 8 4 5 6 7	1 2	1 6 2 7 3 8 4 5	1 6 2 7 3 8 4 5
Aft Group				
Devices Used				
Cargo Wt. ____				

TOTAL OF RESTRAINT DEVICES = 8

Figure 35. Restraint of 1/4-Ton Utility Truck Using Existing Army Methods (Cargo Weight, 2,665 Pounds; Air Transported)



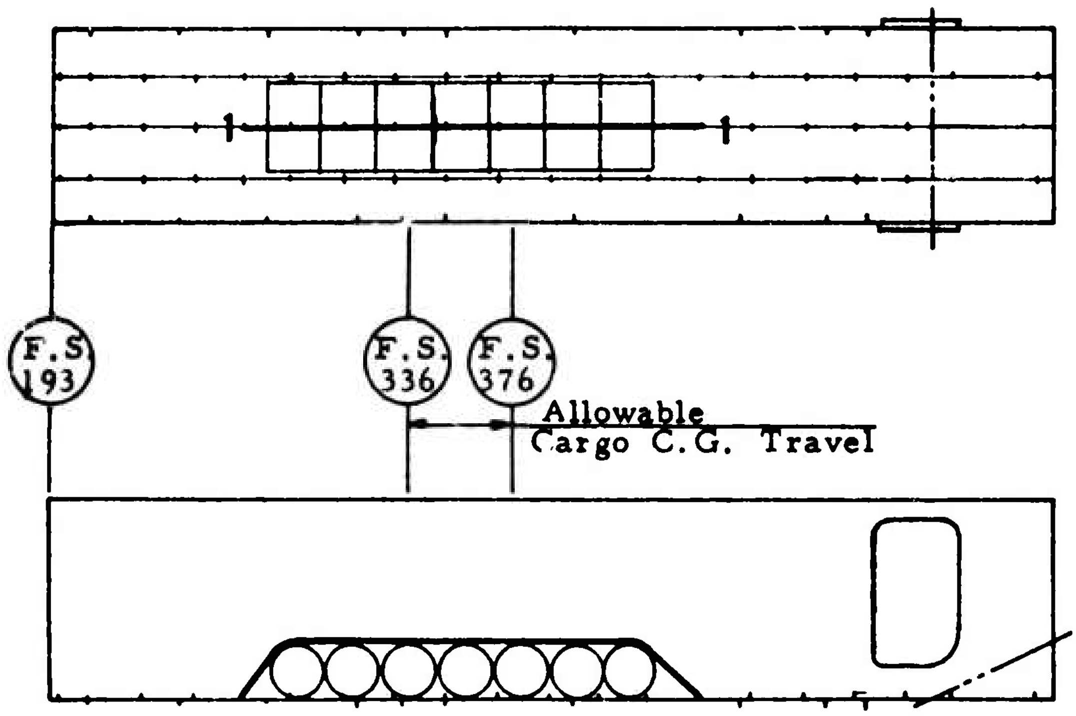
CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 2 Chains 4 Straps Cargo Wt. 2,665 lb	3 4 5 C 6 C	1 2	1 6 C 2 3 4 5 C	1 6C 2 3 4 5 C
Midship Group Devices Used { 6 Chains Cargo Wt. 2,665 lb	9 C 10 C 11 C 12 C	7 C 8 C	7 C 12C 8 C 9 C 10C 11C	7C 12C 8C 9C 10C 11C
Aft Group Devices Used { Cargo Wt.______				

TOTAL OF RESTRAINT DEVICES = 12

C = CHAIN

Figure 36. Restraint of 1/4-Ton Utility Truck Using Existing Army Methods (Cargo Weight, 5,350 Pounds; Air Transported)

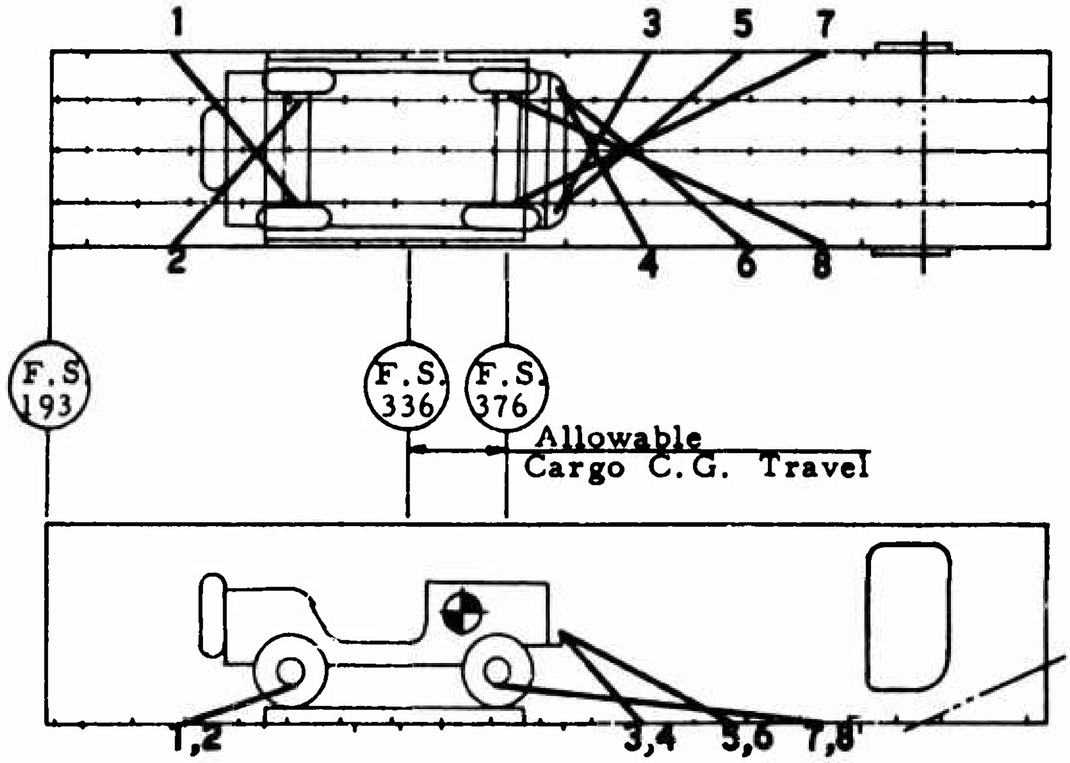


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt.				
Midship Group				
Devices Used: 1 Strap	1D	1D		1D
Cargo Wt. 2,513 lb				
Aft Group				
Devices Used				
Cargo Wt.				

TOTAL OF RESTRAINT DEVICES = 1
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 37. Restraint of Seven 55-Gallon Drums Using Existing Army Methods (Cargo Weight, 2,513 Pounds; Air Transported)

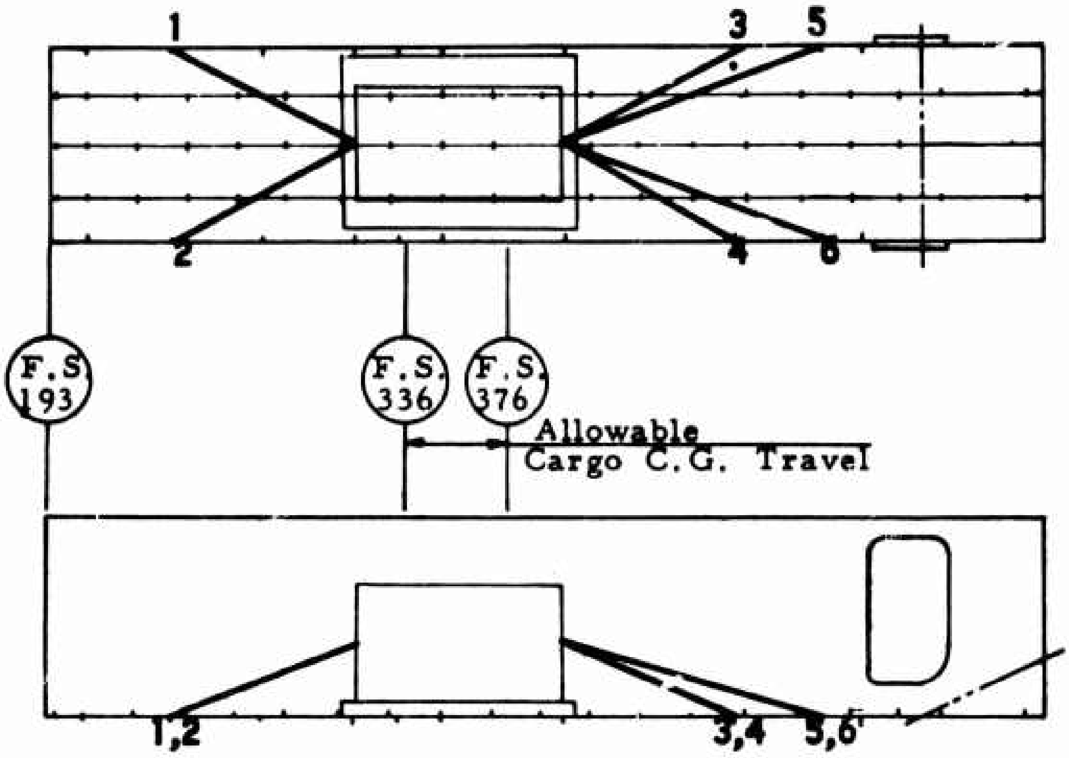


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt. ______				
Midship Group	3 8	1	1 6	1 6
Devices Used {8 Straps	4	2	2 7	2 7
	5		3 8	3 8
	6		4	4
Cargo Wt. 2,665 lb	7		5	5
Aft Group				
Devices Used				
Cargo Wt. ______				

TOTAL OF RESTRAINT DEVICES = 8

Figure 38. Restraint of 1/4-Ton Utility Truck With Pallet Using Existing Army Methods (Cargo Weight, 2,665 Pounds; Air Dropped)

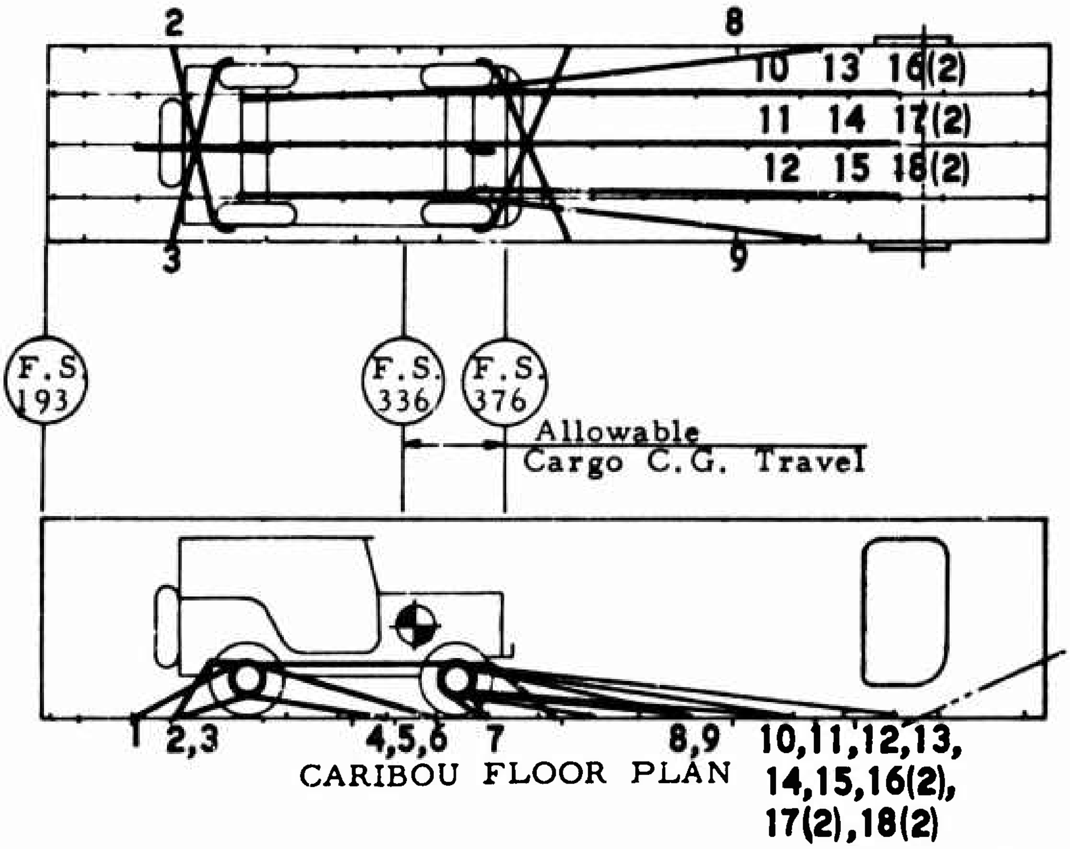


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt.____				
Midship Group	3	1	1 6	1 6
Devices Used { 6 Straps	4 5 6	2	2 3 4 5	2 3 4 5
Cargo Wt. 3,500 lb				
Aft Group				
Devices Used				
Cargo Wt.____				

TOTAL OF RESTRAINT DEVICES = 6

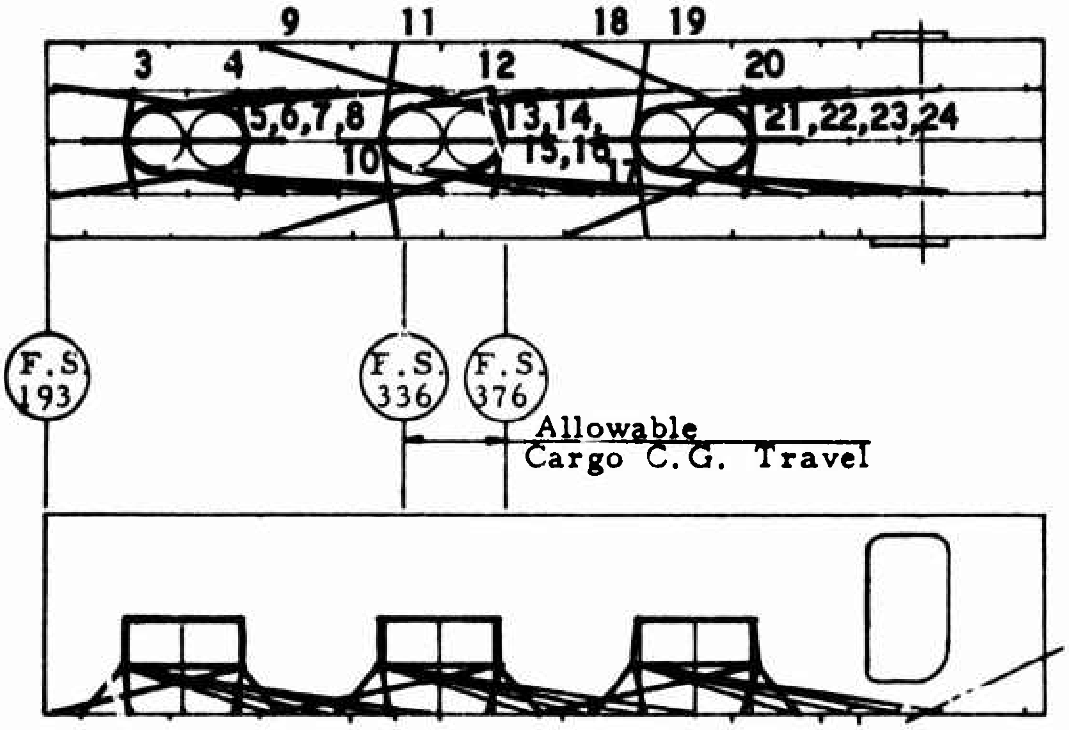
Figure 39. Restraint of Sealdbin "70" Container Using Existing Army Methods (Cargo Weight, 3,500 Pounds; Air Dropped)



RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt._____				
Midship Group	4D 12D	1D	2D	1D
Devices Used 21 Straps	5D 13D		3D	7
	6D 14D			
	8D 15D			
Cargo Wt. 2,665 lb	9 16D (2			
Aft Group	10D 17D (2			
Devices Used	11D 18D (2			
Cargo Wt._____				

(2) = ADDITIONAL STRAP LENGTH REQUIRED
TOTAL OF RESTRAINT DEVICES = 21
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 40. Restraint of 1/4-Ton Utility Truck Using Existing Methods With Correct Restraint (Cargo Weight, 2,665 Pounds; Air Transported)



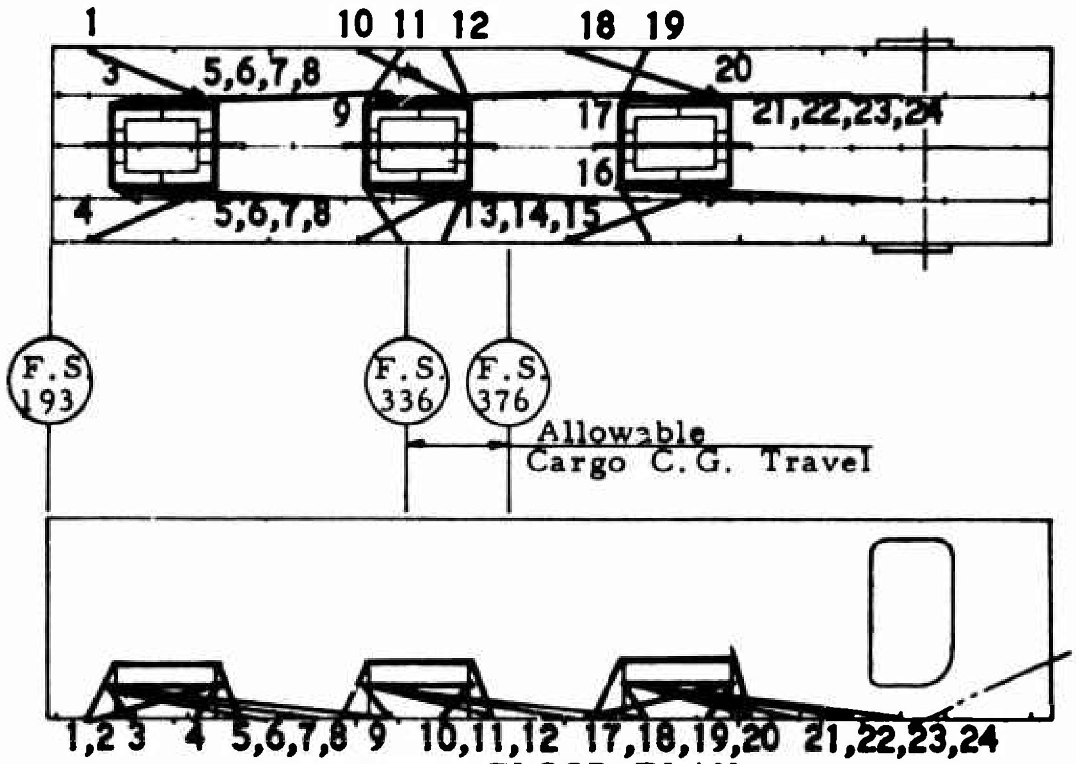
CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 8 Straps **Cargo Wt. 718 lb**	5D 6D 7D 8D	1D	3D 4D	2D
Midship Group Devices Used { 8 Straps **Cargo Wt. 718 lb**	13D 14D 15D 16D	9D	11D 12D	10D
Aft Group Devices Used { 8 Straps **Cargo Wt. 718 lb**	21D 22D 23D 24D	18D	19D 20D	17D

TOTAL OF RESTRAINT DEVICES = 24

D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 41. Restraint of Six 55-Gallon Drums Using Existing Methods With Correct Restraint (Cargo Weight, 2,154 Pounds, Air Transported)

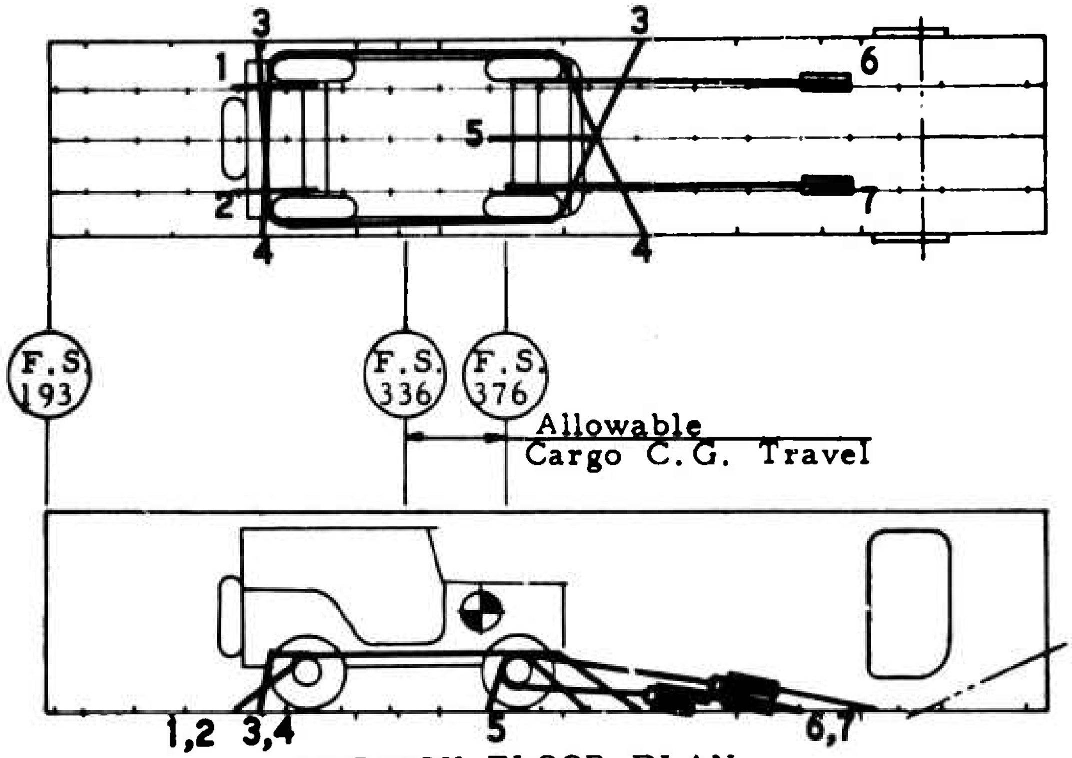


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 8 Straps Cargo Wt. 600 lb	5D 6D 7D 8D	1D	3D 4D	2D
Midship Group Devices Used { 8 Straps Cargo Wt. 600 lb	13D 14D 15D 16D	10D	11D 12D	9D
Aft Group Devices Used { 8 Straps Cargo Wt. 600 lb	21D 22D 23D 24D	18D	19D 20D	17D

TOTAL OF RESTRAINT DEVICES = 24 PLUS
15 PALLETS = 39
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 42. Restraint of Thirty-Six 81-mm Cartridge Boxes Using Existing Methods With Correct Restraint (Cargo Weight, 1,800 Pounds; Air Transported)

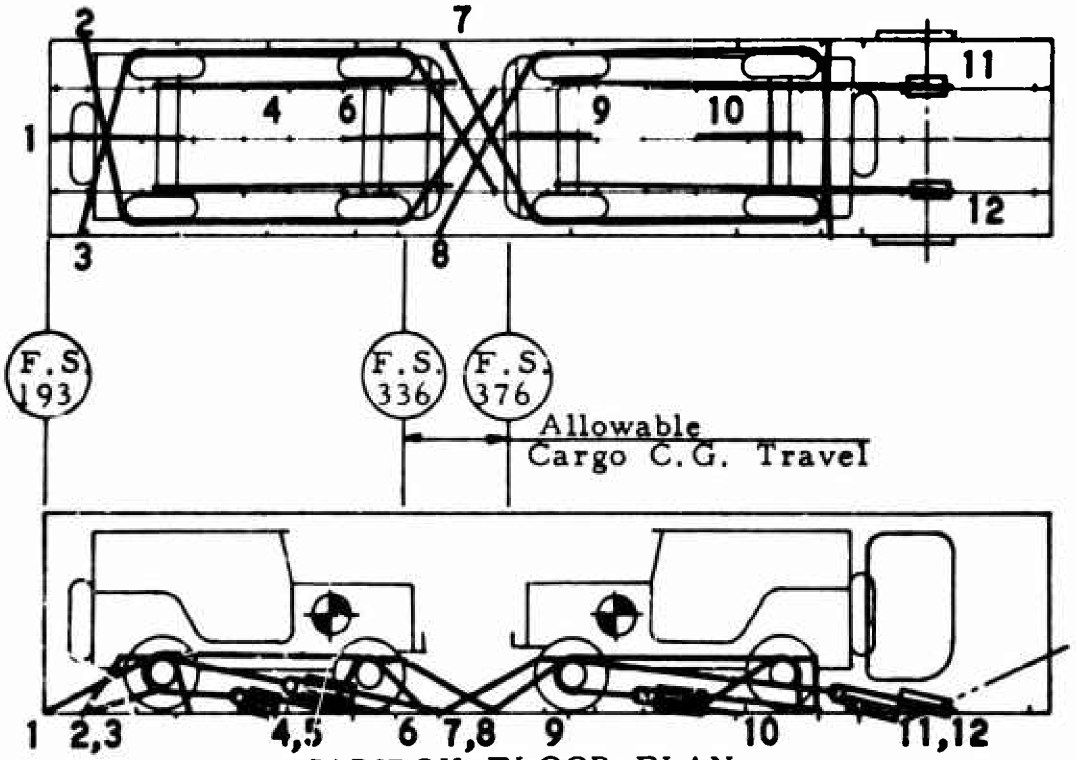


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used {				
Cargo Wt. ______				
Midship Group	6D 7D	1 2	3D 4D	1 2 5D
Devices Used { 7 Straps				
Cargo Wt. 2,665 lb				
Aft Group				
Devices Used {				
Cargo Wt. ______				

TOTAL OF RESTRAINT DEVICES = 7

Figure 43. Restraint of 1/4-Ton Utility Truck Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,665 Pounds; Air Transported)

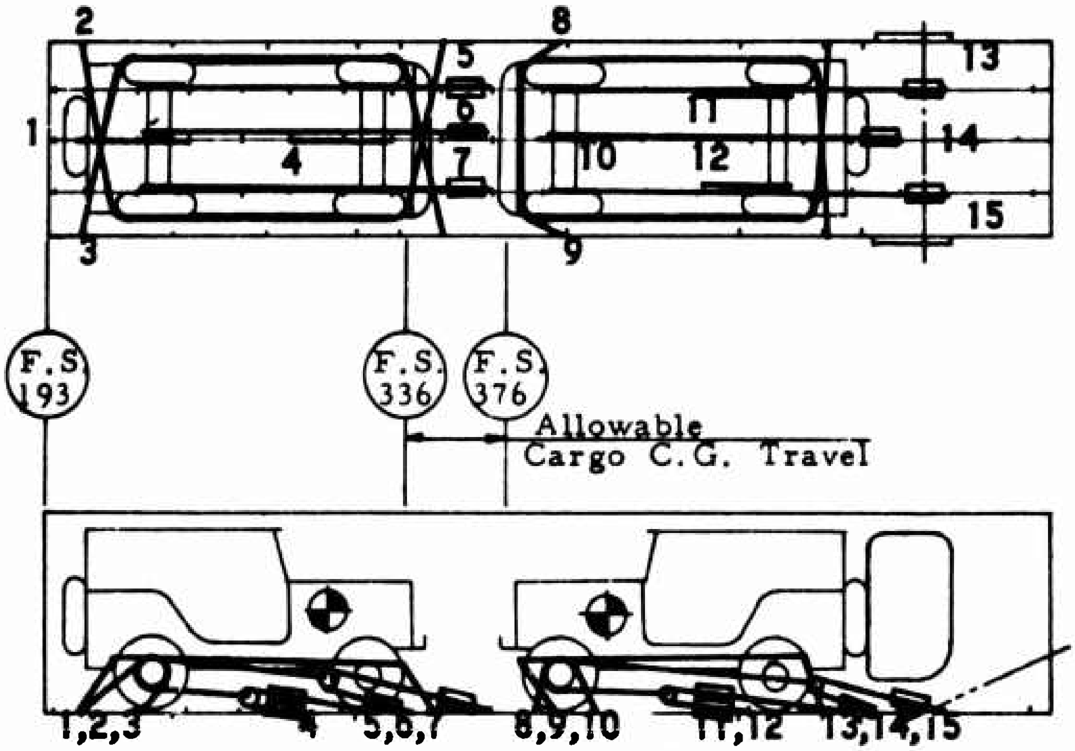


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 6 Straps Cargo Wt. 2,665 lb	4D 5D	1D	2D 3D	1D 6D
Midship Group Devices Used { 6 Straps Cargo Wt. 2,665 lb	11D 12D	9D	7D 8D	9D 10D
Aft Group Devices Used { Cargo Wt. ______				

TOTAL OF RESTRAINT DEVICES = 12
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 44. Restraint of Two 1/4-Ton Utility Trucks Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 5,330 Pounds; Air Transported)

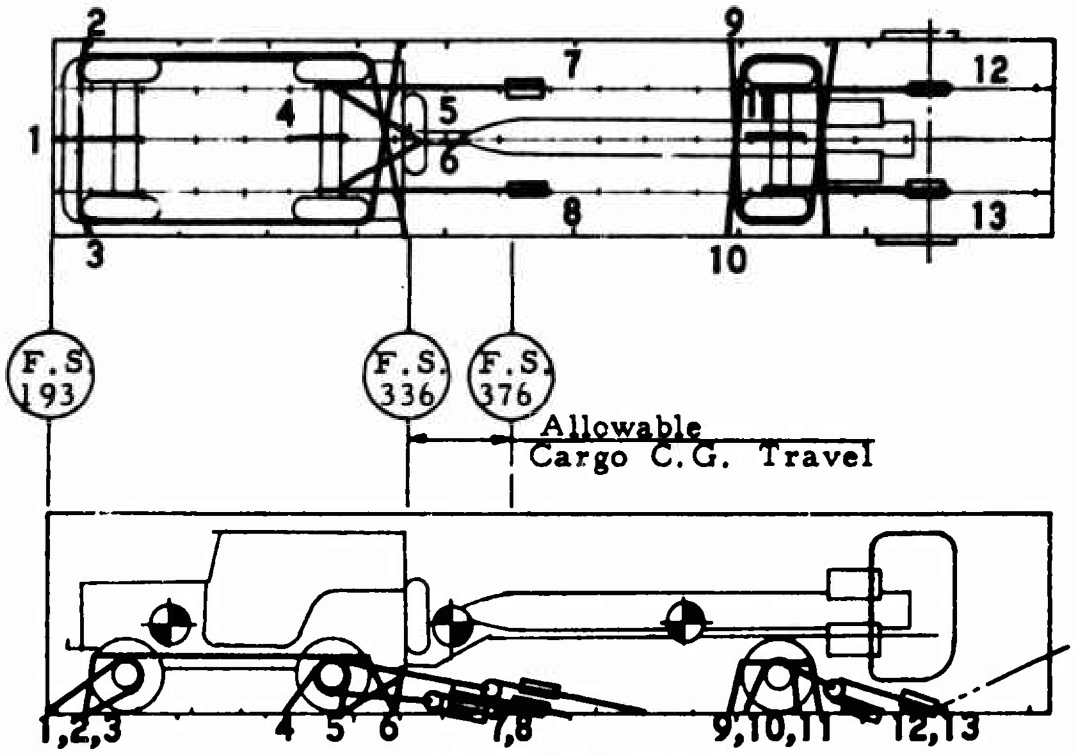


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 7 Straps **Cargo Wt. 3,750 lb**	5D 6D 7D	1D	2D 3D	1D 4
Midship Group Devices Used { 8 Straps **Cargo Wt. 3,750 lb**	13D 14D 15D	11 12	8D 9D	10D
Aft Group Devices Used { Cargo Wt._____				

TOTAL OF RESTRAINT DEVICES = 15
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 45. Restraint of Two 1/4-Ton Utility Trucks Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 7,500 Pounds; Air Transported)

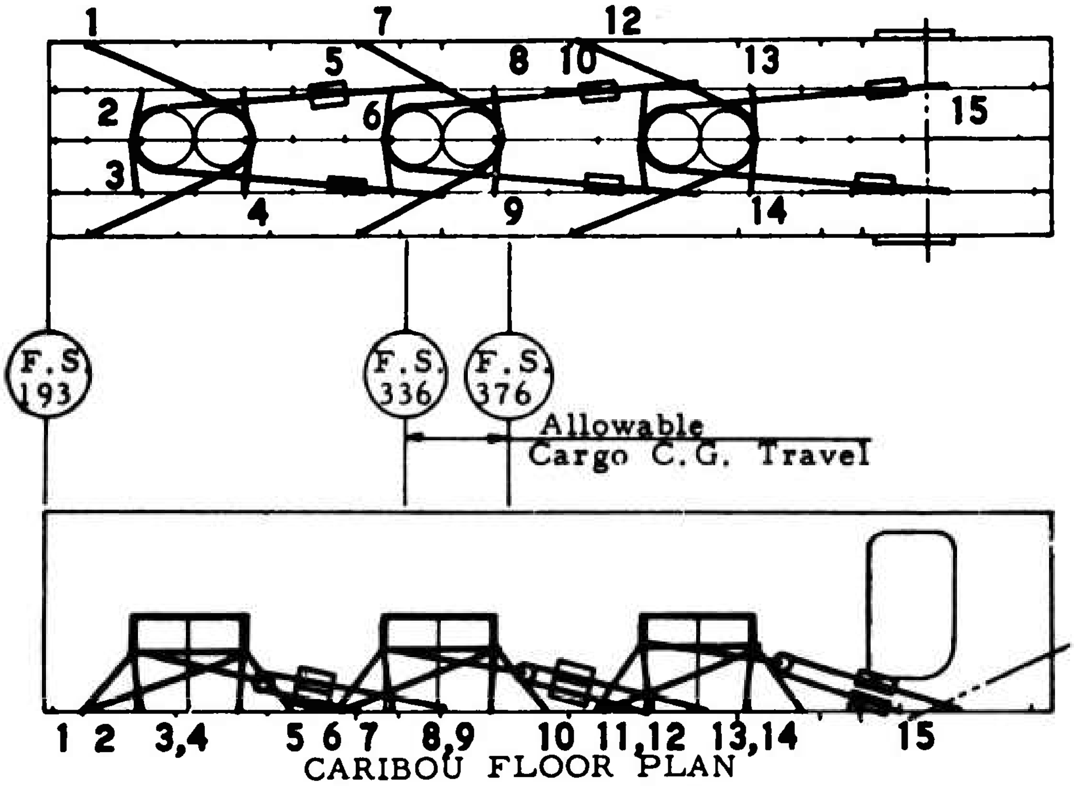


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 6 Straps Cargo Wt. 2,665 lb	7D 8D	1D 4D	2D 3D	1D 4D
Midship Group Devices Used { 7 Straps Cargo Wt. 1,389 lb	12 13	5D	9D 10D	6 11D
Aft Group Devices Used { Cargo Wt.______				

TOTAL OF RESTRAINT DEVICES = 13
D = STRAP DOUBLED FOR 2 × CAPACITY

Figure 46. Restraint of 1/4-Ton Utility Truck With Rocket and Trailer Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 4,054 Pounds; Air Transported)

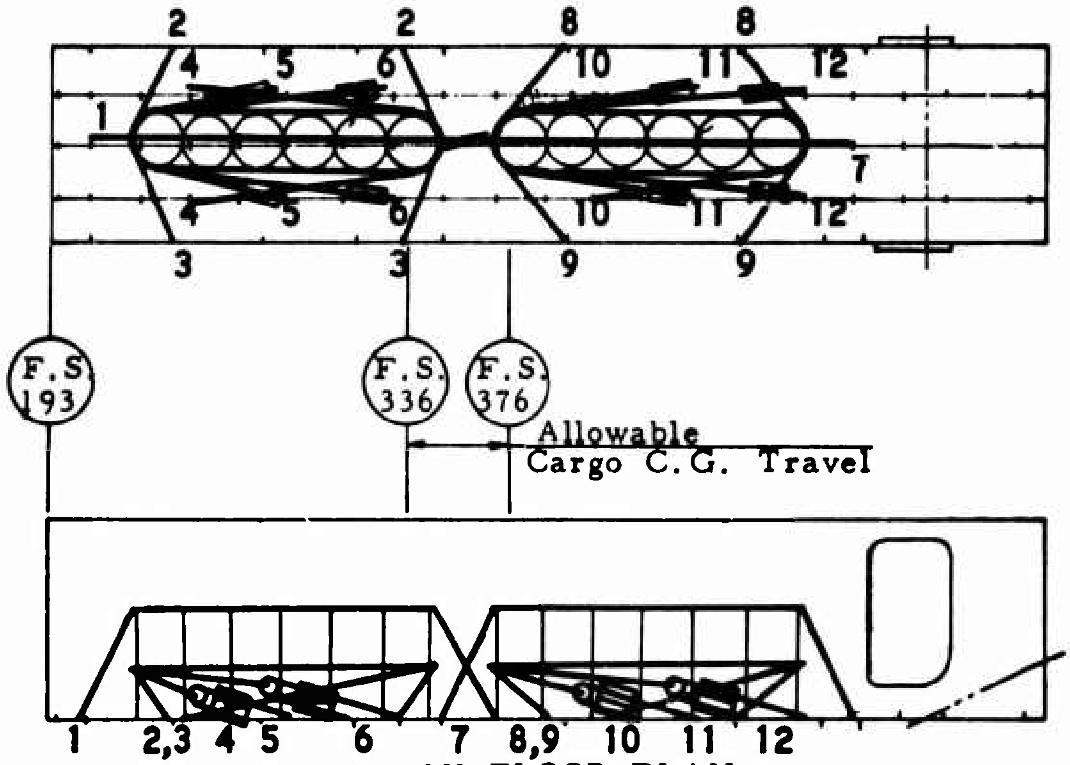


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE					
Direction of Restraint		Forward	Aft	Side	Vertical
Forward Group		5D	1D	3D 4D	2D
Devices Used	5 Straps				
Cargo Wt. 718 lb					
Midship Group		10D	7D	8D 9D	6D
Devices Used	5 Straps				
Cargo Wt. 718 lb					
Aft Group		15D	12D	13D 14D	11D
Devices Used	5 Straps				
Cargo Wt. 718 lb					

TOTAL OF RESTRAINT DEVICES = 15
D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 47. Restraint of Six 55-Gallon Drums Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,154 Pounds; Air Transported)

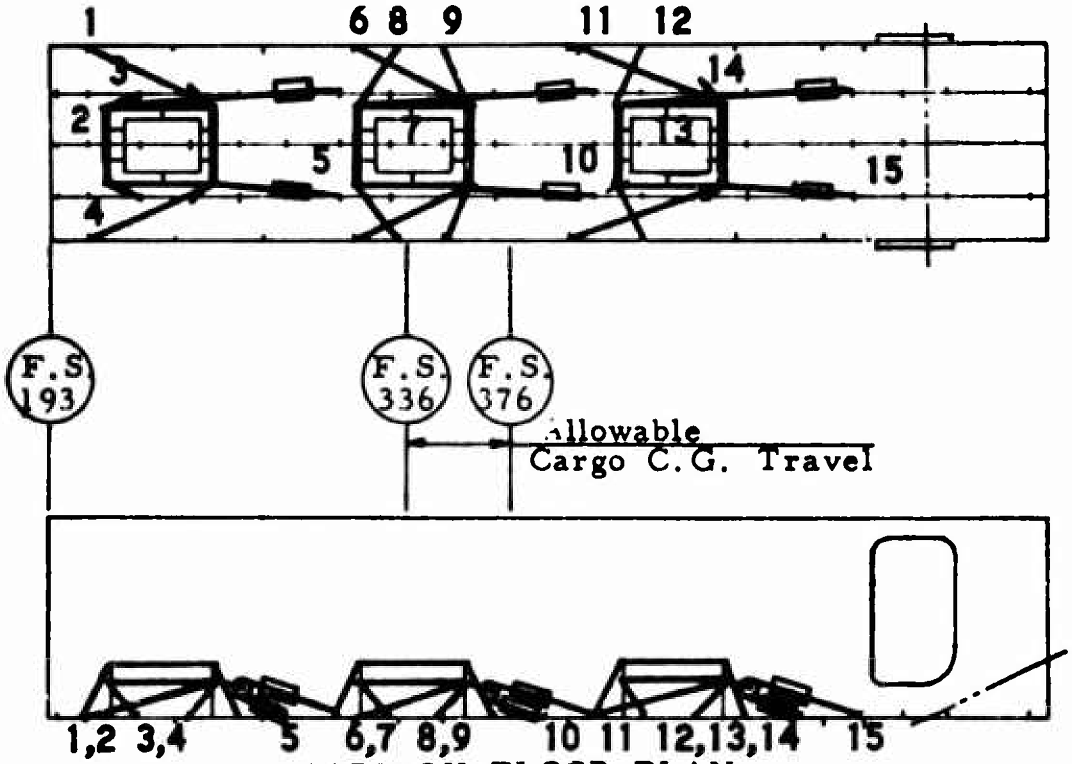


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 6 Straps **Cargo Wt. 2,154 lb**	**5D** **6D**	**4D**	**2D** **3D**	**1D**
Midship Group Devices Used { 6 Straps **Cargo Wt. 2,154 lb**	**11D** **12D**	**10D**	**8D** **9D**	**7D**
Aft Group Devices Used { Cargo Wt. ____				

TOTAL OF RESTRAINT DEVICES = **12**
D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 48. Restraint of Twelve 55-Gallon Drums Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 4,308 Pounds; Air Transported)

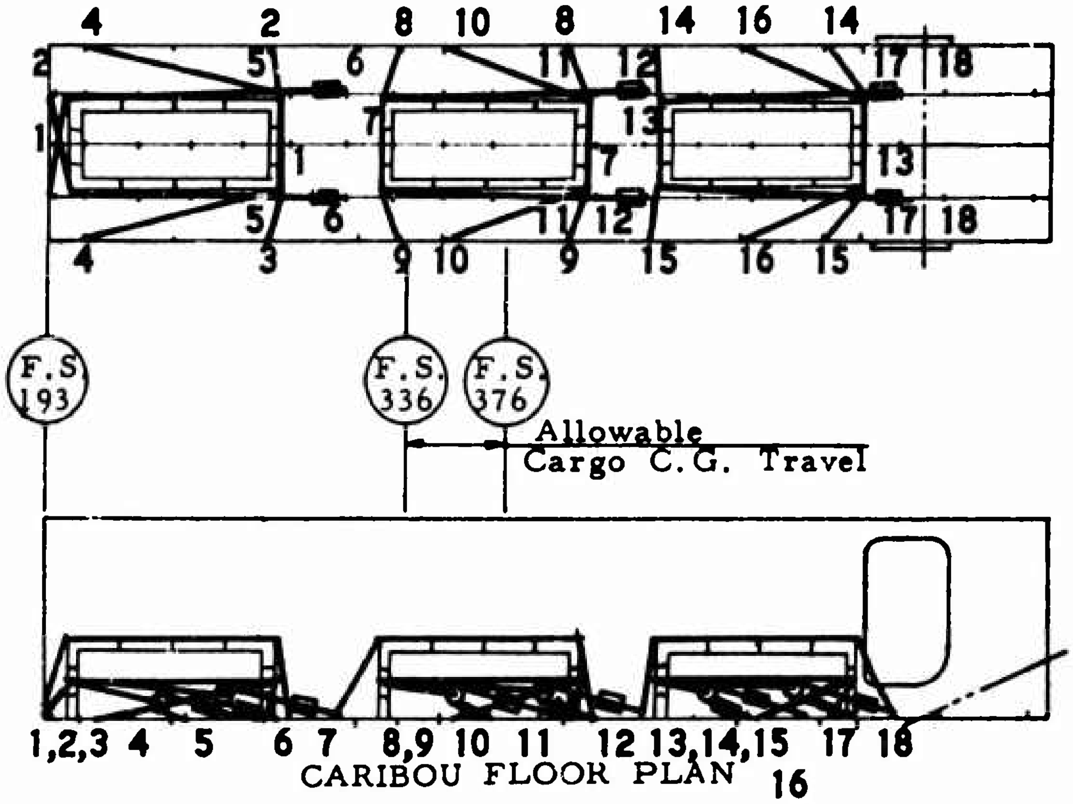


CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 5 Straps Cargo Wt. 600 lb	5D	1D	3D 4D	2D
Midship Group Devices Used { 5 Straps Cargo Wt. 600 lb	10D	6D	8D 9D	7D
Aft Group Devices Used { 5 Straps Cargo Wt. 600 lb	15D	11D	12D 14D	13D

TOTAL OF RESTRAINT DEVICES = 15 PLUS
15 PALLETS = 30
D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 49. Restraint of Thirty-Six 81-mm Cartridge Boxes Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 1,800 Pounds; Air Transported)

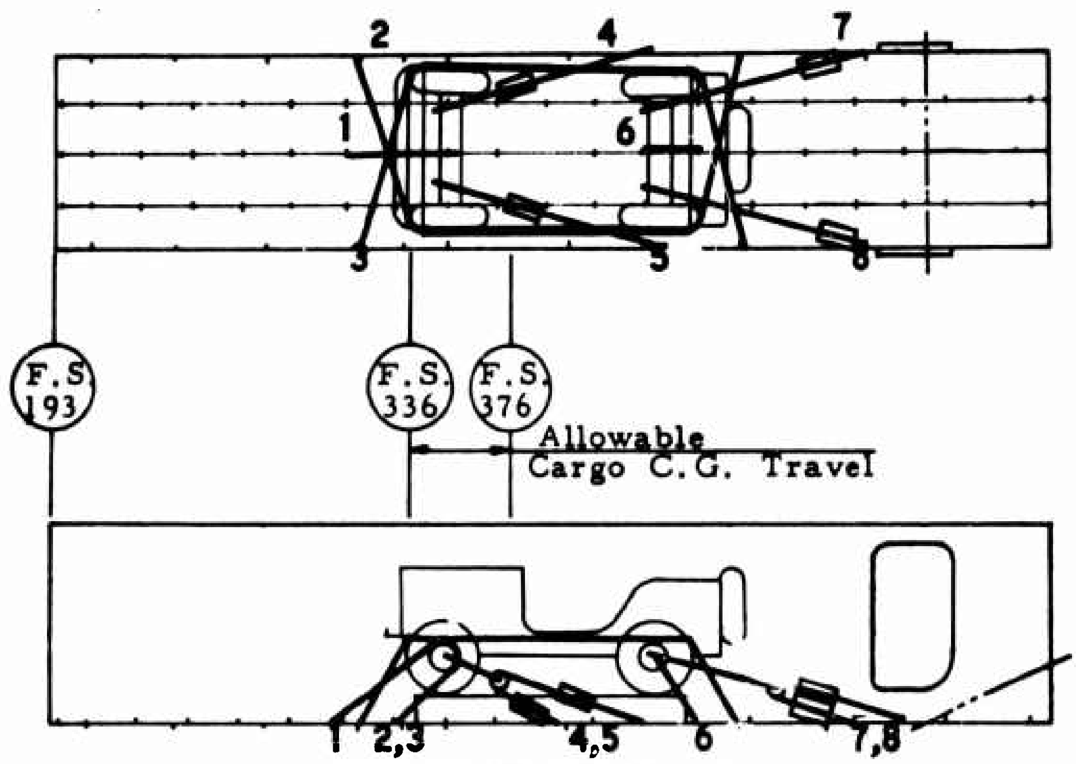


RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group Devices Used { 6 Straps Cargo Wt. 1,800 lb	5D 6D	4D	2D 3D	1D
Midship Group Devices Used { 6 Straps Cargo Wt. 1,800 lb	11D 12D	10D	8D 9D	7D
Aft Group Devices Used { 6 Straps Cargo Wt. 1,800 lb	17D 18D	16D	14D 15D	13D

TOTAL OF RESTRAINT DEVICES = 18

D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 50. Restraint of One Hundred and Eight 81-mm Cartridge Boxes Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 5,400 Pounds; Air Transported)



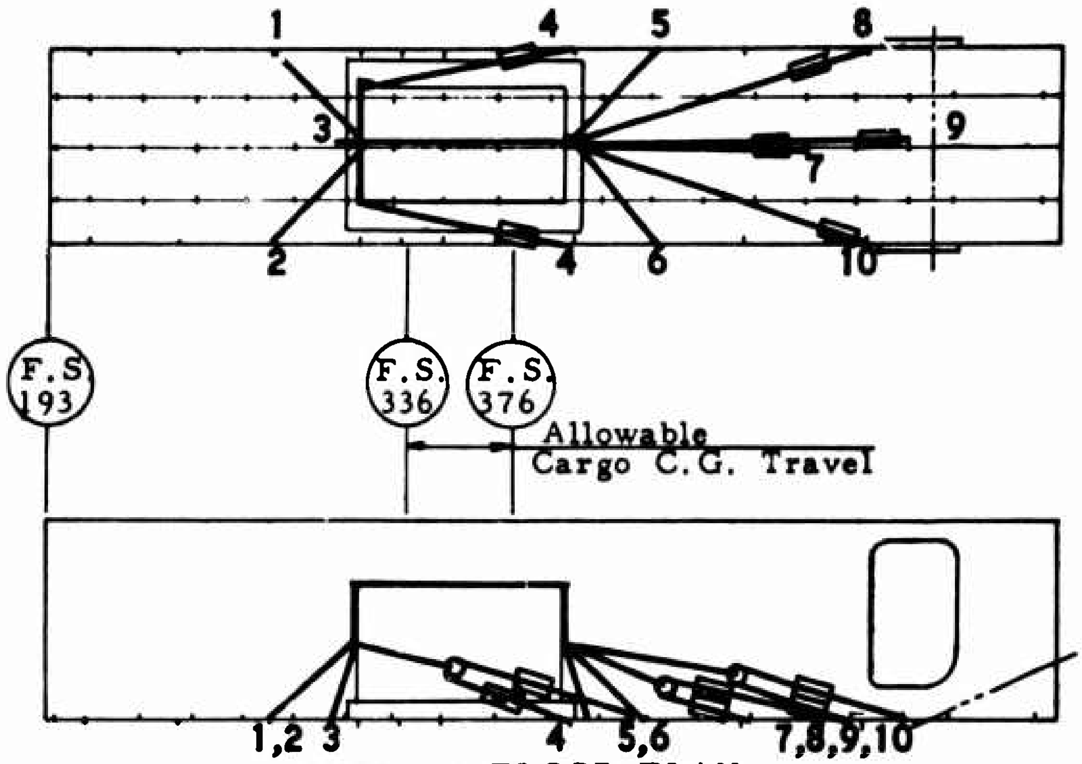
CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt. ____				
Midship Group	4 5 7 8	1D	2D 3D	1D 6
Devices Used { 8 Straps				
Cargo Wt. 2,665 lb				
Aft Group				
Devices Used				
Cargo Wt. ____				

TOTAL OF RESTRAINT DEVICES = 8

D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 51. Restraint of 1/4-Ton Utility Truck With Pallet Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 2,665 Pounds; Air Dropped)



CARIBOU FLOOR PLAN

RESTRAINT DEVICE CODE				
Direction of Restraint	Forward	Aft	Side	Vertical
Forward Group				
Devices Used				
Cargo Wt.____				
Midship Group	4D 7 8 9 10	1 2	1 2 5 6	3
Devices Used { 10 Straps				
Cargo Wt. 3,500 lb				
Aft Group				
Devices Used				
Cargo Wt.____				

TOTAL OF RESTRAINT DEVICES = 10

D = STRAPS DOUBLED FOR 2 × CAPACITY

Figure 52. Restraint of Sealdbin "70" Container With Pallet Using Proposed Strap Device With Energy Absorbers and Pulleys (Cargo Weight, 3,500 Pounds; Air Dropped)

TABL
OPERATIONAL

① Description of Item	② Report Page	③ Cargo Weight (lb)	④ Number of Restraint Devices Used	⑤ Equiv. Cargo Wt. for Devices Used (lb)	⑤ Equiv. Cargo Wt. for Devices Used (lb)
1 of Truck, Utility, 1/4 Ton, 4 × 4, M38A1 and M38A1C - Air Transported.					
1. Existing Army Restraint Method	50	2,665	8	266	266
2. Existing Method with Correct Restraint	55	2,665	21	2,665	2,665
3. Proposed Strap Device w/Load Limiters & Pulleys	58	2,665	7	2,665	2,665
2 of Truck, Utility, 1/4 Ton, 4 × 4, M38A1 and M38A1C - Air Transported.					
1. Existing Army Restraint Method	51	5,330	12	small	small
2. Existing Method with Correct Restraint	**	-	-	-	-
3. Proposed Strap Device w/Load Limiters & Pulleys	59	5,330	12	5,330	5,330
1 each of Truck, Utility, 1/4 Ton, 4 × 4, M38A1 and M38A1C, and Rocket, 318MM-M51 & Trailer XM420 - Air Transported.					
1. Existing Method with Correct Restraint	**	-	-	-	-
2. Proposed Strap Device w/Load Limiters & Pulleys	61	4,054	13	4,054	4,054
Drums - 55 Gallon - Air Transported.					
1. Existing Army Restraint Methods	52	2,513	1	small	small
2. Existing Method with Correct Restraint	56	2,154	24	2,154	2,154
3. Proposed Strap Device w/Load Limiters & Pulleys	63	4,308	12	4,308	4,308
Cartridge Boxes, 81MM, M43A1 - Air Transported.					
1. Existing Method with Correct Restraint	57	1,800	24	1,800	1,800
2. Proposed Strap Device w/Load Limiters & Pulleys	65	5,400	18	5,400	5,400
1 of Truck, Utility, 1/4 Ton, 4 × 4, M38A1 and M38A1C - Air Drop.					
1. Existing Army Restraint Method	53	2,665	8	210	210
2. Existing Method with Correct Restraint	**	-	-	-	-
3. Proposed Strap Device w/Load Limiters & Pulleys	66	2,665	8	2,665	2,665
1 of Sealdbin "70" Container - Air Drop.					
1. Existing Army Restraint Method	54	3,500	6	235	235
2. Existing Method with Correct Restraint	**	-	-	-	-
3. Proposed Strap Device w/Load Limiters & Pulleys	67	3,500	10	3,500	3,500

** Not feasible

A

TABLE I

NAL FEASIBILITY

	(6) Equalizing Factor 5 / 3	(7) Method Effectiveness 6 / 4 × 100	(8) Space Occupied, Cargo & Tie-Down Devices ($in.^2$)	(9) Cargo Space Occupied ($in.^2$)	(10) Space Effectiveness 9 / 8 × 6 × 10	(11) Tie-Down Fittings Unused After Restraint	(12) Floor Fittings Before Restraint	(13) Tie-Down Fitting Effectiveness 11 / 12 × 6 × 10	(14) Number of Unused Devices	(15) Device Effectiveness 14 / 4 × 6 × 10	(16) Est. Restraint Time (man min.)	(17) Est. Time Effectiveness 6 / 16 × 100
6	.100	1.20	14,965	9,100	.610	68	76	.89	18	2.20	7	1.4
5	1.000	4.80	22,557	9,100	4.000	43	76	5.70	4	1.90	20	5.0
5	1.000	14.30	17,520	9,100	5.200	58	76	7.60	18	25.70	7	14.3
11	-	-	-	-	-	-	-	-	-	-	-	-
	-	-	-	-	-	-	-	-	-	-	-	-
0	1.000	8.40	25,185	16,560	6.600	44	76	5.80	13	10.80	16-1/2	6.1
	-	-	-	-	-	-	-	-	-	-	18	-
4	1.000	7.70	25,185	12,000	4.800	46	76	6.00	12	9.30	15	6.7
11	-	-	-	-	-	-	-	-	-	-	-	-
4	.500	2.10	25,185	2,400	.476	22	70	1.60	1	.21	20	2.5
8	1.000	5.90	24,020	4,800	2.000	38	65	5.90	13	10.80	13	7.7
0	.334	1.40	25,185	3,360	.446	22	70	1.10	1	.14	20	1.7
0	1.000	5.60	25,185	6,700	2.700	29	64	4.50	7	3.90	18	5.6
0	.080	1.00	16,510	9,100	.440	26	40	.52	18	1.80	7	1.1
	-	-	-	-	-	-	-	-	-	-	-	-
5	1.000	12.70	18,250	9,100	5.000	25	40	6.30	17	21.30	8	11.8
5	.070	1.17	18,542	13,160	.520	29	40	.51	20	2.34	3	2.3
	-	-	-	-	-	-	-	-	-	-	-	-
0	1.000	10.00	16,980	13,160	7.800	23	40	5.80	15	15.00	4	25.0

TABLE II
PAYLOAD COMPARISON

Description of Items	Cargo Weight (Pounds)	Total Number of Restraint Devices Used		Number of Devices For Forward Restraint		Omni-directional Restraint Comparison	Forward Restraint Comparison
		Existing Method with Correct Restraint (1)	Proposed Strap Device with Load Limiters and Pulleys (2)	Existing Method with Correct Restraint (3)	Proposed Strap Device with Load Limiters and Pulleys (4)	(1)/(2)	(3)/(4)
1 Truck, Utility 1/4 Ton, 4 × 4, M38A1 and M38A1C - Air Transported.	2,665	22	7	17	2	3.1	8.5
Drums, 55-gallon - Air Transported.	2,154	24	15	12	3	1.6	8.0
Cartridge Boxes, 8MM, M43A1 - Air Transported.	1,800	24	15	12	3	1.6	8.0

COST AND WEIGHT OF EXISTING AND PROPOSED RESTRAINT SYSTEMS

Listed in Table III are the actual and estimated cost and weights of the existing restraint devices and strap device with energy absorbers and pulleys. All items listed are individual parts, and the costs are based on production items.

TABLE III
COST AND WEIGHT OF EXISTING AND PROPOSED RESTRAINT SYSTEMS

No. Required per Aircraft	Item	Cost ($) per Item	Weight (lb) per Item
20	Existing web tie-down device MC-1	8.50*	3.5*
6	Existing chain tie-down device MB-1	23.50*	11.0*
6	Proposed D .cron strap device	20.00†	3.0†
24	Proposed self-storing, wire-bending energy absorber (for cost breakdown see Appendix V)	35.00†	2.3*
12	Proposed pulley assembly	2.50†	2.5†

*Actual
†Estimated on production of 2,000 energy absorbers, 500 straps, and 1,000 pulleys.

ANALYSIS

The most critical shortcoming of the existing Army restraint technique is due to insufficient restraint for the considered cargo weight. In the cases studied, Figures 36 through 39, it was found that the existing restraint method resulted in a range of little or no restraint to 10 percent of its required restraint. See column 6, Table I, based on the pulse duration envelope criteria given on page 19. This is primarily due to insufficient restraint in the forward direction. A particular case, shown in Figure 36, delineates two vehicles restrained by mixing chain and web devices. This resulted in improper restraint in practically all the restraint directions. Because of the large differences in the elongation percentages, the chain devices, for the considered load factors, are loaded to their rated capacity, while the web devices are loaded to an estimated 6 percent of their rated capacity. See the section entitled "Elasticity Problems", page 14. Combining these devices for restraint in a given direction results in extreme degradation of the restraint system. In addition, chain devices are relatively inelastic and, therefore, are responsive to high-frequency oscillations that may occur when the input acceleration pulse is imparted to the aircraft. This is important to keep in mind because the crew is not afforded any protection. Of all the cargo delineated in Figures 35 through 39, the 55-gallon drums as restrained by the existing method seem to have little, if any, restraint except for the vertical direction.

For existing methods with correct restraint, the tie-down devices shou l be located so that a minimum of devices will properly restrain the cargo in all the pertinent restraint directions. Each device will then be capable of some restraint in the forward, aft, vertical, and lateral directions. See the section entitled "Optimum Loading Condition", page 17. In a majority of cases, it is impossible to achieve these angles because of cargo shape, mass distribution, availability of cargo attachment locations, and distribution of aircraft floor fittings. For practical purposes, it is necessary to consider each piece of cargo separately.

The basic restraint problem lies in the forward direction. This is evident when inspecting Figure 40, delineating an Army jeep with correct restraint. Predicated on the pulse duration envelope criteria, seventeen forward restraints are required, versus four for the other restraint directions. Because a large number of forward restraints are required, the distribution of aircraft floor fittings is such that variant restraints of sizeable lengths will be utilized. Therefore, it is apparent that the elasticity effects of variant lengths have been considered (see section entitled "Elasticity Problems") and account for 41-percent additional forward restraint devices. The Army jeep covers about 36 percent of the floor area; with the addition of tie-down devices, it covers 90 percent. In other words, the vehicle and the tie-down devices together cover 2.5 times more area than the vehicle alone. It is obvious that the devices utilize valuable cargo space when cargo is completely restrained, which is mainly important in both the forward and aft directions.

The weights that are correctly restrained by the existing method are 1,800, 2,154, and 2,665 pounds; this ranges from 24 to 35 percent of the maximum cargo payload, which is 7,500 pounds for the Caribou aircraft. No additional weight can be added and restrained securely because of the limitations of floor tie-down fittings.

If air transported, the minimum weight cargo restrained is cartridge boxes (see Figure 42). A total of thirty-six boxes is capable of being correctly restrained in groups of twelve. Pallets or plywood platforms are used against the sides and tops of the boxes to reduce the number of tie-down devices to augment their effectiveness. Elimination of the plywood pieces would require that the total number of boxes be reduced in order to obtain correct restraint. Any group is sufficiently separated from another to permit the correct application of restraint by tie-down devices. To utilize the devices at optimum angles, the cargo and tie-down devices covered about seven and a half times as much floor area as the cargo. See columns 8 and 9 of Table I. This shows how inefficient the existing method can be in restraining cargo.

The time to rig and derig cargo, shown in Table I, is estimated at 20 minutes. This restraining time required can be attributed to crawling under the vehicle to attach the tie-down devices to the axles and to passing the devices around the cartridge boxes and 55-gallon drums, in addition to attaching two devices together to make a longer device. Also, for proper restraint, most of the available devices are used.

Vehicle-type cargo rigged f... air-drop operations is incapable of correct restraint with the existing m...d. For air-drop operations, cargo on pallets and roller conveyors covers up about 47 percent of the avai. ole floor fittings. The number of required restraints is in excess of the number of floor fittings available in both the forward and aft directions of the cargo. For the cartridge boxes and 55-gallon drums, restraint for air-drop operations is feasible at a total weight much less than those shown for air-transported operations.

From the above discussion, the following criteria should be considered in designing a cargo restraint system:

1. Avoid differential elongation between chains and straps.

2. Design a system which negates the problems inherent in achieving total restraint.

3. Completely restrain maximum cargo weights.

4. React the forward g load with a minimum of restraint devices.

5. Permit additional floor space and cargo weight utilization.

6. Reduce the number and types of tie-down devices for cargo restraint.

7. Provide rapid and simple installation along with a quick method of derigging cargo for air-transport and air-drop operations.

8. Minimize the number of fittings used in the installation, and do not modify the airframe in any case.

9. Keep design concept weight to a minimum, and use a rugged construction capable of reacting cargo loads based on the pulse duration envelope criteria.

10. Consider the possible cargo C.G. location.

11. Have design universally applicable to all Army aircraft.

12. Design for sturdiness to withstand abuse inherent in cargo handling environment.

The strap device with energy absorbers and pulleys is the concept that appears to be most capable of meeting the requirements of the above criteria. See page 40. The recommended energy absorbers are the self-storing, wire-bending type.

An optimum designed self-storing, wire-bending energy absorber is one that will utilize to the fullest extent the variables associated in a plausible combination of wires and platen. The experimental data in Tables IV through IX, Appendix I, show that the optimum mean wire pull load can be attained when the minimum hole diameter and hole spacing on the platen are utilized. This assumes that all other variables are the same. This is a result of maximum bending of the wire after the wire is threaded through the platen. The greater the bend, the greater the force required to pull the wire through the platen holes. It can also be shown that by increasing the platen hole diameter and/or hole spacing, the mean pull load will decrease. By increasing the wire diameter and platen thickness, the load will increase; the converse is also true.

For the data presented, the holes used in the platen were either broken at the edges or chamfered, or the edges were polished to a rounded contour. It can be seen from tests numbers 1 and 2 (Table VII) that the mean pull load will be of less intensity if platens with chamfered holes are used rather than platens with broken hole edges. Also, tests numbers 90 and 127 show that the pull load intensity will be further decreased if the platen considered has hole edges which are polished to a rounded contour instead of chamfered. This is attributed to the fact that the contour of the hole edge becomes wider when polished than if just chamfered. Therefore, the bend radius of the wire between holes is increased; this results in a decreased pull load during operation. However, tests numbers 2 and 126 show that the mean pull load developed with polished holes was greater than the mean pull load utilizing a platen with chamfered holes. This contradiction to the above statement can be attributed to the fact that less hole edge polishing was utilized for this specimen. It is then realized that, by close scrutiny of polished holes in platens, a desired tolerance of workable loads can be achieved during operation. This is also true for the load limiters with the other hole conditions.

In general, as the wire diameter increases, the fluctuation from minimum to maximum pull load in the wire will decrease. The pull load data listed in Table VII, for 0.105-inch-diameter wire, ranges on the average of about 11.0 percent in load fluctuation from minimum to maximum for a given test. In comparison, the pull load data listed in Table VII, for 0.092-inch-diameter wire, ranges on the average of about 15.5 percent in load fluctuation. However, from these tables, it can be shown that there is less consistency in the load fluctuation for the smaller diameter wire.

For compact storage purposes, it is felt that the 0.105-inch-diameter wire is the maximum feasible size capable of being used in minimum quantities. In order to provide a factor of safety, the mean pull load of a unit should be achieved at no more than 75 percent of the ultimate strength of the particular wire lot; the maximum unit fluctuation load should be no greater than 80 percent (see Table X). This is important because the failure of the unit is a small percentage less than the wire ultimate strength, as demonstrated in the column Peak Loads in the tables for those tests in which failure occurred.

As can be seen from the tables, an initial peak pull load was obtained during each test run. A method of eliminating or reducing this peak load sufficiently was found, with favorable results. Elimination of the peak intensity was accomplished by initially pulling the threaded wire for a stroke of 1 inch and then by backing off to zero load, thus establishing a peak and a mean pull load. The test was restarted, and the same mean load was attained without the effect of the peak load. Tests 68 and 69 were conducted as such.

It was ascertained during the testing phase that using one single wire, threaded through both sides and looped around the end of the platen (see page 35 and Figure 26), establishes inherent problems that using two single wires would eliminate (see page 36). The test runs from 1 through 60 were conducted with the single wire threaded through both sides and looped around the end of the platen, and two single wires as threaded were utilized for the remainder of the test runs. Threading a single wire as such has caused inadvertent local bends and kinks in the wire, which has resulted in failure. A prime example of this is shown by a comparison of tests numbers 41 and 81, and 27 and 56 (Tables V and VI). It is felt that some of the failures shown in Table IV would not have resulted under a more favorable threading condition. Threading two single wires seems to overcome these deficiencies. It is also anticipated that it may require an appreciable bending of the wire for storage purposes. This could appreciably affect the mean pull load during operation. The selection of variables will have to be definitively chosen when the storage area is designed.

The variables that seem to be pertinent to an optimum load limiter design are 0.105-inch-diameter annealed stainless-steel-type wire and a 1/4-inch 2024-T3 aluminum platen with a 1/4-inch hole diameter and 3/4-inch hole spacing.

The self-storing units are capable of storing the required wire in the most compact area possible with the maximum feasible wire size. These units are lightweight and compact, consist of no moving parts, and are conducive to economical production.

The strap device with load limiters and pulleys is the best method of cargo restraint in both air-transport and air-drop operations. Also, this concept provides total forward load protection to the aircraft crew even when the aircraft is loaded to its full cargo capacity. The method effectiveness column of Table I, which is the measure of the relative restraint capabilities of the three cargo restraint techniques, indicates that the proposed strap device with energy absorbers and pulleys is more effective than the existing methods for the cargo considered. This can be attributed to the fact that the total number of restraint devices used for the proposed concept was less than for the existing method with correct restraint. The total restraint devices used with the existing Army restraint technique are not necessarily less than the proposed design concept, but the existing method did not restrain the cargo properly. With the use of an equalizing factor, column 6, a comparison of the results of the existing restraint method was made with those achieved by the other restraint methods, whereby the results of the existing method effectiveness were far below those of the proposed concept.

The strap devices with energy absorbers and pulleys are to be used for forward restraint only, while the existing devices are used in the other restraint directions (see Figures 43 through 52). As can be seen from Table II, columns 3 and 4 indicate that a reduction of about 9 to 15 forward restraint devices can be realized when using the design concept instead of the correct restraint method for cargo weights of 1,800 to 2,665 pounds. If heavier weight cargo were capable of correct restraint with existing methods, then a more definitive trend would be evident; a large reduction in the number of forward restraint devices required by the design concept would be shown as the cargo payload is increased. From Table II, the omnidirectional restraint comparison indicates that the reduction of total restraint devices resulting from the use of the proposed design concept instead of the existing method with correct restraint is less than the forward restraint comparison. This can be attributed to the fact that while the forward restraint devices required are quite different in number between the two methods, the restraint devices required in the other directions are relatively the same. The total number of restraint devices is of a relatively large magnitude in relation to forward devices; therefore, a large percentage increase in forward restraint devices produces a smaller percentage increase in total number of restraint devices. For the proposed design concept, the restraint devices used to restrain the cargo, shown in Figures 43 through 52, range from 7 to 18, which is from 28 to 72 percent of the total available restraint devices. The cartridge boxes weighing 5,400 pounds require the maximum total restraints, while the single Army jeep, air transported and air dropped, requires the minimum. For the remaining cargo, an average of about 57 percent of the total available devices is used for restraint. This includes the restraining of two gross weight jeeps (total weight, 7,500 pounds) air transported. The maximum number of forward restraint devices required for the gross weight jeeps is six, which is 24 percent of the total available devices. On the average, the number of forward restraint devices required for complete restraint is about 16.5 percent of available devices. The available devices refer to both existing devices and proposed devices, totaling twenty-five. However, it is evident from the foregoing discussion that six proposed design concept devices and eighteen Army devices are sufficient to restrain cargo of maximum payload in the Caribou aircraft.

Table I, column 9, shows that for the strap device with energy absorbers and pulleys, the cargo listed covers a floor area in the range of 19 to 66 percent of the total floor area. The higher percentage of the range is due to restraining two Army jeeps, which utilize both length and width in relation to the cargo compartment dimensions. From column 8, the percentage of floor area covered by the cargoes and their restraining devices ranges from about 68 percent to 100 percent. On the average, the cargo and tie-down devices listed cover 80 percent of the aircraft floor area as compared to 43 percent for the cargo alone. In other words, the cargo and tie-down devices together cover 1.86 times more area than the cargo alone. This indicates that the devices require a large portion of the cargo space when cargo is completely restrained. This can be attributed to the distribution of aircraft floor fittings in relation to aircraft size, shape, and mass.

The strap device with load limiters and pulleys can be utilized to restrain vehicles and Sealdbin "70" containers correctly for air-drop, which is not possible with the existing method. See Figures 51 and 52. Only a single Army vehicle can be restrained properly for air-dropped cargo, where two gross weight Army jeeps (7,500 pounds total) can be restrained adequately for air-transported cargo. (see Figure 45).

A comparison of the estimated time to rig and derig cargo depicted in column 16, Table I, shows that the proposed design concept combined with existing restraints takes less time than those cases capable of restraint by the existing method with correct restraint. As shown in column 17, Estimated Time Effectiveness, the proposed concept is as much as 3.3 times as effective as the existing method with correct restraint, and 10 times as effective as the existing Army restraint method. For the latter method, this is not evident at a first glance at the estimated restraint times listed. However, an equalizing factor, column 6, makes possible the comparison of the results of the proposed design concept and the existing Army method. The results of the existing method effectiveness are far below those of the strap device with energy absorbers and pulleys.

If comparable cargo is considered, the proposed design concept is much more effective in relation to restraint capabilities, space available, tie-down fittings utilized, tie-down devices used, and allotted restraint time to rig and derig cargo than are the existing Army method and the existing method with correct restraint. Columns 7, 10, 13, 15, and 17 confirm the effectiveness of the proposed design concept. A distinct, but not so obvious, advantage of the new concept is its application to the cargo. For the cargo analyzed in this report, few devices are needed for forward restraint. Predicated on this, the elasticity effects of various length devices (see section entitled "Elasticity Problems") can be almost eliminated. This is accomplished by aligning the required restraint devices with approximately equal lengths. Small disparity in device lengths will not have any appreciable effect on device strength capacity. When preparing Figures 43 through 52, the required forward restraint devices were carefully aligned. Only in the cases of the 55-gallon drums and cartridge boxes was aligning of the forward restraints found to be difficult. However, for each group of cargo, two restraint devices of equal length are needed, and their length disparity, as shown, is not sufficient to warrant additional devices.

For correct restraint, the existing MC-1 devices would require, at a minimum, 5.3 times as many restraints as the proposed restraint system. Considering the variant length effects associated with multi-strap installation, the number of required MC-1 devices could be further increased by an estimated 125 percent or about 6.5 times as many as the new system. The weight of each new restraint is estimated to be about 6.5 times each MC-1 device (see Table III). On a comparative basis, no considerable weight difference is realized between the new and the existing methods

For total restraint, utilizing six new devices and fourteen existing devices, the weight per aircraft would be 183 pounds. The total weight of existing restraints, webbing, and chains, carried as part of the aircraft equipment, is 136 pounds. The difference in total restraint system weight between the new and the existing methods is considered small.

Army equipment which did not exceed 7,500 pounds and which could be physically placed in the Caribou aircraft was considered. If each piece of Army equipment is considered individually, it is estimated that about 80 percent of this cargo can be properly restrained. Any combination of the considered equipment would require a complete analysis in order to prove proper restraint. It is not feasible to consider all the possible combinations because of the magnitude of work involved.

CONCLUSIONS

It is concluded that:

1. The mixing of chain (MB-1) and nylon (MC-1) tie-down devices to augment the restraint capability for a given restraint direction results in serious degradation of the restraint system. In addition, the restraint system capability is degraded with the use of various length MC-1 straps in a given restraint direction (see page 14). The MB-1 chain tie-down devices, which have relatively high stiffness characteristics, are to be considered unfit as restraint devices because they may become responsive to [illegible] oscillations that occur during a crash.

2. Current restraint techniques delineated in technical manuals do not display or describe proper restraint application (see page 5).

3. In fixed- and rotary-wing aircraft, the forward restraint direction is the most critical in determining restraint requirements.

4. The use of existing tie-down devices with the proposed survivable crash restraint criterion could seriously affect the allowable cargo-carrying capability of fixed and rotary wing aircraft (see page 8).

5. A high-strength, low-elongation strap device in series with load limiters will significantly reduce the problems associated with restraining cargo under survivable crash conditions.

6. Of the two design concepts studied in this report, the load-limiting, wire-bending-type energy absorber offers the most advantages.

7. A load-limiting, wire-bending-type energy absorber with self-storing features can be incorporated into a restraint system with a minimum of weight (approximately 2 pounds) and complexity.

8. If the proposed survivable crash criterion is adopted, the proposed restraint system incorporating a high-strength, low-elongation strap device with load limiters and pulleys as opposed to the existing MC-1 devices against the same criterion will provide the following:

a. The capability of restraining maximum payloads in all Army aircraft and providing full load protection to the aircraft crew with no appreciable increase in system weight (see Figure 45 and page 73).

b. A considerable reduction in the number of restraints required to restrain typical cargo. As an example, 90 percent fewer forward restraints are required to restrain an Army jeep (see Figures 40 and 43).

c. The capability of restraining heavier individual cargoes (see Figures 40 through 50).

d. Time saved in cargo rigging and derigging averaging about 32 percent (see Table I, column 16).

9. For future system design criteria, the design objectives outlined on pages 20 and 21 of this report should be given serious consideration.

10. A total of six strap devices with load limiters and pulleys to be carried per aircraft would cost an estimated $990.00 (see Table III). The cost of the strap part of the proposed concept can be depreciated by the cost of the existing web device when counted as a replacement item. In addition, a reduction of about six existing webbing devices and the elimination of the chain-type devices per aircraft are feasible with a consequent reduction in replacement cost.

11. For the purpose of dynamic testing, a decelerator device taking the form of a deformed-tube-type shock strut should be considered (see Appendix IV).

BIBLIOGRAPHY

1. REPORTS:

a. Avery, J. P., Dr., Cargo Restraint Concepts For Crash Resistance, USAAVLABS Technical Report 65-30, Aviation Safety Engineering and Research, Phoenix, Arizona, June 1965.

b. Buckson, W. C., Schlegel, W. R., and Smith, I. R., Design, Development, and Evaluation of a Crash-Resistant Fuel System Installation, Preliminary, Report number unassigned, Federal Aviation Agency, Washington, D. C., August 1964.

c. Ringleb, F. O., Cable Dynamics, NAEF-ENG-6169, U.S. Naval Air Material Center, Philadelphia, Pennsylvania, December, 1956.

d. Spindler, J. P., Four-Engine Jet Car Brake Tests, N-313, All American Engineering Company, Wilmington, Delaware, May 1957.

e. Wysor, R. B., Development of the Type XMB-1 Fighter Arresting Barrier, WADC Technical Report 58-654, Wright Air Development Division, Wright-Patterson Air Force Base, Ohio, February 1960.

f. Wysor, R. B., and Nissley, W. J., Jr., Tests of the Type XMB-1 Aircraft Arresting Energy Absorber, WADC Technical Report 58-258, Wright Air Development Division, Wright-Patterson Air Force Base, Ohio, February 1960.

g. Lipson, S., Cellular Aluminum for Use in Energy Dissipation Systems, NASA-93, National Aeronautics and Space Administration, Washington, D. C., September 1964.

h. Bixby, H. W., Development of a Paperboard Honeycomb Decelerator for Use with Large Platforms in Aerial Delivery Systems, WADC Technical Report 59-776, Wright Air Development Division, Wright-Patterson Air Force Base, Ohio, May 1959.

i. Parker, R. V., Shock Strut Air Drop Kit Feasibility Demonstration with M-37 3/4-Ton Truck, AF-212, All American Engineering Company, Wilmington, Delaware, January 1965.

2. MANUALS:

a. RM101-10; Staff Officers' Field Manual; Organizational, Technical and Logistical Data; Part 1 - Unclassified Data; October, 1961.

b. TM1-1U-1A-1; Flight Manual, U.S. Army Series U-1A Aircraft; January, 1960.

c. TM1-1U-1A-9; Cargo Loading, U.S. Army Series U-1A Aircraft; September, 1958.

d. TM9-500; Ordnance Corps Equipment Data Sheets; September, 1962.

e. TM10-500-6; Air Delivery of Supplies and Equipment, L-19E (Bird Dog) Army Aircraft; July, 1960.

f. TM10-500-8; Airdrop of Supplies and Equipment, Army Helicopters (UH-1A and CH-21C); March, 1963.

g. TM10-500-9; Airdrop of Supplies and Equipment, U-1A (Otter) Army Airplane, Draft Copy; June 1962.

h. ST10-501-32; Special Test-Draft; Airdrop of Supplies and Equipment, CV-2B (Caribou) U.S. Army Aircraft; February, 1963.

i. TM55-1520-10; Operator's and Crew Members' Instructions, Army Model H-34A, H-34C Helicopters (Sikorsky); February, 1962.

j. TM55-1510-206-10; Operator's Manual AC-1 Aircraft; June, 1962.

k. TM55-1520-203-10; Operator's and Crew Members' Instructions, Army Model H-37A and H-37B Helicopters (Sikorsky); August, 1961.

l. TM55-1520-205-10; Operator's Manual, H-21 Helicopter; August, 1961.

m. TM55-1520-205-20; Organizational Maintenance Manual, H-21C Helicopter; October, 1961.

n. TM55-1520-208-10; Operator's Manual, HU-1B Helicopter; September, 1961.

o. TM55-1520-209-10; Excerpt from Operator's Manual, Army Model CH-47A Helicopter, Chapter 13, Aircraft Loading.

p. TM57-210; Air Movement of Troops and Equipment; October, 1960.

3. MISCELLANEOUS:

a. AEROG 5.1.G.1, Issue 1; Caribou II Model Specifications, Appendix IV, Conveyor Rail Restraint Cargo Handling System, de Havilland Aircraft of Canada, Ltd., Toronto, Canada, September 1962.

DISTRIBUTION

US Army Materiel Command	4
US Army Aviation Materiel Command	6
US Army Forces Southern Command	1
Chief of R&D, DA	1
US Army Aviation Materiel Laboratories	20
US Army R&D Group (Europe)	2
US Army Human Engineering Laboratories	1
US Army Test and Evaluation Command	1
US Army Medical R&D Command	1
US Army Combat Developments Command, Fort Belvoir	2
US Army Combat Developments Command Transportation Agency	1
US Army Aviation School	1
US Army Infantry Center	2
US Army Aviation Maintenance Center	2
US Army Aviation Test Board	2
US Army Transportation Engineering Agency	1
US Natick Laboratories	1
US Army Electronics Command	2
Air Force Flight Test Center, Edwards AFB	2
Systems Engineering Group (RTD), Wright-Patterson AFB	2
US Directorate of Flight Safety Research, Norton AFB	1
Naval Air Systems Command	3
Chief of Naval Research	2
Naval Supply Systems Command	1
US Naval Supply R&D Facility	1
Bureau of Medicine and Surgery, DN	4
Aero Space Medical Laboratory, Wright-Patterson AFB	1
US Naval Air Station, Norfolk	1
Marine Corps Liaison Officer, US Army Transportation School	1
Testing and Development Division,US Coast Guard	1
NASA Scientific and Technical Information Facility	2
NAFEC Library (FAA)	2
US Army Aviation Human Research Unit	2
US Army Board for Aviation Accident Research	1
Bureau of Safety, Civil Aeronautics Board	2
US Naval Aviation Safety Center, Norfolk	1
Federal Aviation Agency, Washington, D. C.	1
CAMI Library, FAA	1
Engineering and Manufacturing Division, FAA	1
Civil Aeromedical Research Institute, FAA	2
The Surgeon General	1
US Army Aeromedical Research Unit	1

Armed Forces Institute of Pathology	1
Naval Air Engineering Center, Philadelphia	1
Naval Air Development Center, Johnsville	1
Helicopter Combat Support Squadron TWO, NAS, Lakehurst	1
HQ, US Marine Corps	1
Director of Army Aviation, ODCSOPS	1
Office of Assistant Chief of Staff for Force Development	1
Office of Deputy Chief of Staff for Personnel	1
National Library of Medicine	1
National Institutes of Health	1
Defense Documentation Center	20
US Government Printing Office	1

APPENDIX I

PULL TEST RESULTS

TABLE IV
0.062-INCH-DIAMETER WIRE, 6061-T6 ALUMINUM PLATEN, HOLE EDGES BROKEN

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (lb)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
3	1/4	3/4	1/8	-	101-191	146	
5	1/4	3/4	1/4	406	320-367	343	
6	1/4	3/4	5/16	544	Failure	-	
7	1/4	3/4	3/8	560	Failure	-	
67	1/4	3/4	1/8	116‡	90-100	95	1
8	1/4	5/8	1/8	160	120-140	130	
9	1/4	5/8	1/4	438	Failure	-	
11	1/4	5/8	3/8	584	Failure	-	
12	1/4	1/2	1/8	200	140-185	162	
13	1/4	1/2	1/8	460	Failure	-	
14	1/4	1/2	1/8	580	Failure	-	
86	1/4	1/2	5/16	430	320-400	360	2
16	5/16	3/4	1/8	104	85-90	87	
30	5/16	3/4	1/8	150	60-105	82	
31	5/16	3/4	1/4	288	140-180	160	
20	5/16	3/4	5/16	304	160-195	177	
32	5/16	3/4	3/8	380	200-260	230	
33	5/16	3/4	1/8	94	80-84	82	3
34	5/16	3/4	1/4	154	100-130	115	3
35	5/16	3/4	3/8	275	160-210	185	3
10	5/16	5/8	5/16	345	195-210	202	
15	5/16	1/2	3/8	430	335-390	367	
17	3/8	3/4	1/8	80	70-80	75	
28	3/8	3/4	1/8	83	60-82	71	
29	3/8	3/4	1/4	140	105-130	117	
21	3/8	3/4	5/16	235	135-160	147	
36	3/8	3/4	1/8	85	50-70	60	3
37	3/8	3/4	1/4	120	90-110	100	3
53	3/8	3/4	3/8	184	140-160	150	1
18	7/16	3/4	1/8	97	78-80	79	
22	7/16	3/4	5/16	145	105-120	112	
19	1/2	1	1/8	56	40-47	44	
23	1/2	1	5/16	118	75-100	87	

* All loads for two wires
‡ Eliminated peak load by loading and unloading unit prior to testing

1 2024-T3 Al platen
2 Hole edge rounded by polishing
3 Chamfered hole edge

TABLE V
0.072-INCH-DIAMETER WIRE, 2024-T3 PLATEN MATERIAL, HOLE EDGES BROKEN

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (in.)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
40	1/4	3/4	1/8	248	160-220	190	1
68	1/4	3/4	1/8	227‡	160-178	169	
41	1/4	3/4	1/4	731	Failure	-	
81	1/4	3/4	1/4	299	230-268	249	
42	1/4	3/4	3/8	719	Failure	-	
87	1/4	1/2	5/16	668	620-660	640	1 2
54	3/8	3/4	1/8	184	120-180	150	3
55	3/8	3/4	1/4	188	164-180	172	3
56	3/8	3/4	3/8	414	210-320	265	3
57	3/8	3/4	1/8	168	130-160	145	
58	3/8	3/4	1/8	212	180-192	186	4
59	3/8	3/4	1/8	310	240-280	260	5

* All loads for two wires
‡ Eliminated peak load by loading and unloading unit prior to testing
1 6061-T6 Al platen
2 Hole edge rounded by polishing
3 Staggered pattern - one wire pulled - two-wire value given
4 Five-hole staggered pattern
5 Four-hole "T" pattern

TABLE VI
0.080-INCH-DIAMETER WIRE, 2024-T3 ALUMINUM PLATEN MATERIAL, HOLE EDGE BROKEN

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (in.)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
24	1/4	3/4	1/8	450	300-430	365	
61	1/4	3/4	1/8	452	300-340	320	
27	1/4	3/4	1/4	840	Failure	-	
66	1/4	3/4	1/4	602	488-550	519	
25	1/4	3/4	3/8	894	Failure	-	
84	1/4	3/4	3/8	560	475-550	512	
82	1/4	3/4	5/16	610	510-580	545	1
38	1/4	3/4	1/8	700	360-410	385	2
47	5/16	3/4	1/8	278	190-250	220	
48	5/16	3/4	1/4	494	400-470	435	
49	5/16	3/4	3/8	728	480-560	520	
44	3/8	3/4	1/8	223	204-212	208	
45	3/8	3/4	1/4	416	318-368	343	
46	3/8	3/4	3/8	664	440-520	480	
72	3/8	1/2	3/8	912	Failure	-	
88	1/4	1/2	5/16	840	740-820	780	1 3
108	1/4	1/2	1/8	573	515-570	542	
109	1/4	1/2	1/4	865	Failure	-	
110	1/4	1/2	3/8	935	Failure	-	
117	1/4	1/2	1/8	575	445-525	488	3
118	1/4	1/2	1/4	855	Failure	-	3
119	1/4	1/2	3/8	950	Failure	-	3

* All loads for two wires
1 6061-T6 Al platen
2 304 stainless steel platen
3 Hole edges rounded by polishing

TABLE VII
0.092-INCH-DIAMETER WIRE, 2024-T3 ALUMINUM PLATEN, HOLE EDGES BROKEN

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (in.)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
1	1/4	3/4	1/8	646	450-560	505	1
2	1/4	3/4	1/8	394	310-360	335	1 2
63	1/4	3/4	1/8	796	460-580	520	
65	1/4	3/4	1/8	845	Failure	-	3
69	1/4	3/4	1/8	942	Failure	-	2 3
90	1/4	3/4	1/8	435	300-375	337	4
126	1/4	3/4	1/8	597	400-500	450	1 4
127	1/4	3/4	1/8	525	365-450	407	2
79	1/4	3/4	1/8	668	470-500	485	2 3
60	1/4	3/4	1/4	903	800-870	835	1
64	1/4	3/4	1/4	1167	Failure	-	
78	1/4	3/4	1/4	535	450-500	475	1
83	1/4	3/4	1/4	1050	775-950	862	1
91	1/4	3/4	1/4	815	680-750	715	4
99	1/4	3/4	1/4	1100	Failure	-	1
85	1/4	3/4	5/16	980	825-875	850	1
75	1/4	5/8	1/8	820	560-690	625	
76	1/4	5/8	1/8	1175	Failure	-	5
96	1/4	5/8	1/8	750	500-635	567	
104	1/4	5/8	1/8	1135	800-950	875	5
97	1/4	5/8	1/4	975	800-900	850	
98	1/4	5/8	3/8				Did not run
102	1/4	1/2	1/8	800	650-700	675	
111	1/4	1/2	1/8	1050	700-750	725	
120	1/4	1/2	1/8	900	645-745	695	4
112	1/4	1/2	1/4	1130	Failure	-	
121	1/4	1/2	1/4	1300	Failure	-	4
89	1/4	1/2	5/16	1185	1000-1170	1085	4
113	1/4	1/2	3/8	1208	Failure	-	
122	1/4	1/2	3/8	1208	Failure	-	4
70	5/16	3/4	1/8	420	272-292	282	3
77	5/16	5/8	1/8	765	650-730	690	5
71	3/8	3/4	1/8	350	238-248	243	3
73	3/8	5/8	3/8	898	700-850	775	
74	3/8	1/2	3/8	1058	Failure	-	

* All loads for two wires
1 6061-T6 Al platen
2 Chamfered hole edges
3 304 stainless steel platen
4 Hole edges rounded by polishing
5 Four holes in line

TABLE VIII
0.105-INCH-DIAMETER WIRE, 2024-T3 ALUMINUM PLATEN,
HOLE EDGES ROUNDED BY POLISHING

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (in.)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
26	1/4	3/4	1/8		Failure	-	1
93	1/4	3/4	1/8	685	535-575	555	
94	1/4	3/4	1/4	1155	950-1050	1000	
95	1/4	3/4	3/8	1325	1140-1200	1170	
99	1/4	5/8	1/8	1150	825-950	887	
105	1/4	5/8	1/8	1360	Failure	-	2
106	1/4	5/8	1/8	1240	1075-1225	1150	2
107	1/4	5/8	1/8	1435	Failure	-	2
100	1/4	5/8	1/4	1450	Failure	-	
101	1/4	5/8	3/8				Did not run
103	1/4	1/2	1/8		Failure	-	
114	1/4	1/2	1/8	1430	Failure	-	
123	1/4	1/2	1/8	1310	Failure	-	
115	1/4	1/2	1/4	1350	Failure	-	
124	1/4	1/2	1/4	1435	Failure	-	
116	1/4	1/2	3/8			-	Did not run
125	1/4	1/2	3/8			-	Did not run
62	3/8	3/4	1/8	382	340-380	360	1
51	3/8	3/4	1/4	830	700-800	750	1
128**	1/4	3/4	1/4	1020	960-1000	980	3 4
129**	1/4	3/4	1/4	950	916-940	930	3 5
130**	1/4	3/4	1/4	836	820-836	828	3 6

* All loads for two wires
** Test includes wires, platen, and guides at one end only
1 Hole edge broken
2 Four holes in line
3 Leading hole edge $\rho_1 = 1/8R$
4 All other edges $\rho_2 = 1/16R$
5 All other edges $\rho_2 = 5/64R$
6 All other edges $\rho_2 = 3/32R$

TABLE IX
0.120-INCH-DIAMETER WIRE, 304 STAINLESS STEEL PLATEN, HOLE EDGES BROKEN

Pull Test No.	Hole Dia. (in.)	Hole Spacing (in.)	Platen Thickness (in.)	*Peak Load (lb)	*Pull Load Fluctuation (lb)	*Mean Load (lb)	Remarks
39	1/4	3/4	1/8	1840	Failure	-	
43	3/8	3/4	1/8	1170	650-900	775	

* All loads for two wires

TABLE X
WIRE ULTIMATE TENSILE VALUES

Wire Diameter (in.)	*Ultimate Pull Load (Single Wire) (lb)	80 Pct of Ultimate Load (lb)
0.062	305	244
0.070	396	317
0.080	529	424
0.092	662	529
0.105	725	580
0.120	1120	895

*Actual pull test of wires to destruction.

TABLE XI
TEST PLAN

Test No.	Wire* Size (in.)						Number of Wires			Platen Material			Platen Thickness (in.)				Hole Diameter (in.)					Hole Spacing (in.)				Weaving Arrangement In-line Holes			Weaving Arrangement Staggered	
	0.062	0.070	0.080	0.092	0.105	0.120	1	2	4	6061-T6AL	2024-T3AL	304SS (annealed)	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	3	4	5	3	5"T"
1				X				X		X			X				X							X		X				
2				X				X		X			X				X							X		X				
3	X							X		X			X				X							X		X				
4	X							X		X			X				X							X				X		
5	X							X		X				X			X							X		X				
6	X							X		X					X		X							X		X				
7	X							X		X						X	X							X		X				
8	X							X		X			X				X						X			X				
9	X							X		X				X			X						X			X				
10	X							X		X					X			X					X			X				
11	X							X		X						X	X						X			X				
12	X							X		X			X				X					X				X				
13	X							X		X				X			X					X				X				
14	X							X		X					X		X					X				X				
15	X							X		X						X		X				X				X				
16	X							X		X			X					X						X		X				
17	X							X		X			X						X					X		X				
18	X							X		X			X							X				X		X				
19	X							X		X			X								X				X	X				
20	X							X		X					X			X						X		X				
21	X							X		X					X				X					X		X				
22	X							X		X					X					X				X		X				
23	X							X		X					X						X				X	X				
24			X					X			X		X				X							X		X				
25			X					X			X					X	X							X		X				
26					X			X			X		X				X							X		X				
27			X					X			X			X			X							X		X				
28	X							X		X			X						X					X		X				
29	X							X		X				X					X					X		X				

*Wire Material: 304SS annealed

TABLE XI (continued)
TEST PLAN

Test No.	Wire* Size (in.)						Number of Wires			Platen Material			Platen Thickness (in.)				Hole Diameter (in.)					Hole Spacing (in.)				Weaving Arrangement In-line Holes			Weaving Arrangement Staggered		
	0.062	0.070	0.080	0.092	0.105	0.120	1	2	4	6061-T6AL	2024-T3AL	304SS (annealed)	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	3	4	5	3	5	"T"
30	X							X		X			X					X						X		X					
31	X							X		X				X				X						X		X					
32	X							X		X						X		X						X		X					
33	X							X		X			X					X						X		X					
34	X							X		X				X				X						X		X					
35	X							X		X						X		X						X		X					
36	X							X		X			X						X					X		X					
37	X							X		X				X					X					X		X					
38			X					X				X	X				X							X		X					
39						X		X				X	X				X							X		X					
40		X						X			X		X				X							X		X					
41		X						X			X			X			X							X		X					
42		X						X			X					X	X							X		X					
43						X		X				X	X						X					X		X					
44			X					X			X		X						X					X		X					
45			X					X			X			X					X					X		X					
46			X					X			X					X		X						X		X					
47			X					X			X		X					X						X		X					
48			X					X			X			X				X						X		X					
49			X					X			X					X			X					X		X					
50					X			X			X		X						X					X		X					
51					X			X			X			X					X					X		X					
52					X			X			X					X			X					X		X					
53	X							X			X					X			X					X		X					
54		X					X				X		X						X					X					X		
55		X					X				X			X					X					X					X		
56		X					X				X					X			X					X					X		
57		X					X				X		X						X					X		X					

*Wire Material: 304SS annealed
**3 In-line holes with a fourth hole offset at the second hole

TABLE XI (continued)
TEST PLAN

Test No.	Wire Size (in.)						Number of Wires			Platen Material			Platen Thickness (in.)				Hole Diameter (in.)					Hole Spacing (in.)				Weaving Arrangement In-line Holes			Weaving Arrangement Staggered		
	0.062	0.070	0.080	0.092	0.105	0.120	1	2	4	6061-T6AL	2024-T3AL	304SS (annealed)	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	3	4	5	3	5	"T"
58		X					X				X		X						X					X						X	
59		X					X				X		X						X					X							X
60				X			X			X				X			X							X		X					
61			X				X				X		X				X							X		X					
62						X	X				X		X						X					X		X					
63				X			X				X		X				X							X		X					
64				X			X				X			X			X							X		X					
65				X			X					X	X				X							X		X					
66			X				X				X			X			X							X		X					
67							X				X		X				X							X		X					
68		X					X			X			X				X							X		X					
69				X			X					X	X				X							X		X					
70				X			X					X	X					X						X		X					
71				X			X					X	X						X					X		X					
72			X				X				X					X			X			X				X					
73				X			X				X					X			X				X			X					
74				X			X				X					X			X			X				X					
75				X			X				X		X				X						X			X					
76				X			X				X		X				X						X				X				
77				X			X				X		X					X					X				X				
78				X			X			X				X			X							X		X					
79				X			X					X	X				X							X		X					
80				X			X				X			X			X							X		X					
81		X					X				X			X			X							X		X					
82			X				X			X					X		X							X		X					
83				X			X			X				X			X							X		X					
84			X				X				X					X	X							X		X					
85				X				X		X					X		X							X		X					

*Wire Material: 304SS annealed

**3 In-line holes with a fourth hole offset at the second hole

TABLE XI (continued)
TEST PLAN

Test No.	Wire* Size (in.)						Number of Wires			Platen Material			Platen Thickness (in.)				Hole Diameter (in.)					Hole Spacing (in.)				Weaving Arrangement: In-line Holes			Weaving Arrangement: Staggered		
	0.120	0.105	0.092	0.080	0.070	0.062	1	2	4	304SS (annealed)	2024-T3AL	6061-T6AL	1/8	1/4	5/16	3/8	1/4	5/16	3/8	7/16	1/2	1/2	5/8	3/4	1	3	4	5	3	5	"T"
86						X		X				X			X		X					X				X					
87					X			X				X			X		X					X				X					
88				X				X				X			X		X					X				X					
89			X					X				X			X		X					X				X					
90			X					X			X		X				X							X		X					
91			X					X			X		X				X							X		X					
92			X					X			X					X	X							X		X					
93		X						X			X		X				X							X		X					
94		X						X			X			X			X							X		X					
95		X						X			X					X	X							X		X					
96			X					X			X		X				X						X			X					
97			X					X			X			X			X						X			X					
98			X					X			X					X	X						X			X					
99		X						X			X		X				X						X			X					
100		X						X			X			X			X						X			X					
101		X						X			X					X	X						X			X					
102			X					X			X		X				X					X				X					
103		X						X			X		X				X					X				X					
104			X					X			X		X				X						X				X				
105		X						X			X		X				X						X				X				
106		X						X			X		X				X						X				X				
107		X						X			X		X				X						X				X				
108				X				X			X		X				X					X				X					
109				X				X			X			X			X					X				X					
110				X				X			X					X	X					X				X					
111			X					X			X		X				X					X				X					
112			X					X			X			X			X					X				X					
113			X					X			X					X	X					X				X					
114		X						X			X		X				X					X				X					

*Wire Material: 304SS annealed

TABLE XI (continued)
TEST PLAN

Test No.	Wire Size* (in.)						Number of Wires			Platen Material			Platen Thickness (in.)				Hole Diameter (in.)				Hole Spacing (in.)				Weaving Arrangement: In-line Holes			Weaving Arrangement: Staggered	
	0.120	0.105	0.092	0.080	0.070	0.062	1	2	4	304SS (annealed)	2024-T3AL	6061-T6AL	1/8	1/4	5/16	3/8	1/4	5/16	3/8	7/16 1/2	1/2	5/8	3/4	1	3	4	5	3	5"T"
115		X						X			X			X			X				X				X				
116		X						X			X					X	X				X				X				
117				X				X			X		X				X				X				X				
118				X				X			X			X			X				X				X				
119				X				X			X					X	X				X				X				
120			X					X			X		X				X				X				X				
121			X					X			X			X			X				X				X				
122			X					X			X					X	X				X				X				
123		X						X			X		X				X				X				X				
124		X						X			X			X			X				X				X				
125		X						X			X					X	X				X				X				
126			X					X				X	X				X				X				X				
127			X					X			X		X				X				X				X				
128		X						X			X			X			X						X		X				
129		X						X			X			X			X						X		X				
130		X						X			X			X			X						X		X				

*Wire Material: 304SS annealed

APPENDIX II

ENERGY ABSORBER CONFIGURATIONS

WIRE-BENDING ENERGY ABSORBERS (LOAD LIMITERS)

The following energy absorbers are configurations that are plausible. They have been considered primarily for 5000-pound capacity.

Two-Spool, Single-Platen Unit

This load limiter concept has two spools attached to one end of the platen, either one above the other (Figure 53) or side by side (Figure 54). Both concepts include the same operational aspects. One spool will store all the wires woven through the top side of the end hole in the platen, and the bottom spool will store all wires woven out the bottom side of the same hole.

The platen will have the same slots and spacing as discussed for the self-storing unit (page 36). However, since the spool requires a greater bend radius, it will not be feasible to use 0.105-inch wire diameter with small spools. To have a compact load limiter, a decrease in wire size and an increase in the quantity of wire are required. Also, a greater resistance to the pull load will be realized. Only through tests will this additional load be determined and the platen construction be altered to reflect the increased intensity.

The prepulling of the device and the testing of additional hardware to complete the restraint device will be accomplished in the same manner as discussed for the self-storing unit (page 36).

There are two feasible methods of protecting these units. One is to cast a plastic cover in two halves and bind them together. The second method requires stamping out sheet metal covers and riveting them together, using the rivets to hold the spools in place.

Two-Spool, Double-Platen Unit

This counterflexing load limiter concept is similar in all aspects to the two-spool, single-platen unit concept except that two platens, instead of one, are utilized to function as one unit. The upper spool feeds wire into the top side of each platen, and the lower spool feeds into the bottom side of each platen. See Figure 55. This concept offers no additional advantages over the two-spool, single-platen unit.

Canister Storage Unit

The canister storage unit is the same as the previously discussed concepts except for the storage area. The wires are stored like a ball of twine in a canister. The canister will have a contoured conic section so that the wire can pay out without any interference. See Figure 56.

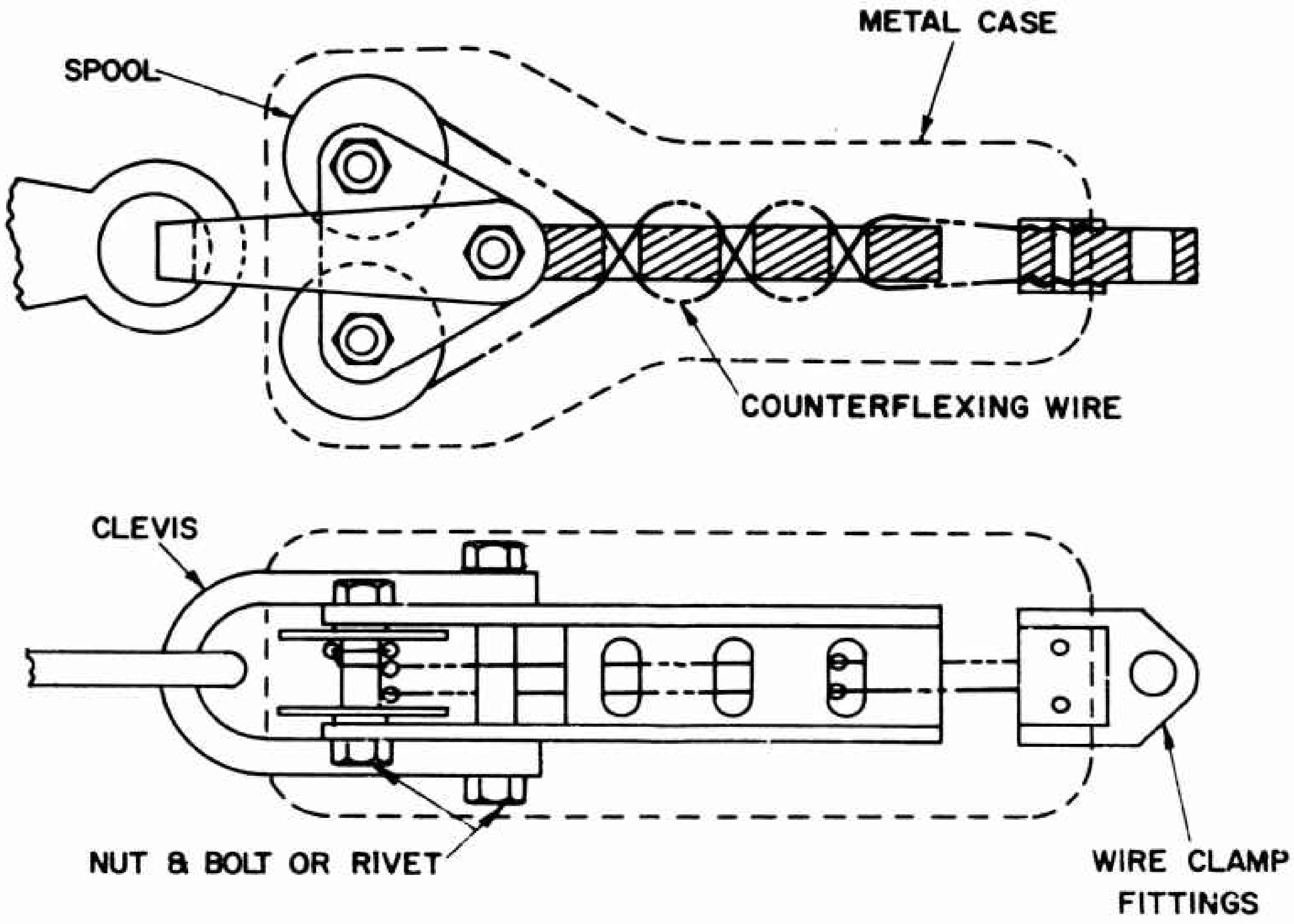


Figure 53. Single-Platen Unit; Two Spools, One Above the Other

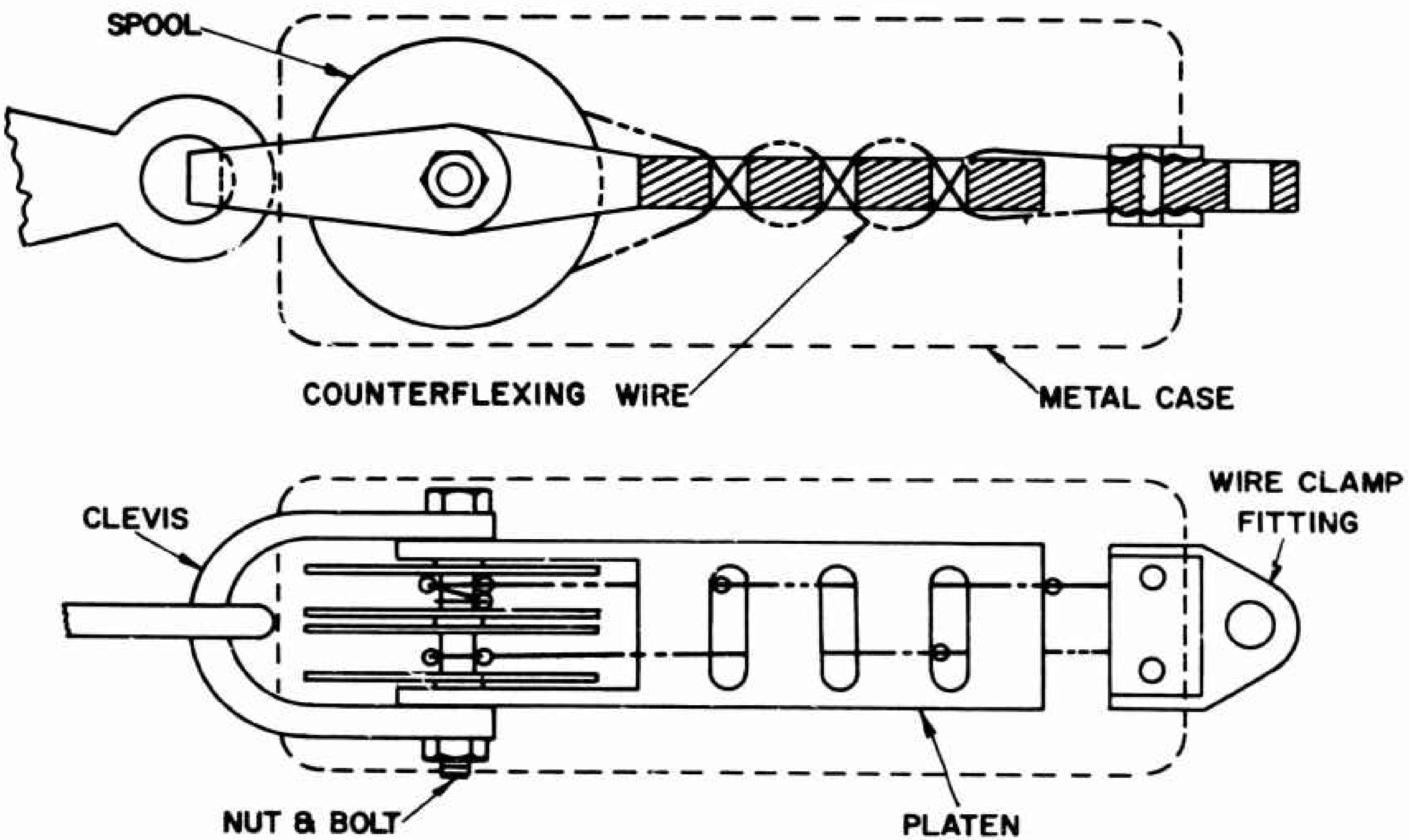


Figure 54. Single-Platen Unit; Two Spools, Side by Side

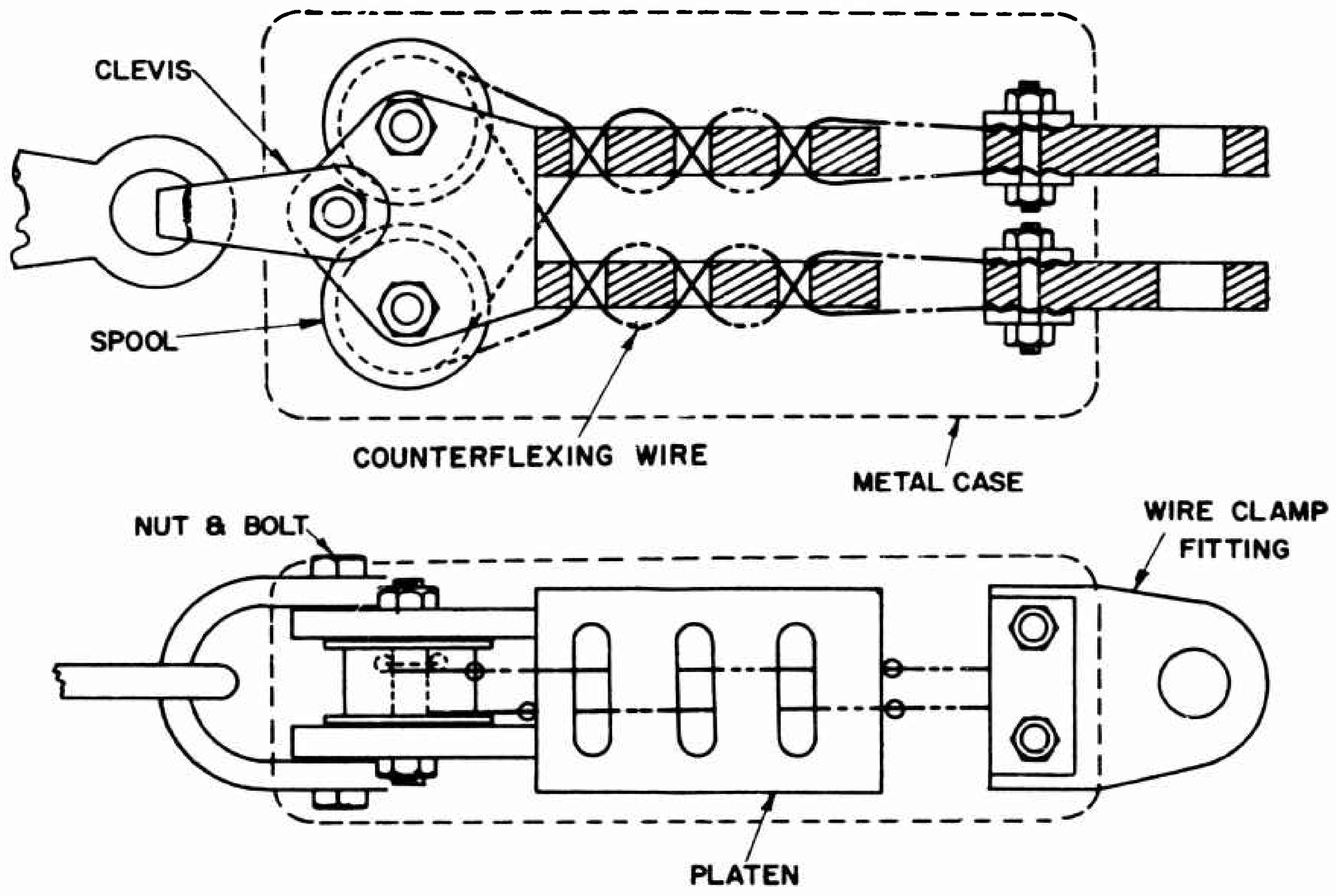


Figure 55. Two-Spool, Double-Platen Unit

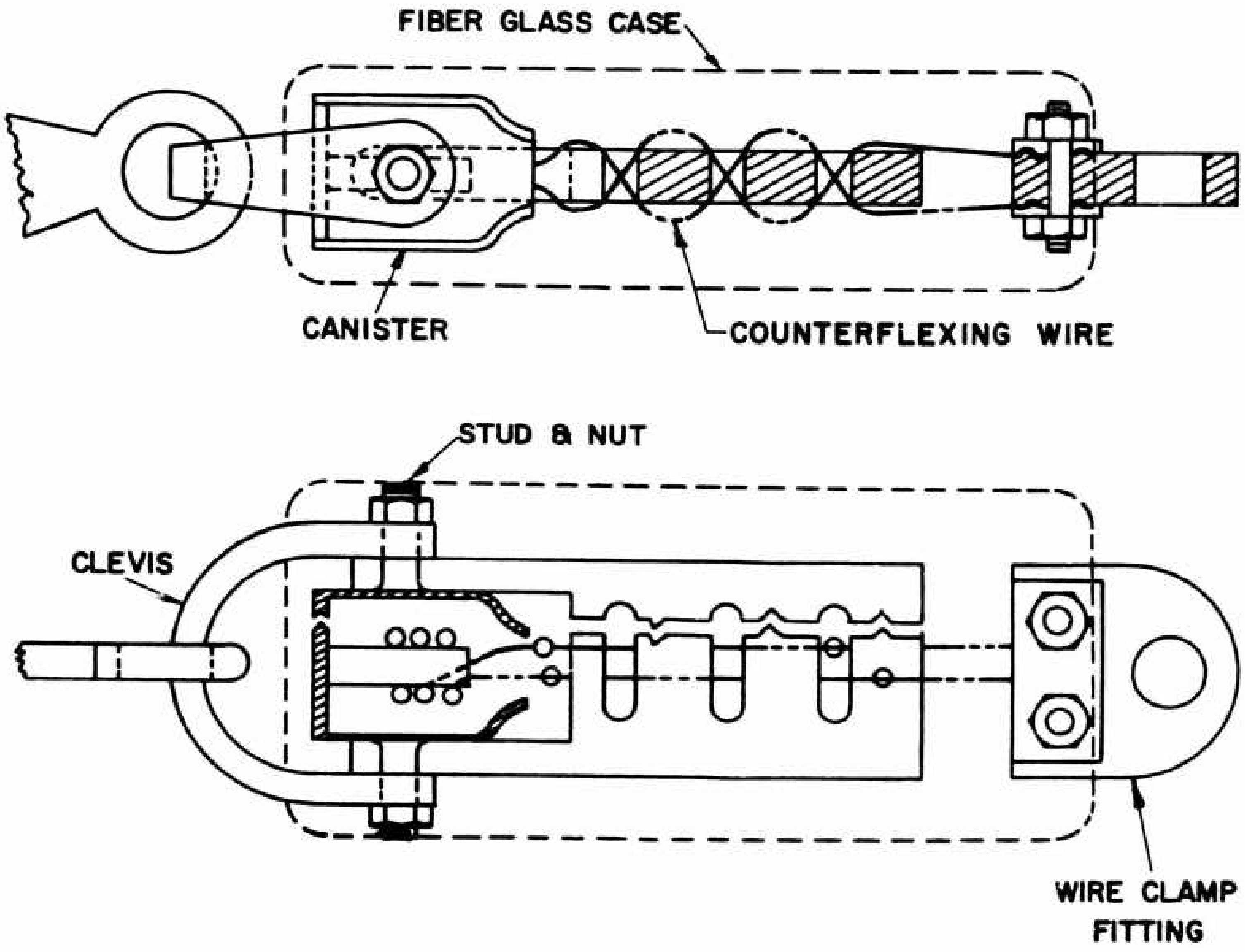


Figure 56. Canister Storing Unit

APPENDIX III

VALIDATION TESTS

OF SELF-STORING, WIRE-BENDING ENERGY ABSORBERS

The self-storing, wire-bending energy absorber has been designed and tested with an 8-inch-stroke capability to replace the design criterion of 2 to 3 feet as requested by the Government in the latter phase of the contract.

The results of the laboratory test data are shown in Table XII and are in reference to test unit configuration, pre-pull loads, final pull loads, and description of test results.

A succession of additional tests has been conducted to eliminate the inconsistency of pull test values that arose from the process of encapsulating the energy absorbers. A series of experiments was conducted to isolate the factors that could conceivably cause the load changes; they are reviewed below.

1. A polyethylene bag, which encapsulated the wire-platen assembly and then was potted with polyurethane, leaked.

2. A polyethylene bag was replaced with heat-shrinkable mylar polyester film. High peak loads resulted.

3. Putty placed in proximity over pertinent tongue location and mylar film resulted in loads below acceptable limits.

4. To cope with case movement and polyurethane shrinkage, grooves were cut out of the platen and spacers were added to guide areas with no reduction in loads (7,000 pounds).

5. Shields were placed over all exposed wires. Of two tests performed, one was within acceptable limits.

When the first test of item 3 resulted in low loads (4,000 pounds), the wire-platen assembly was cleaned in a Chlorothene solution. The remaining test in this group still showed no variance in results. The test of item 5 that resulted in good results was not cleaned in the Chlorothene solution. It was then concluded that the degreasing operation had increased the friction coefficient between wires and platen and, consequently, amplified the pull loads. Tests 43B and 43B1 further substantiated this.

The addition of shields placed over exposed wire locally at tongue location eliminated high initial peak load that resulted during final pull testing. In general, tests 50 through 57 show consistently good test results that validate the energy absorber concept. The majority of the test units have been pulled on the basis of rated 5,000-pound units. Validation of rated 10,000-pound units is realized when comparing tests 9A and 9B with tests 7A, 7B, 8A, and 8B. The final pull load results of tests 9A and 9B are approximately one-half the intensity of the other two tests, and, therefore, it is felt that no further testing of 10,000-pound units is necessary.

The testing was conducted on a Baldwin-Emery SR-4, Model F.G.T. test machine. The rate of loading was consistently held in the range of 3.0 to 5.0 inches per minute. The machine is capable of a load rate in the range of 0.008 to 9.6 inches per minute. It is to be noted from test 55 that an increase in the rate of pull results in increased load intensity.

From tests 52, 54, 55, and 57, it can be seen that the peak load intensity realized during the pre-pull loading was eliminated during the final pull loading.

The above data clearly demonstrates the capability of the self-storing, wire-bending energy absorber to develop the desired level of load limitations.

Figures 57 through 68 depict the self-storing units in sequence from the basic parts to the final pull testing.

TABLE XII
LABORATORY TEST RESULTS
OF THE
SELF-STORING, WIRE-BENDING ENERGY

Test No.	PREPULL LOADS Peak (lb.)	PREPULL LOADS Fluctuation (lb.)	FINAL PULL LOADS Peak (lb.)	FINAL PULL LOADS Fluctuation (lb.)	TEST UNIT CONFIG
1			9380		Polyethylene bag (3M) and ... potting
2	4910	4800-4900	4780	4400-4700	Polyethylene bag (6M) an... potting.
3	5125	4880	2500	2400-2500	Polyethylene bag (6M) and ... potting.
4		4860	8000		Polyethylene bag (6M) and ... potting.
5	5100	4800-4900	5400	4800-4900	Heat-Shrinkable Mylar F... potting.
6	5400	5140	5730	4600-5200	Heat-Shrinkable Mylar F... potting.
7A	4680	4400-4500	9200	8100-8300	Putty located on front end... Mylar Film, and Polyure... units pulled simultaneous... of 10,000 lb.
B	5520	4800-5000			
8A	5000	4700	8720	7900-8400	Same as Test 7A & B exc... platen assembly in Chlor...
B	5020	4700			
9A	5100	4700	4350	4000-4250	Same as Test 8A & B exc... tested separately.
B	4800	4600-4780	4290	4000-4220	
36	5200	4700	6800		Cleaned wire-platen asse... solution; Mylar Film; put... end of tongue plates betw... Polyurethane Potting.

*This column describes the additional parts and operations undertaken after installation of the ba...

A

LE XII
[TEST RESULTS
THE
ƐNDING ENERGY ABSORBERS

T UNIT CONFIGURATION*	DESCRIPTION OF TEST RESULTS
lene bag (3M) and Polyurethane	Polyethylene bag leaked during Polyurethane curing cycle and failure occurred.
lene bag (6M) and Polyurethane	Test unit functioned within acceptable limits.
lene bag (6M) and Polyurethane	Polyethylene bag leaked during Polyurethane curing cycle. Five wires failed and five wires pulled at one-half acceptable limits.
lene bag (6M) and Polyurethane	Polyethylene bag leaked during Polyurethane curing cyle and wire failure occurred.
inkable Mylar Film and Polyurethane	High peak load on final pull. Fluctuation load within acceptable limits.
inkable Mylar Film and Polyurethane	Prepull loads higher than acceptable limits but fluctuation final pull to prepull ratio depicts acceptable limits. Final peak loads to peak prepull ratio is high and outside acceptable limits. Potting at front end of tongue plates appears to be too stiff.
ated on front end of Tongue Plates, ilm, and Polyurethane Potting. Two led simultaneously for rated capacity) lb.	Final fluctuation load was too low.
Test 7A & B except cleaned wire ssembly in Chlorothene solution.	Final peak and fluctuation loads were too low.
Test 8A & B except each unit was eparately.	Final peak and fluctuation loads were too low. Tests 7A through 9B conclusively prove that putty became volatile during Polyurethane curing cycle and lubricated wires.
wire-platen assembly in Chlorothene ; Mylar Film; putty located at front ongue plates between Mylar Film and thane Potting.	Wire failure. Indications of Polyurethane bearing against wires over guides. This could be attributed to potting shrinkage and potting cover shifting during loading.

allation of the basic wire-platen assembly (wire, platen, guides, filler, and tongue).

b

TABLE XII (continued)
LABORATORY TEST RESU
OF THE
SELF-STORING, WIRE-BENDING ENE

Test No.	PREPULL LOADS Peak (lb.)	Fluctuation (lb.)	FINAL PULL LOADS Peak (lb.)	Fluctuation (lb.)	TEST UNIT CONFI
37	5000	4700-4800	7000		Same as Test 36 except at guides location prior Mylar Film and potting.
38	5100	4600-4900	7000		Same as Test 37 except side edges of platen ne included.
39	5500	4600-4800	6800		Same as Test 38 except replaced by 3/16" spac
40	4600	3900-4200	4000	3800-4000	Same as Test 39 except was included on side ed location; wire-platen as and cut potting paper w tongue plates.
43	5000	4400-4800	7500		Same as Test 40 except was cleaned in Chloroth
45 (40)	5490	5400-5200	4800	4000-4400	Steel shields (25 gauge) wires and shrunk with platen assembly; was n thene solution. No Polyu
43A	5760	5400-5600			Same as Test 45 except was cleaned in Chloroth urethane potting.
43B	5135	4800-5000			Wire-platen assembly thene solution. Prepull stroke and halted test. potting used.
43B1			6000	Halted Test	Used same wire-platen 43B; was cleaned in Chl Continued pull test.
47	4800	4400-4600	5200	4580-4820	Mylar Film and Polyur

*This column describes the additional parts and operations undertaken after installation of the

A

ιLE XII (continued)
ιTORY TEST RESULTS
OF THE
RE-BENDING ENERGY ABSORBERS

TEST UNIT CONFIGURATION*	DESCRIPTION OF TEST RESULTS
ne as Test 36 except 1/8" spacers added guides location prior to installation of lar Film and potting. Also removed putty.	Wire failure. Similar to above.
ıe as Test 37 except a set of grooves on ɜ edges of platen near tongue location was luded.	Wire failure. Similar to above.
ıe as Test 38 except 1/8" spacers were laced by 3/16" spacers.	Wire failure. Similar to above.
ıe as Test 39 except another set of grooves included on side edges of platen at guides ıtion; wire-platen assembly was not cleaned; cut potting paper was thin at front end of gue plates.	Prepull and final pull loads are in acceptable ratio. Attributed to reducing of potting stiff-ness at pertinent tongue location.
ıe as Test 40 except wire-platen assembly cleaned in Chlorothene solution.	Wire failure. Potting shifted during pull test.
l shields (25 gauge) placed over exposed ɜs and shrunk with Mylar Film to wire-en assembly; was not cleaned in Chloro-e solution. No Polyurethane potting.	Final pull loads were lower than prepull loads.
e as Test 45 except wire-platen assembly cleaned in Chlorothene solution. No Poly-hane potting.	Wire failure. Concluded that cleaning in Chlorothene solution reacted with wire and platen to produce adverse pull loads.
ɜ-platen assembly not cleaned in Chloro-e solution. Prepulled assembly a short ke and halted test. No Mylar Film or ng used.	Tests 43B and 43B1 definitely prove that cleaning assembly in Chlorothene solution has adverse effect on pull loads.
l same wire-platen assembly as for Test was cleaned in Chlorothene solution. inued pull test.	
.r Film and Polyurethane Potting.	High peak load - fluctuation in acceptable range.

nstallation of the basic wire-platen assembly (wire, platen, guides, filler, and tongue).

TABLE XII (continued)
LABORATORY TEST RES
OF THE
SELF-STORING, WIRE-BENDING EN

Test No.	PREPULL LOADS Peak (lb.)	PREPULL LOADS Fluctuation (lb.)	FINAL PULL LOADS Peak (lb.)	FINAL PULL LOADS Fluctuation (lb.)	TEST UNIT CONF
53	4910	4400-4700	4780	4400-4780	Shields over tongue a potting. Potting cut at Wire-platen assembly on platen.
50	5100	4800-4900	5100	4600-4900	Same as Test 53 abov
51	5120	4800-4900	5060	4400-4900	Same as Test 50 abov not cut at front end of
52	5150	4900-5000	4900	4400-4900	Same as Test 51 abov
56	5120	4600-4800	5400	5100-5200	Same as Test 51 abov after potting were sea lucent Sealant.
54	4940	4800-4900	4820	4600-4800	Same as Test 56 exce sealed by taping prior Translucent Sealant.
55	5160	4730-4970	4950 5300	4700-4900 4900-5100	Same as Test 54.
57	4960	4600-4800	4800	4900-5000	Same as Test 54.

*This column describes the additional parts and operations undertaken after installation of the

ıLE XII (continued)
ATORY TEST RESULTS
OF THE
RE-BENDING ENERGY ABSORBERS

TEST UNIT CONFIGURATION*	DESCRIPTION OF TEST RESULTS
ɘlds over tongue area, Mylar Film and ing. Potting cut at front end of tongue. e-platen assembly has two sets of grooves ılaten.	Pull loads are in acceptable range.
ıe as Test 53 above.	High peak load was caused by the potting on the front end of the tongue as it separated. It rotated and hung upon tongue until complete separation.
e as Test 50 above except potting was cut at front end of tongue.	Final peak load of 5060 was momentary and is not felt to be significant.
e as Test 51 above.	Pull loads are in acceptable range.
e as Test 51 above except all openings r potting were sealed with RTV-108 Trans-nt Sealant.	Sealant became volatile during its curing cycle and amplified the final pull loads as indicated.
e as Test 56 except all openings were ed by taping prior to applying RTV-108 ıslucent Sealant.	Pull loads are in acceptable range.
e as Test 54.	The lower set of final loads is based on rate of pull load that is consistent with previous tests. Loads are acceptable. The higher pull loads were a result of increasing rate of pull during test.
ɘ as Test 54.	Pull loads are acceptable.

nstallation of the basic wire-platen assembly (wire, platen, guides, filler, and tongue).

8

Figure 57. Platen, Guides, Filler, and Wire

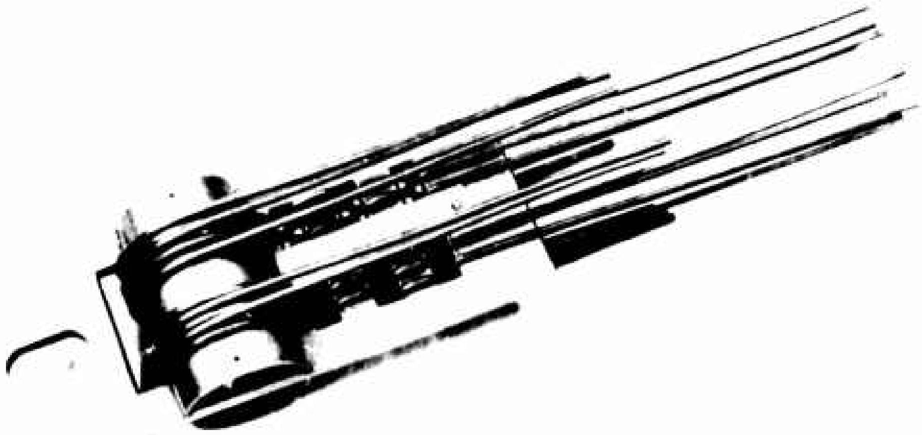


Figure 58. Assembly of Platen, Guides, Filler, and Wires; Plan View

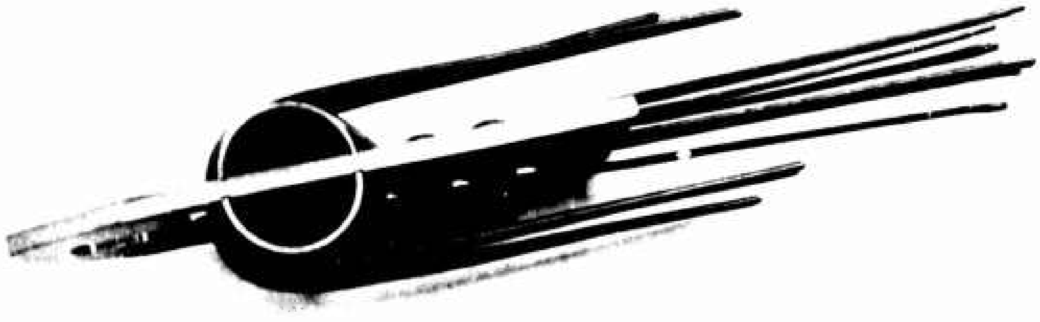


Figure 59. Assembly of Platen, Guides, Filler, and Wires; Side View

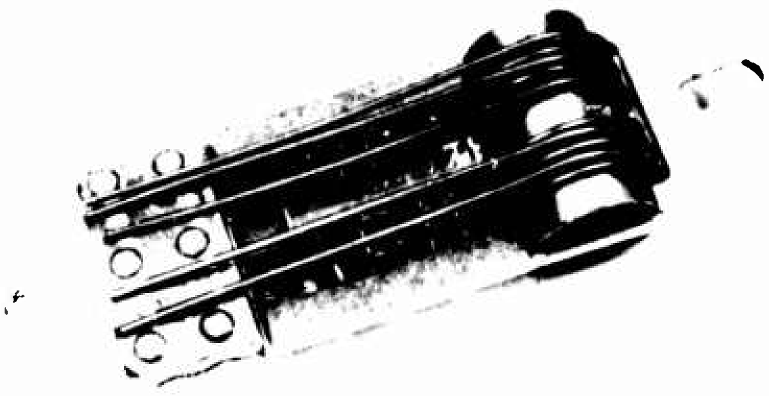


Figure 60. Assembly of Platen, Guides, Filler, Wires, and Tongue; Plan View

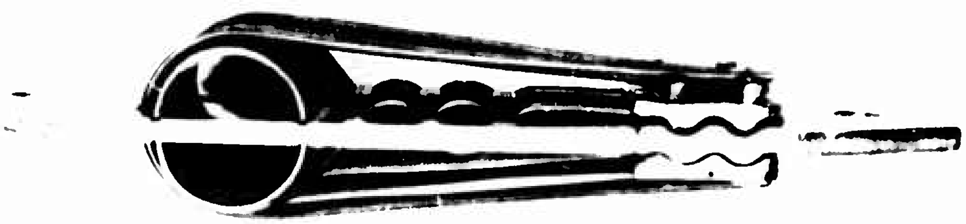


Figure 61. Assembly of Platen, Guides, Filler, Wires, and Tongue; Side View

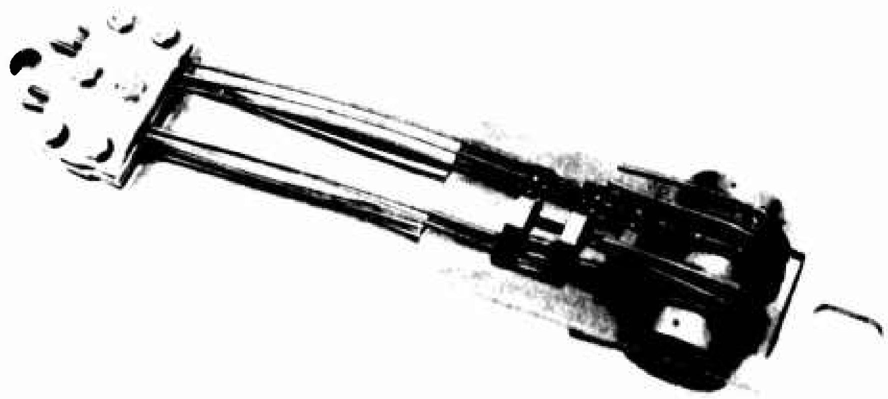


Figure 62. Configuration of Platen, Guides, Filler, Wires, and Tongue Assembly After Pull Tests

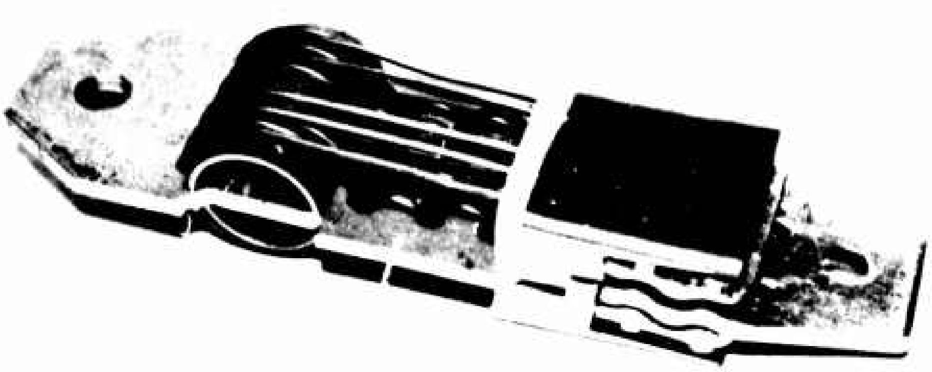


Figure 63. Assembly of Platen, Guides, Filler, Wires, Tongue, and Shields

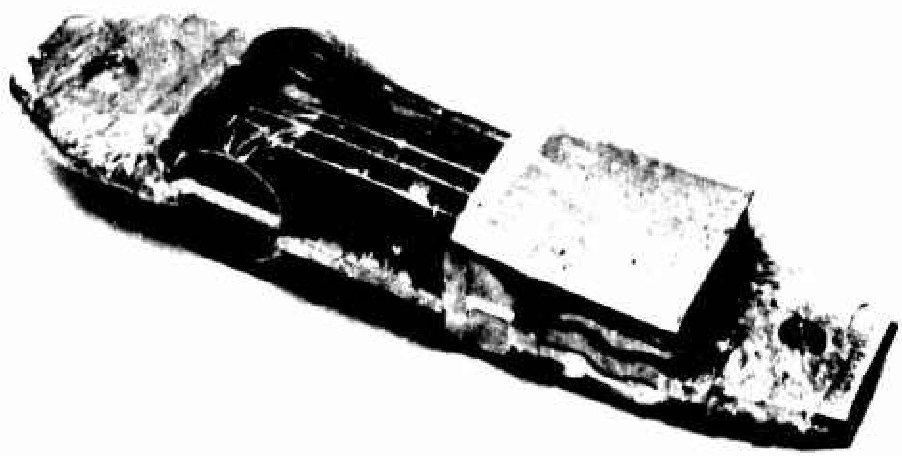


Figure 64. Assembly of Platen, Guides, Filler, Wires, Tongue, Shields, and Mylar Polyester Film

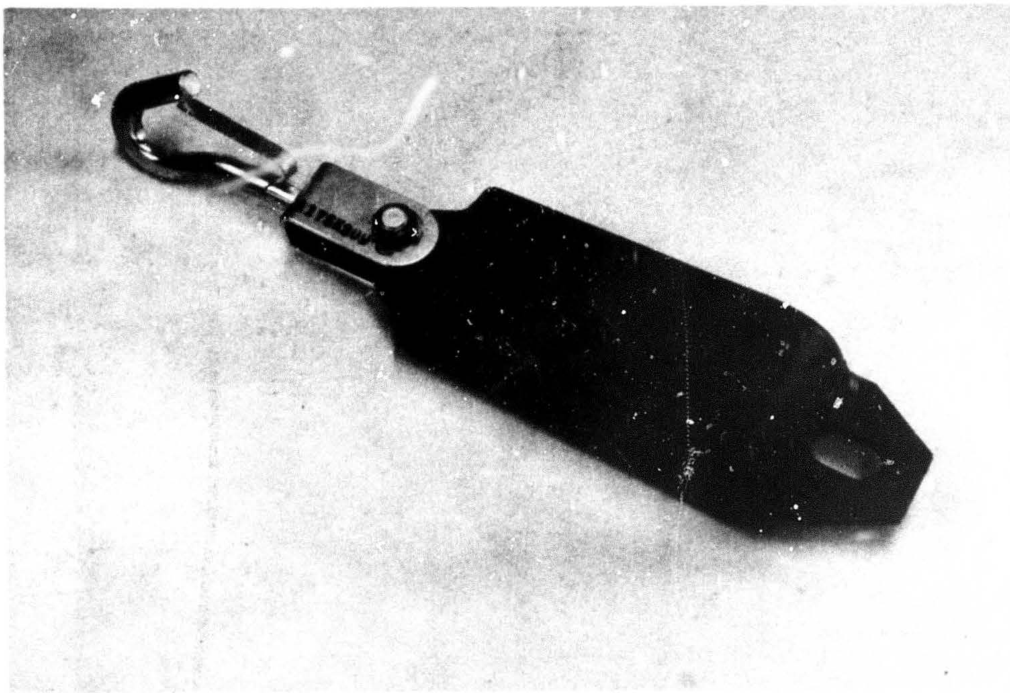


Figure 65. Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity)

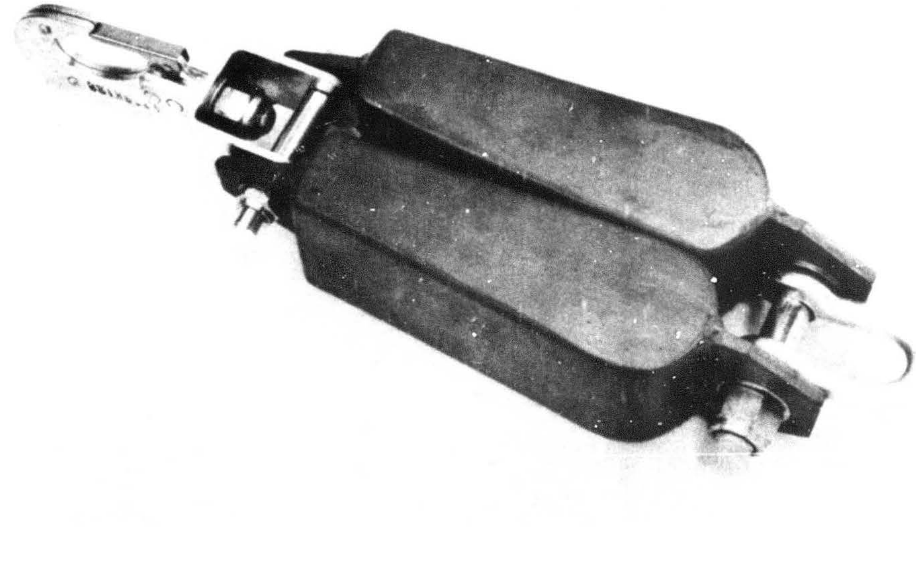


Figure 66. Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (10,000-Pound Rated Capacity)

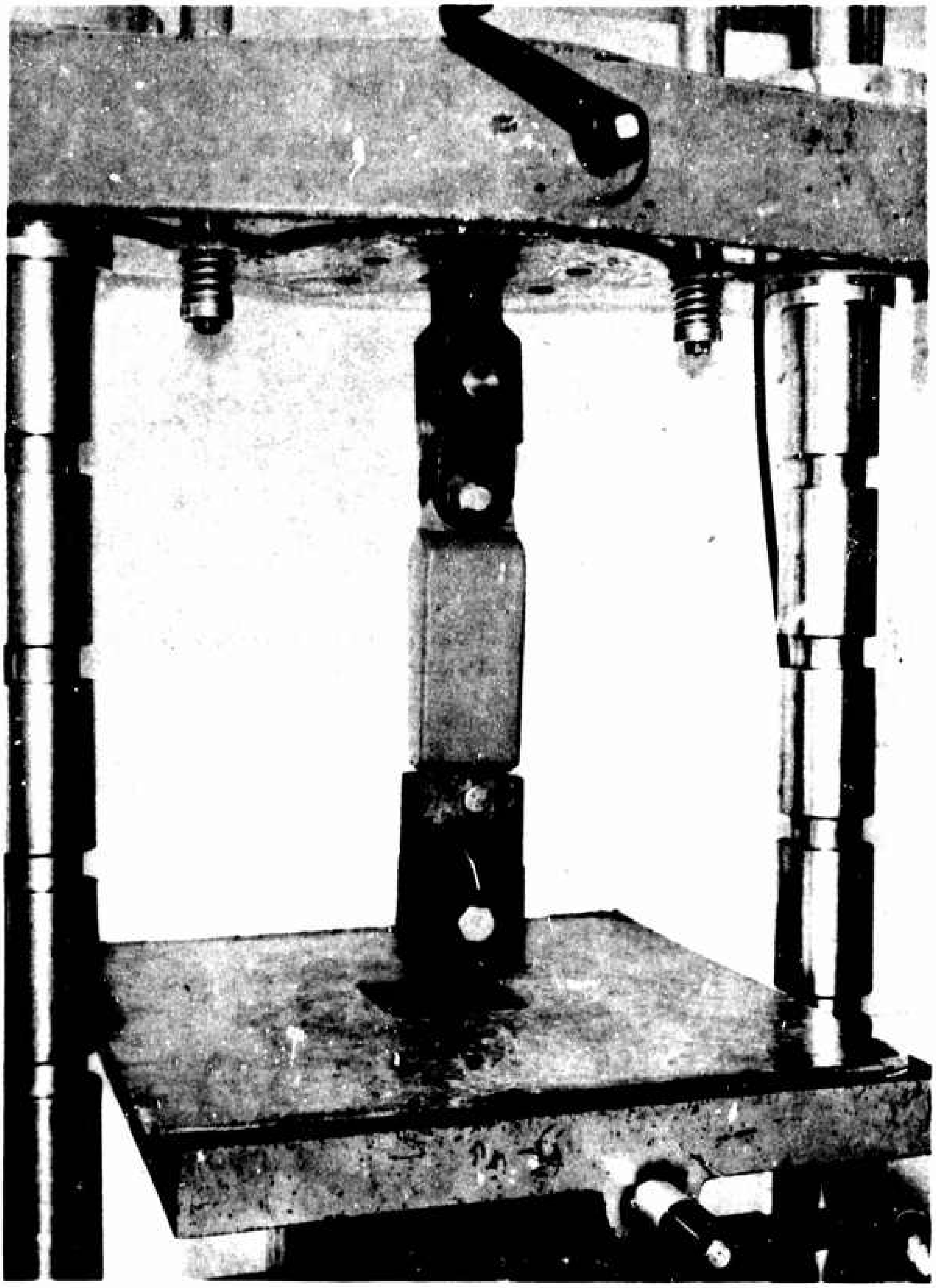


Figure 67. Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity) Attached in Test Machine Prior to Pull Test

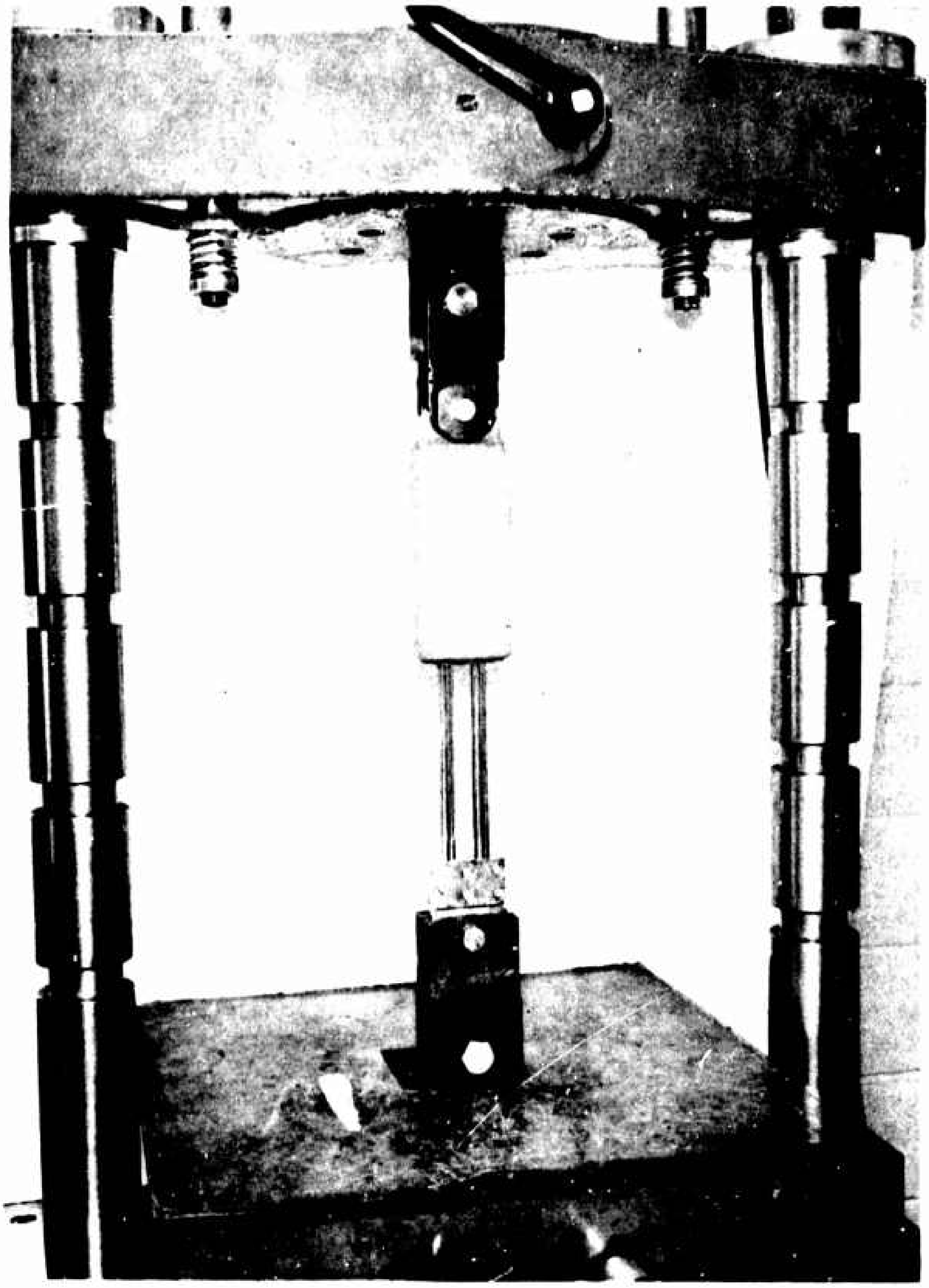


Figure 68. Self-Storing, Wire-Bending Load Limiter Type Energy Absorber (5,000-Pound Rated Capacity) Attached in Test Machine After Pull Test

APPENDIX IV

STUDY OF TEST METHODS

GENERAL

The purpose of the study is to determine the most expedient method of testing for pulse duration of various magnitudes. The test methods described herein are predicated on an equivalent square pulse curve simulating the triangular pulse-duration criterion of 25-g intensity and 0.25-second duration. Each test method is analyzed for its pulse envelope limitations and is depicted in an acceleration pulse envelope.

The forthcoming section is the analysis of the equivalent square pulse curve followed by the test methods. The methods discussed are of the categories of arrestment and momentum exchange. Both full-scale and model testing are considered.

An arrestment method is accomplished by decelerating a moving vehicle, by applying brakes or impacting a cable or rope, and by transferring the vehicle's kinetic energy to an energy absorbing or dissipating system. A momentum exchange method is accomplished by transferring energy from one body to another by impact of the two bodies.

EQUIVALENT SQUARE PULSE CURVE

An equivalent square pulse curve that will simulate the triangular pulse-duration-envelope criterion is dependent on the characteristics of the restraint system. A dynamic analysis will be undertaken to derive such an equivalent square curve that will simulate the equivalent restraint stroke responding to the triangular pulse envelope. Figure 69 is the equivalent curve, and Figure 70 shows an approximated restraint characteristic of the proposed energy absorber tie-down assembly.

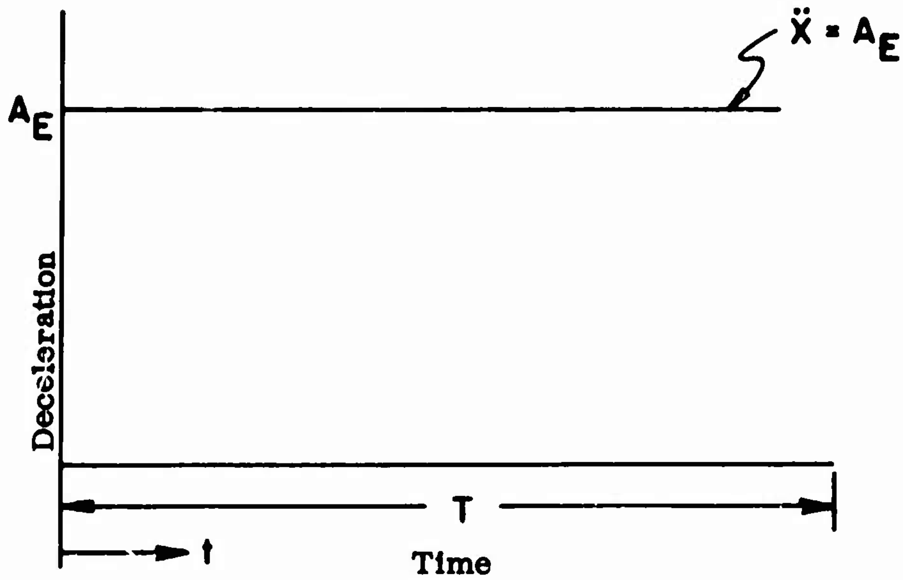


Figure 69. Equivalent Square Pulse Curve

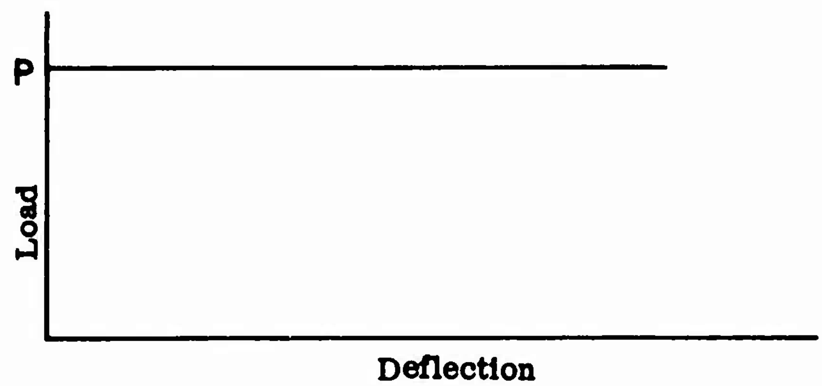


Figure 70. Approximated Characteristics of Restraint System

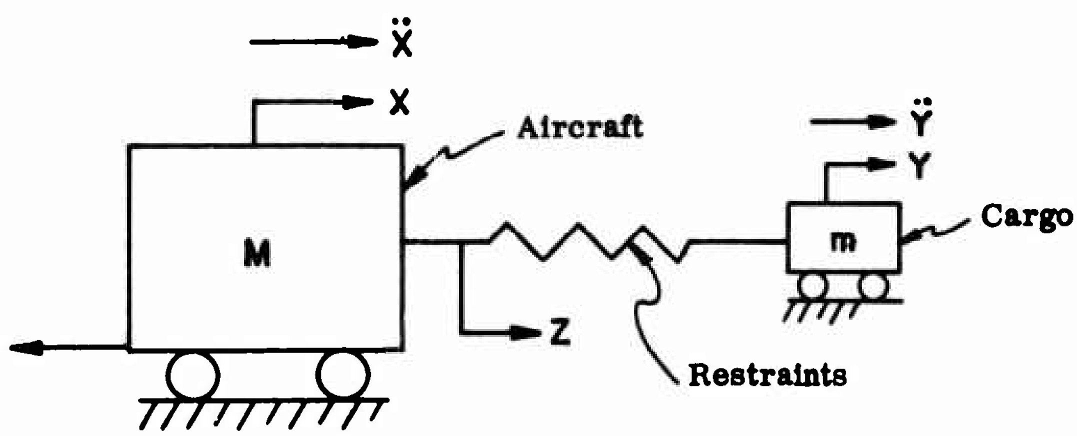


Figure 71. Simulated Aircraft, Cargo, and Restraints

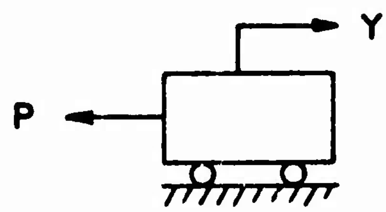


Figure 72. Free Body of Cargo and Restraint Forces

A_E = equivalent deceleration g's of vehicle
R = simulated crash acceleration pulse
P = restraint load
X = aircraft stroke
Y = cargo stroke
Z = restraint deflection
T_E = equivalent pulse duration
t = time interval

The deceleration of the vehicle is

$$\ddot{X} = A_E \qquad \text{for} \quad 0 \leq t \leq T_E$$

$$\ddot{X} = 0 \qquad \text{for} \quad t \geq T_E$$

From Figure 72,

$$m\ddot{Y} = -P$$
$$\ddot{Y} = Q^2$$

where

$$Q^2 = \frac{P}{M}$$

For $0 \leq t \leq T_E$

$$\dot{X} = V_0 + A_E t \tag{40}$$

where

$$\dot{X} = V_o \quad at \quad t = o$$

$$X = V_o t + \frac{A_E t^2}{2} \tag{41}$$

For $t \geq T_E$ (42)

$$\dot{X} = o$$

where

$$\dot{X} = V_T = o \quad at \quad t = T_E$$

$$X = V_o T_E + \frac{A_c T_E^2}{2} \tag{43}$$

where

V_o = impact velocity $= -A_E T_E$ (44)

X = aircraft velocity

Also,

$$\dot{Y} = V_o - Q^2 t \tag{45}$$

where

$\dot{Y}$ = cargo velocity

$$\dot{Y} = V_o \quad at \quad t = o$$

$$Y = V_o t - \frac{Q^2 t^2}{2} \tag{46}$$

From equations (40), (42), and (45), the plot of Figure 73 is delineated. It can be depicted from the figure that the cargo decelerates at a greater velocity than the vehicle; therefore, a stroke results between the two masses that is realized in the restraint system. The difference in area under the two curves results in the restraint deflection and is derived as follows:

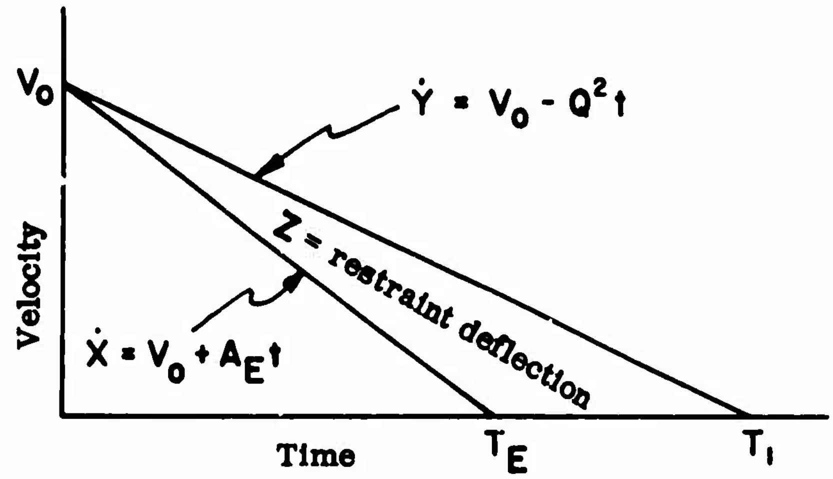


Figure 73. Vehicle and Cargo Velocity Versus Time

$$Z = \frac{V_0 T_1}{2} - \frac{V_0 T_E}{2}$$

when $t = T_1$; then $\dot{Y} = 0$

and $V_0 - Q^2 T_1 = 0$

$$T_1 = \frac{V_0}{Q^2} \tag{47}$$

Therefore,

$$Z = \frac{V_0^2}{2Q^2} - \frac{V_0 T}{2}$$

Using equation (44),

$$Z = \frac{A_E T_E^2}{2} \left[\frac{A_E}{Q^2} + 1 \right] \tag{48}$$

Rewrite the above equation with A_E as a function of T_E, Q, and A.

$$A_E = \frac{-Q^2}{2} \pm \frac{\sqrt{Q^4 + \frac{8 Q^2 Z}{T_E^2}}}{2} \tag{49}$$

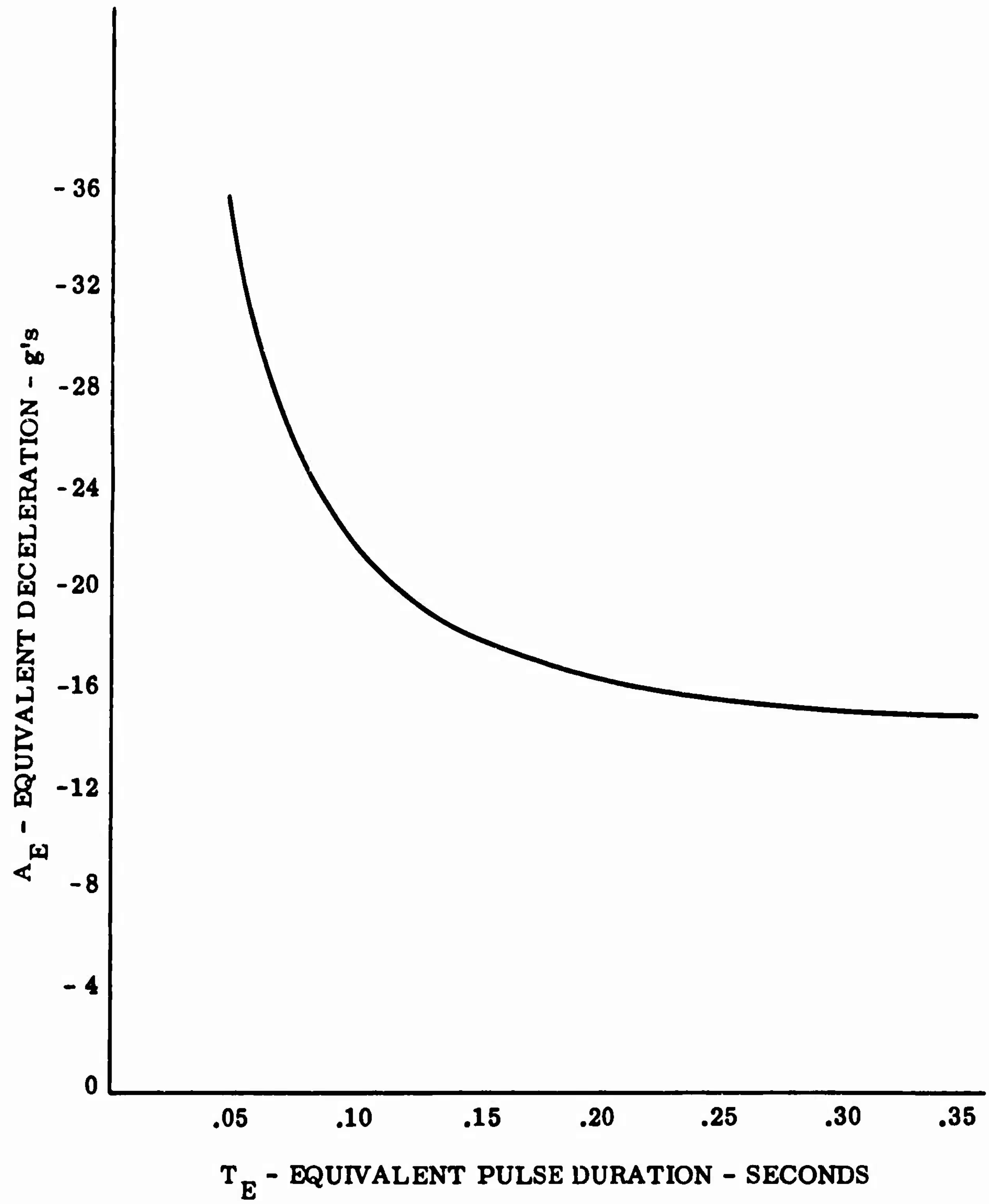


Figure 74. Equivalent Pulse Envelope Corresponding to Triangular Pulse Curve of -25 g's and .25-Second Duration, Which Results in 2 Feet of Restraint Deflection

By substituting G^2 = 15 g's and Z = 2 feet and various values of T_E in equation (49), the curve of Figure 74 is delineated. Any combination of A_E and T_E depicted from the curve will simulate the triangular pulse-duration criterion. It can be seen that the curve is asymptotic about its abscissa and ordinate. A large deviation in the high g value will result in a small change in the pulse duration; a small deviation in the lower g values will result in a large change in the pulse duration.

MODEL MOMENTUM EXCHANGE METHOD

In order to obtain dynamic similarity for a scaled test, it is necessary to know the ratio in which the various parameters of the test will have to be scaled for the test to be extrapolated to a reliable prediction of full-size performance. In addition, the scale factors will affect the extrapolation ratios themselves. To compute the scale factors it is assumed that restraint stroke is a function of the various test parameters listed below. These variables are collected into dimensionless ratios, and the stroke function is expressed in terms of these ratios. For dynamic similarity, it is necessary that the ratios have the same values for the test as for the prototype. This criterion of equal value then provides us with the necessary scaling ratios.

Second	T	Pulse duration
Pounds	P	Restraint load
Feet/Second	V_o	Impact velocity
$\frac{\text{Pounds-Second}^2}{\text{Foot}}$	m	Cargo mass
Feet/Second^2	g	Acceleration of gravity
Feet	S	Restraint stroke

A dimensionless ratio cannot be made from V_o, m, and g. However, one can be made using these and any other listed parameters. Using each of the remaining parameters, the following dimensionless ratios are possible:

1. T (pulse duration ratio)

$$\frac{(Tg)}{(V_o)}$$

2. P (restraint load ratio)

$$\frac{PT}{mV_o}$$

3.

$$\frac{Sg}{V_o^2}$$

The stroke function in terms of the dimensionless ratios is

$$S = \frac{V_0^2}{g} f \left[\frac{Tg}{V_0}, \frac{PT}{mV_0} \right]$$

If V_0 is reduced by a factor of $\sqrt{N}$ where $N<1$, then the scaled test parameters are as follows:

1. $(T)_{Test} = \sqrt{N}\ T$

2. $\frac{(P)}{(m)}_{Test} = \frac{P}{m}$

3. $(S)_{Test} = NS$

It can be seen from the above results that the ratio of restraint load to cargo mass is in the same proportion for the model and prototype. Also indicative is the fact that these parameters are independent of the other variables. The remaining parameters (restraint stroke, impact velocity, and pulse duration) are dependent on each other. The results of the above dimensionless analysis will be utilized in the following section.

1. Honeycomb Decelerator.

Precrushed paper honeycomb can be effectively used. Impacting precrushed honeycomb will result in a flat force-time curve response; therefore, the equivalent square curve pulse will have to be utilized.

Deceleration of the test vehicle is accomplished by impacting the test vehicle into precrushed paper honeycomb, which acts as the energy-absorbing medium. Figure 75 shows arrangement of the test method. An accelerator vehicle will accelerate the test vehicle to the desired initial speed. The test vehicle, with the aid of a bumper, will impact the precrushed honeycomb decelerator material, sustaining a predetermined force-time interval. The chart (Figure 76) depicts the characteristic flat force-time response curve obtained with honeycomb decelerators. The dotted line indicates the initial force that would be developed to initiate buckling of the core. By using precrushed material, as has been indicated above, this peak no longer exists, and a relatively constant force can be maintained during the test stroke.

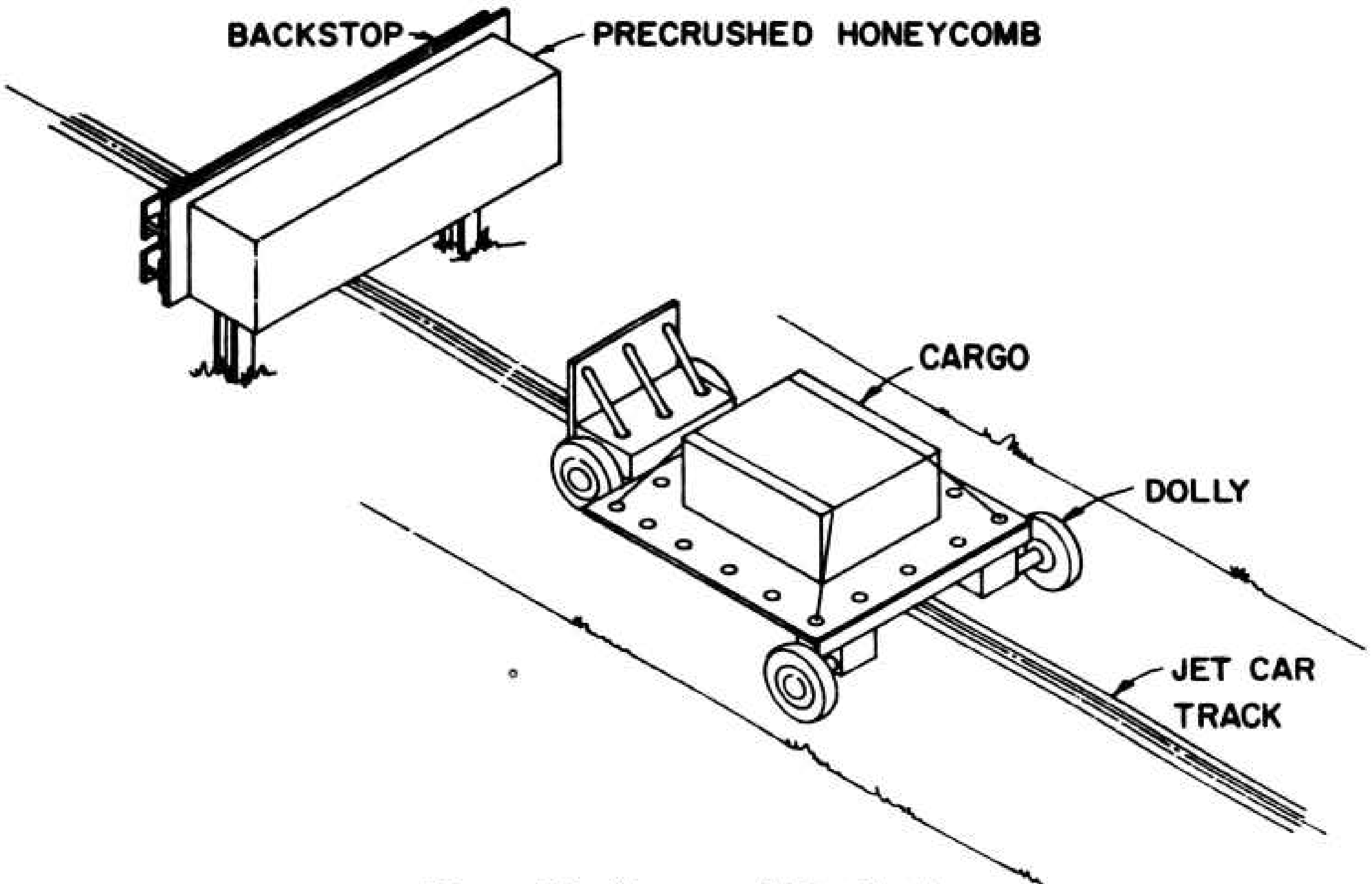


Figure 75. Honeycomb Decelerator

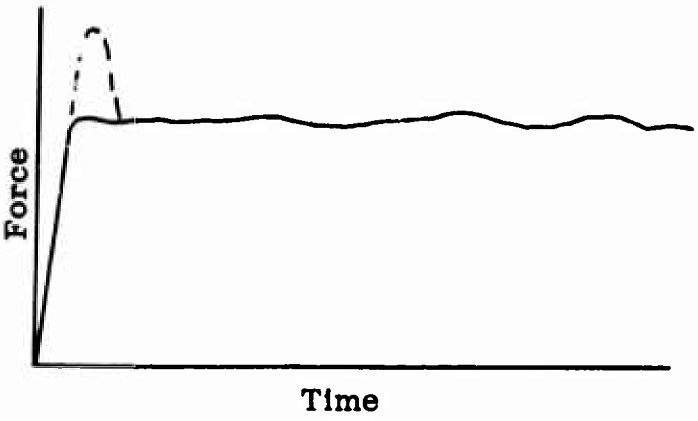


Figure 76. Force-Time Response Curve

Below is a table of the prototype versus model parameters. The scaled value N was selected at 1/6, and A_E and T_E were depicted from Figure 74.

TABLE XIII
PROTOTYPE VERSUS MODEL PARAMETERS

Prototype			Model		
Z	=	24 inches	Z	=	4 inches
A_E	=	15.7 g's	A_E	=	15.7 g's
T_E	=	0.25 second	T_E	=	0.102 second
V_o	=	127 feet/second (87 miles/hour)	V_o	=	51.5 feet/second (35.3 miles/hour)
$\frac{P}{m}$	=	15 g's = 483 feet/second^2	$\frac{P}{m}$	=	15 g's = 483 feet/second^2

The crushing force required is

$$F = -MA_E + P$$

P in this equation depends on the cargo mass. Selecting a cargo weight of 100 pounds, P = 1,400 pounds. This 1,400-pound restraint load will be divided among the number of restraints used. Assuming a 5,000-pound vehicle, the total crushing force is

$$F = -(5,000)(-15.7) + 1,400 = 78,500 \text{ pounds}$$

The allowable paper honeycomb core pressure will vary with each manufactured lot, depending on quality control, humidity, age, and other factors. Therefore, a trial run is necessary to obtain the specific value of the stock to be used at the time of use. It has been ascertained that an average allowable strength of 1/2-inch, 80-pound Kraft material is 6,000 psf. Then

$$\text{Honeycomb decelerator surface area} = \frac{78{,}500}{6{,}000}$$
$$= 13 \text{ square feet}$$

The honeycomb core thickness, S, is

$$S = -1/2\ A_E\ (T_E)^2$$

$$S = 1/2\ (-15.7 \times 32.2)\ (.102)^2$$

$$S = 2.63 \text{ feet (31.7 inches)}$$

Use a minimum of 42 inches of full thickness core.

The above analysis indicates that this test method is operationally feasible. Neglected in the analysis is the onset rate, which is the buildup to the 15.7-g level. A less rapid onset can result in a deviation from the restraint stroke. However, it is expected that such a deviation will be conservative. The onset rate will be determined during the trial run. Any reasonable deviation will be corrected for by recalculation. It is expected that the onset rate for this test method will be rapid and that no correction will be required. The onset rate could be controlled if necessary. This is accomplished by tapering the honeycomb to a preselected depth; thus, the cross-sectional area will vary with each incremental depth.

The decelerator is sensitive to the function of the square of the velocity. Because of this, the engaging velocities should be controlled within very small tolerances at the higher g levels. Operating a truck as the propulsion vehicle, within low velocity tolerances, is possible.

2. Shock Struts Decelerator

The kinetic energy of the impacting vehicle will be utilized to push a mandrel into which balls have been inserted along a preselected number of tubes, causing interference between the balls and tubes. A boom on the front end of the vehicle will act in a telescopic manner as the driving force on the mandrel. The balls will deform the tubes with sufficient force to develop enough work available from the kinetic energy of the vehicle. Figure 77 delineates the balls, mandrel, and struts, and Figure 78 shows the arrangement of the test method.

This method of deceleration is similar to the honeycomb method in that an equivalent square pulse curve will have to be utilized. The tubes will be swaged a prescribed length prior to testing to eliminate any initial peaking force; therefore, a constant force can be maintained throughout the required stroke. Also, the onset rate is rapid, therefore simulating the "equivalent" pulse curve onset rate. A truck, as the propulsion vehicle, can be utilized within allowable velocity tolerances.

Since the force-time response curve is flat (see Figure 76), the analogy between the model and prototype depicted in Table XIII and the previously calculated impacting force and stroke of 78,500 pounds and 31.7 inches is applicable.

In order to prevent a hammering effect at time of impact, a small cylinder of polyurethane will be bonded on the end of the boom. The polyurethane shock absorber will be utilized to maintain the desired onset rate.

All American Engineering Company has designed and proven the capabilities of the shock struts, and pertinent data can be found in bibliography reference 11.

FULL-SCALE MOMENTUM EXCHANGE METHOD

1. Honeycomb Decelerator

This method of testing is similar to the model analysis (page 125) except on a full-scale basis. A chart, Figure 79, was constructed to obtain the core thickness or stroke, S, and impact velocity, V, that are required for an "equivalent" square pulse curve. The curves are also based on an onset rate of 500 g's per second. As an example, for a pulse duration of 10 g's at 0.16 second, a total core thickness of 50 inches and an impact velocity of 53 feet/second are required.

For the characteristics of the honeycomb and all other pertinent aspects, see the section "Model Momentum Exchange Method" on page 124.

2. Spring Decelerator

For this method, two dead loads will be used as delineated in Figure 80. The test load vehicle, being the impact vehicle, will be of the minimum weight capable of simulating the cargo payload and designed to withstand the designated design loads. The other vehicle will be an existing dead load dolly, the property of All American Engineering Company. A set of springs in parallel, either air or gas such as dry nitrogen, with accumulators, will be attached to this vehicle, which will be at rest until impacted by the test vehicle. The impacting is to be accomplished by having a bumper on the test vehicle make contact with a hydraulic spring to relieve the initial shock load and obtain the onset rate, which is an integral part of the hardware that make up the kinematics of the piston rod (see Figure 80). After impact, both vehicles will be moving and, when the pressures

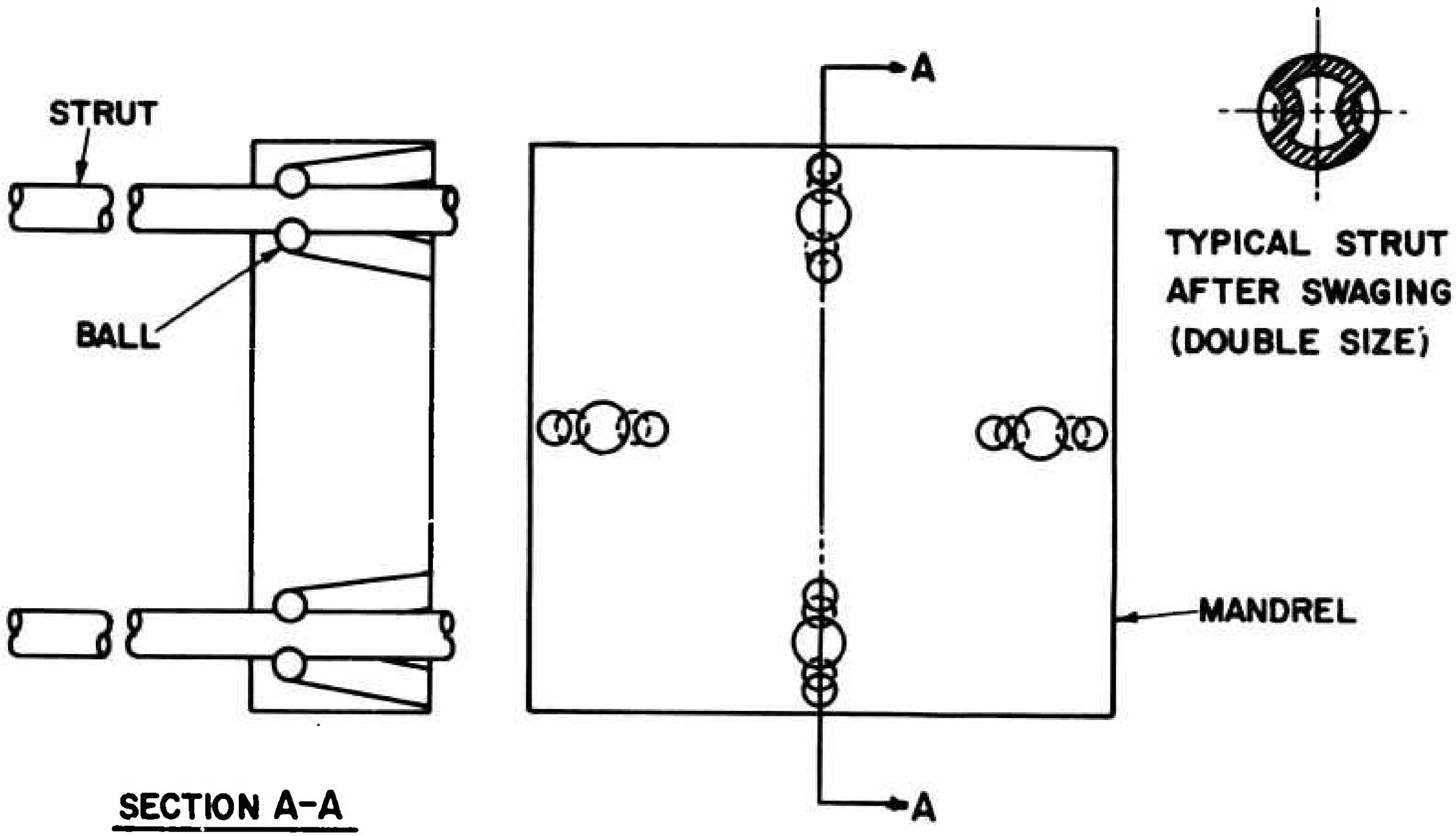


Figure 77. Shock Struts

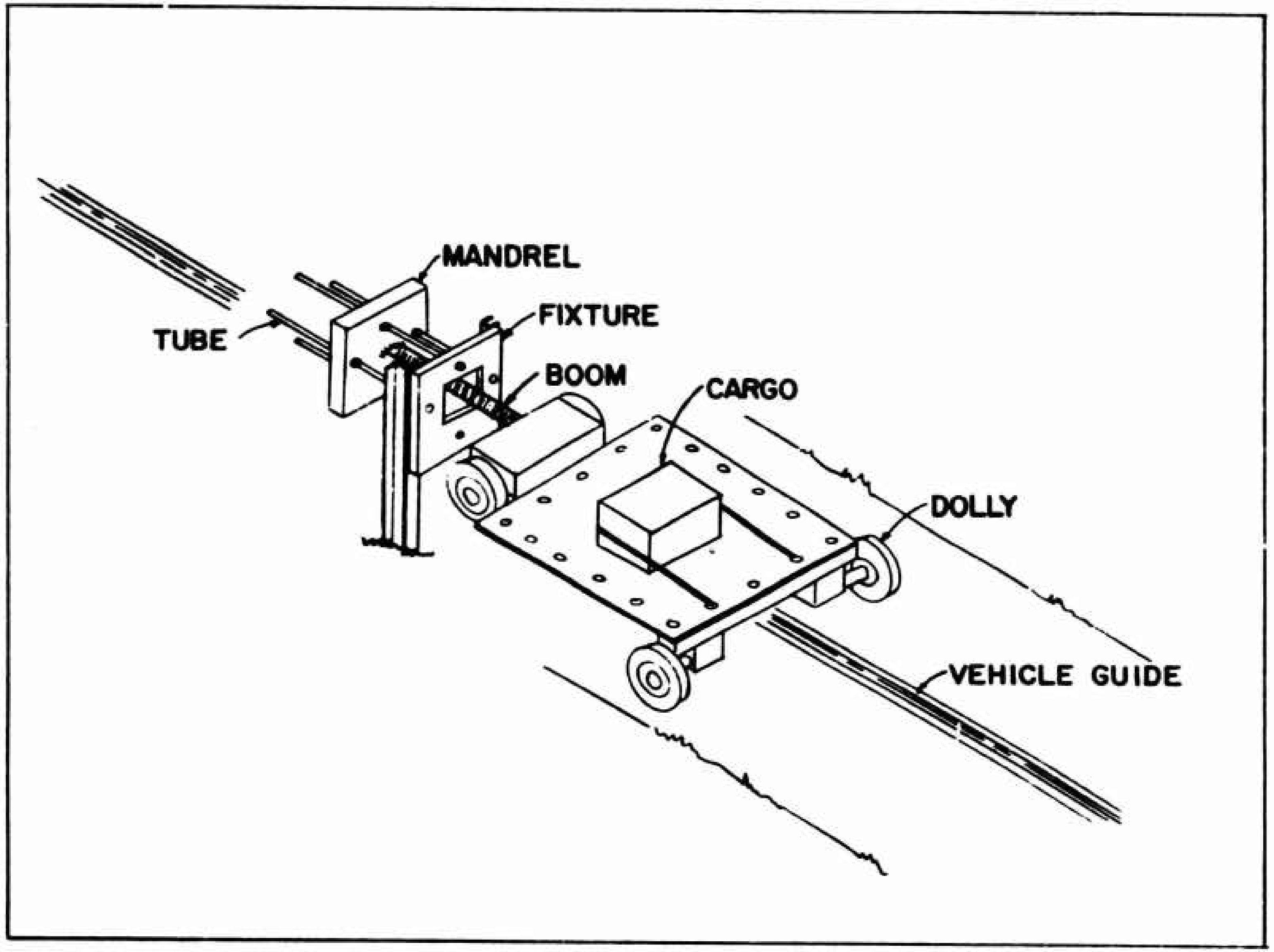


Figure 78. Shock Struts Decelerator

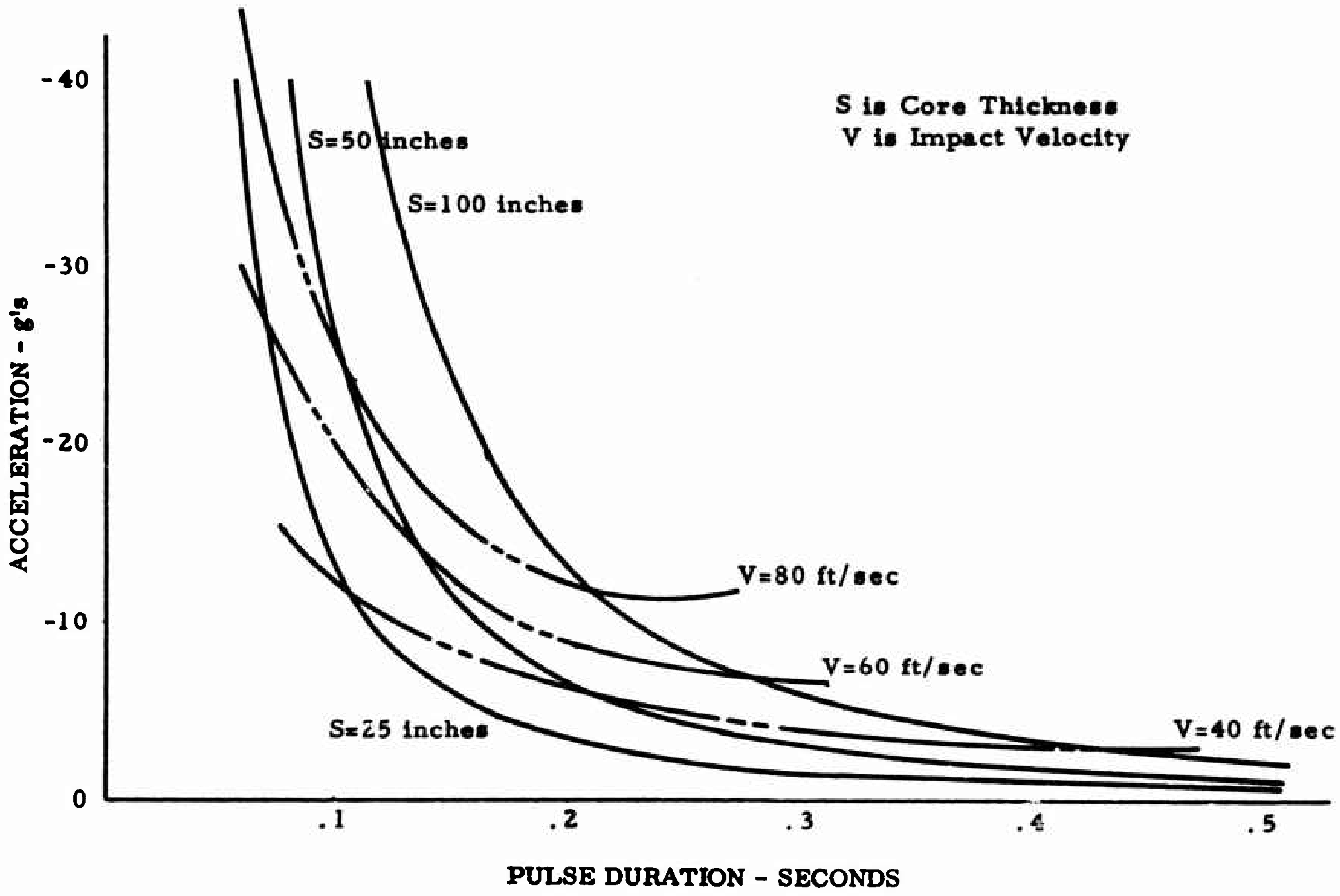


Figure 79. Equivalent Square Acceleration Pulse Limitations for Honeycomb Decelerator

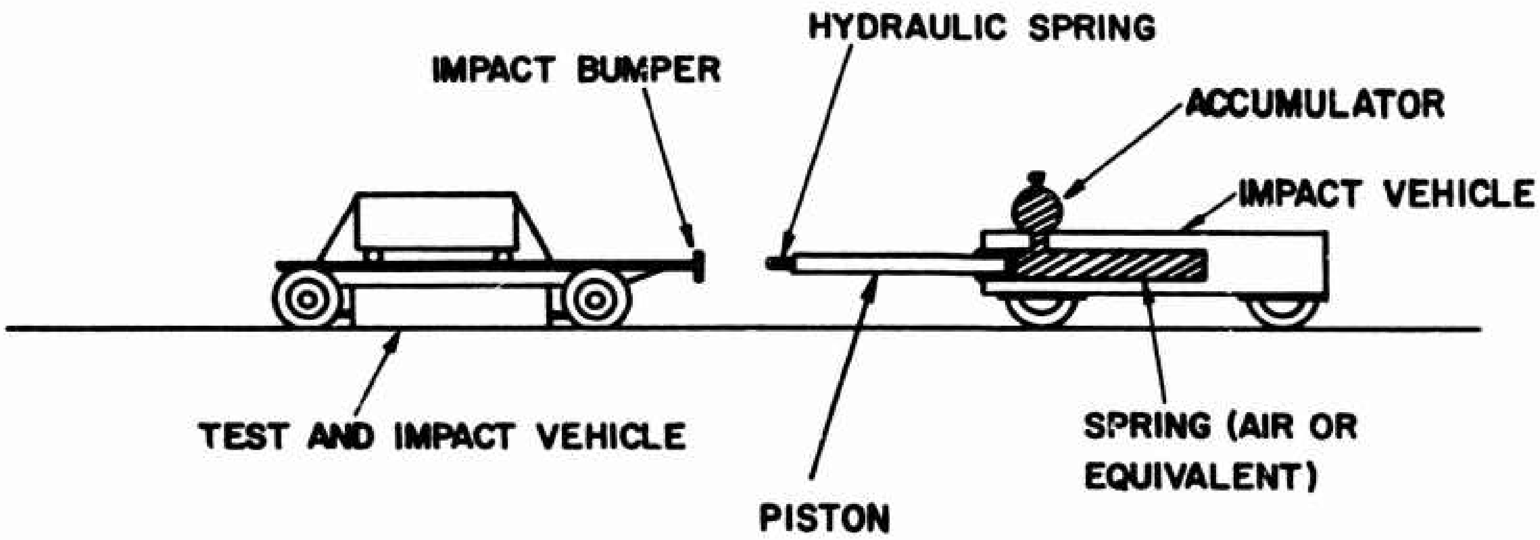
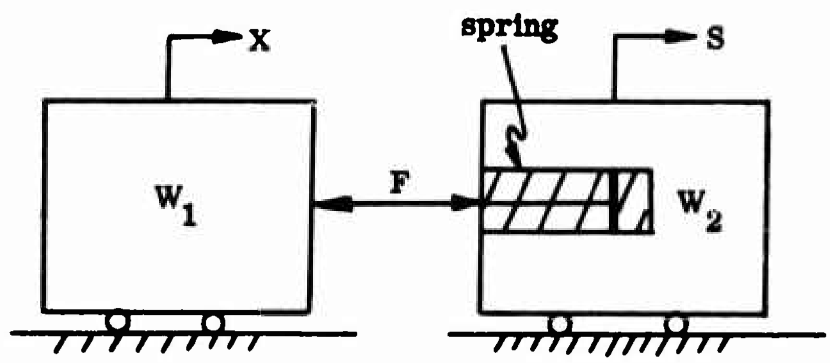


Figure 80. Spring Decelerator

in the springs exceed a predetermined value, the gas will tend to fill the accumulators. The accumulators will also release this gas when the cylinder stroke is reversed, the maximum stroke being accomplished when the piston is returned to its original position prior to impact. A constant force is then felt by both vehicles, resulting in a constant deceleration of the test vehicle. Therefore, a square acceleration pulse is imparted to the test vehicle.

The vehicles at the completion of the test will coast to a stop under their own resistance. The impact vehicle moves at a slower speed than the impacted vehicle. This is true since the cylinder stroke can be reversed during the test operations. As a precautionary measure, a net can be stretched vertically across the end of the runway to catch the vehicles if a complete stop is not imminent.

The equations that define the maximum cylinder strokes and maximum pulse times as a function of the vehicle weights, impact velocity, and spring force are shown below and are related to Figure 81.



Test and Impact Vehicle

Figure 81. Test Vehicle Impacting Stationary Dolly

Maximum cylinder stroke:

$$Z_{max} = \frac{1}{2g} \frac{V_o^2}{F} \frac{(W_1 W_2)}{(W_1 + W_2)}$$

Maximum pulse time:

$$T_{max} = \frac{2V_o}{gF} \frac{(W_1 W_2)}{(W_1 + W_2)}$$

where

V_o is impact velocity

F is spring force

W_1 is weight of test vehicle

W_2 is weight of stationary dolly

These equations were rewritten in order to plot acceleration versus maximum pulse time and cylinder stroke. The rewritten equations are

$$T_{max} = \frac{2V_o X}{A}$$

$$Z_{max} = \frac{V_o^2 X}{2A}$$

where

$$X = \frac{W_2}{W_1 + W_2}$$

A = acceleration

The plots of these equations are delineated in Figures 82 and 83. For simplicity, the vehicle weights were assumed to be equal, and an impact velocity range of 60 to 240 feet per second was considered. Interpolation between these curves would give results for any desired impact velocity. Also, it is possible to determine the maximum time duration and stroke for vehicles of different weights by substituting into X the ratio of W_1 and W_2 and multiplying this result by a factor of two and the readings depicted from the curves. As an example, consider an acceleration of 15 g's, associated with an impact velocity of 120 feet per second, and the test vehicle's weight, W_1, equal to one-half the impacted vehicle's weight, W_2. Then 15 g's refers to 0.33 second of pulse time duration and 10 feet of cylinder stroke as depicted from the figures.

3. Pendulum

The test dolly and its cargo are released in pendular fashion from a tower installed at All American Engineering Company's Georgetown facility. The pendular

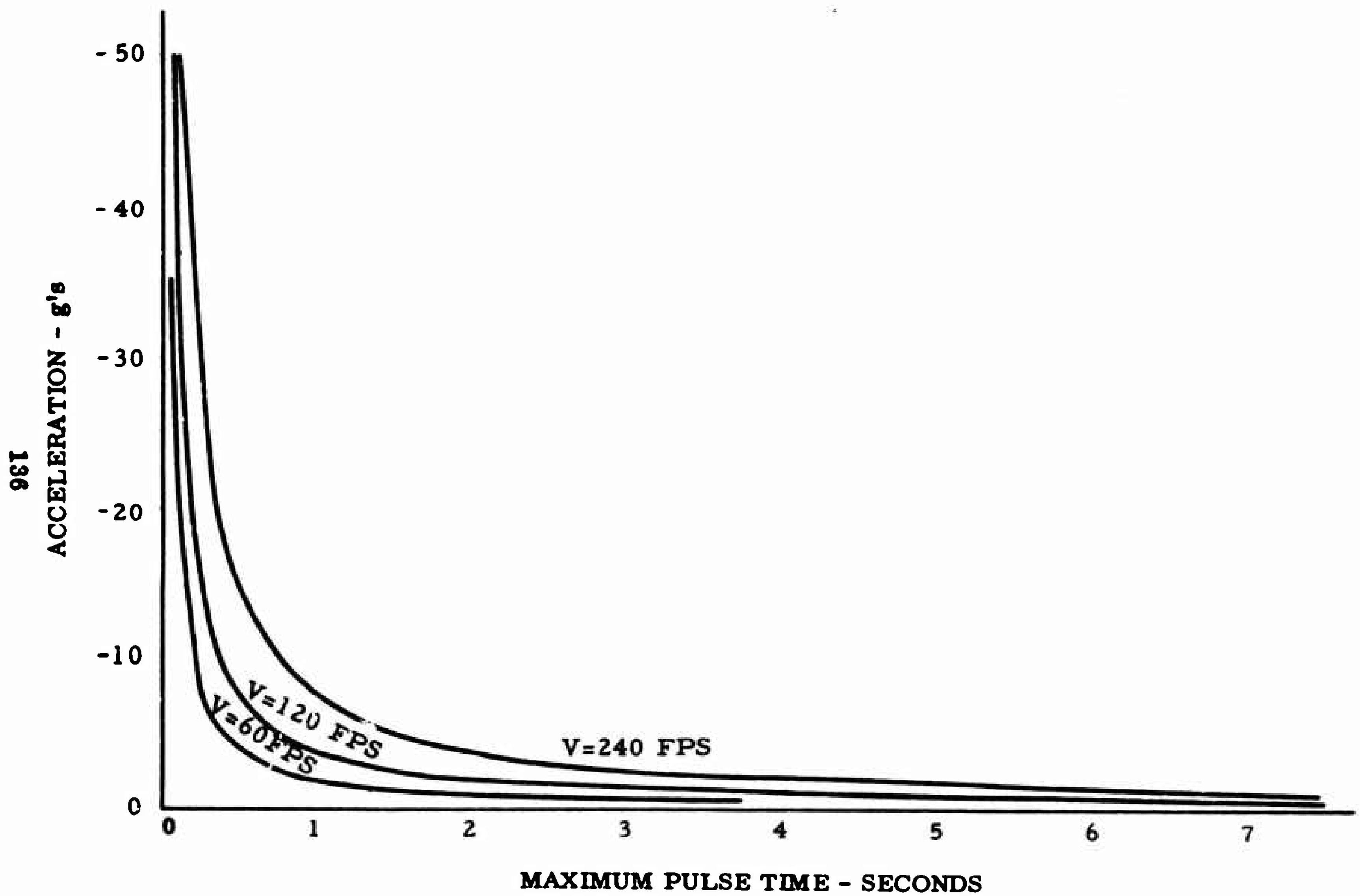


Figure 82. Maximum Pulse Time Limitations for Spring Decelerator

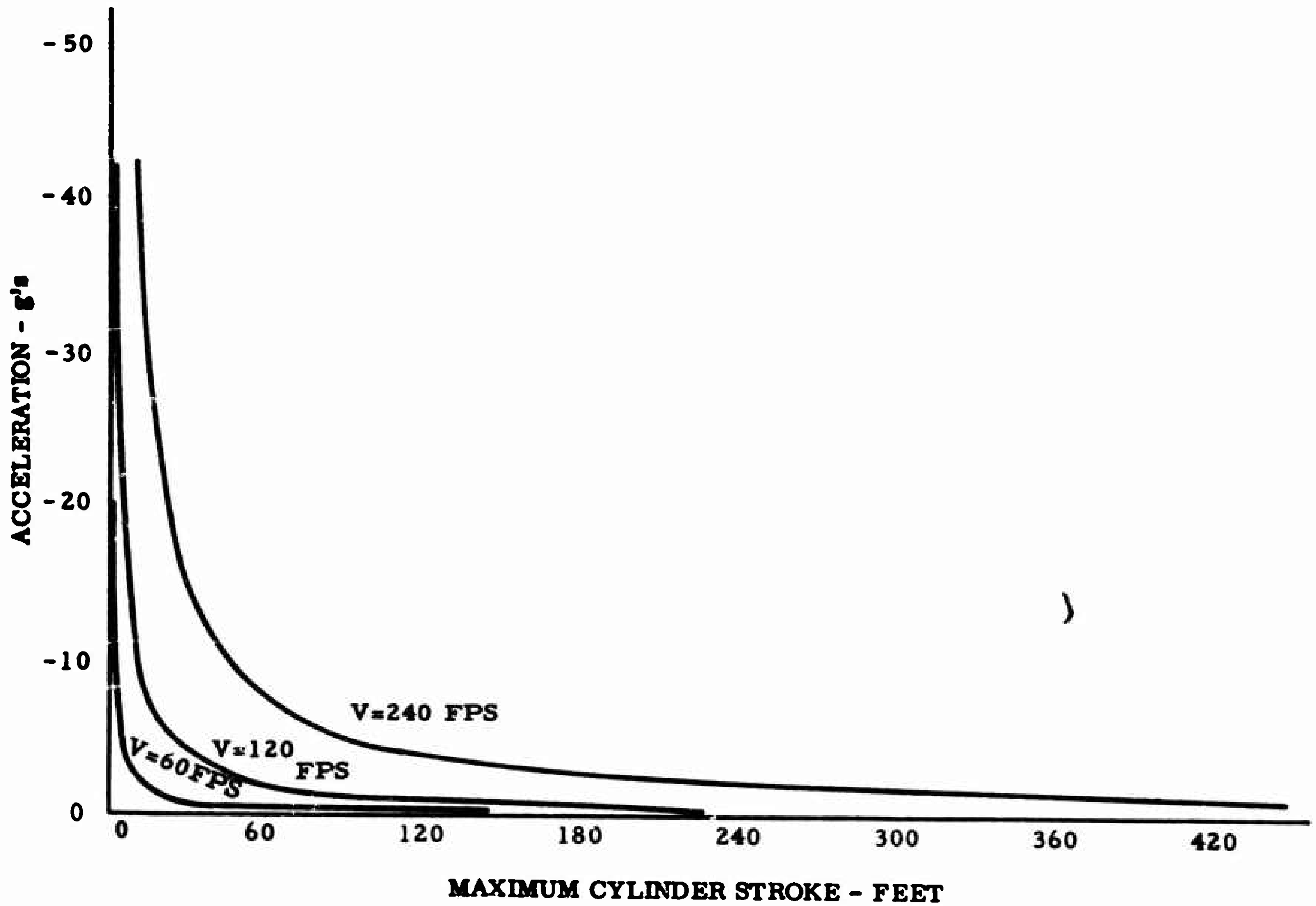


Figure 83. Maximum Stroke Limitations for Spring Decelerator

vehicle will impact an energy-absorbing device, such as the previously mentioned paper honeycomb (see Figure 84). The pendulum pivot is located at the top of the tower, with a winch to raise the test vehicle to a predetermined test position.

The limitation to this test method is the tower's height of 60 feet. If the test vehicle is released at an angle of 180 degrees, the maximum impact velocity is 88 feet per second. Based on utilizing 100 inches of honeycomb core thickness stacked together, the maximum square curve pulse-time duration is 14.5 g's at 0.19 second. These numerical values come from

$$V_o = \sqrt{2gh}$$

$$V_o = \sqrt{2 \times 32.2 \times 120} = 88 \text{ ft./sec.}$$

where

h = height of test vehicle;

and

$$S = 1/2\, V_o T$$

where

S = stroke (core thickness)

V_o = impact velocity

T = pulse time duration

$$T = \frac{2S}{V_o}$$

$$T = \frac{2 \times 100}{88 \times 12} = 0.19 \text{ second}$$

from

$$V_o = ngT$$

$$n = \frac{V_o}{gt}$$

$$n = \frac{88}{32.2 \times 0.19} = 14.4$$

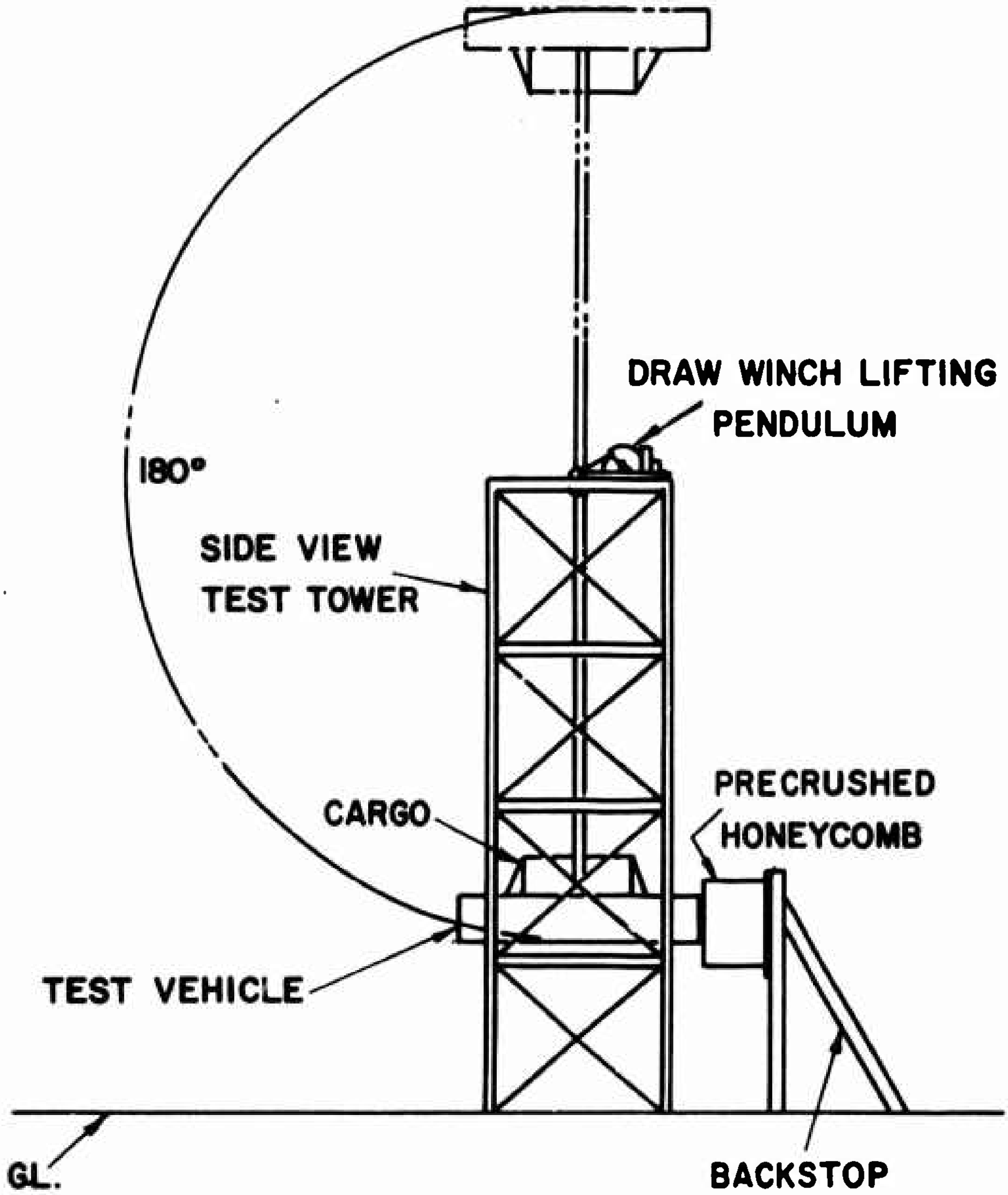


Figure 84. Pendulum

where

n = number of g's

Figure 85 depicts the accelerations and time intervals for a square pulse curve that are applicable to this test method for various pendulum positions and a multiple of honeycomb panels totaling 100 inches of core thickness. As an example, it is possible to decelerate the test vehicle at 5 g's for 0.32 second.

It is easy to visualize rotating the pendulum arm 180 degrees to the position in which the vehicle and its cargo are upside down. The pendulum arm is attached to the dolly, and the cargo will be held in place by its restraints. However, the cargo will oscillate as the pendulum is rotating prior to impact. These oscillations will have to be controlled so as not to produce an appreciable effect in the deflection of the restraints.

4. Guillotine

The object of this method is to drop the test vehicle from a tower on an energy-absorbing device such as paper honeycomb. The test vehicle is raised by a winch, and guide rails are used to insure that the dead load, when released, impacts the desired area. The tower installed at All American Engineering Company's Georgetown facility could be used with modifications. Figure 86 shows the guillotine method.

This method of testing is very limited in that the height of drop is 46 feet. At this height, a maximum impact velocity of 54 feet per second is applicable. Figure 87 depicts the acceleration-time intervals, for a square pulse curve, that can be preselected for testing.

The equations utilized to plot Figure 87 are

$$V = \sqrt{2gh}$$

$$S = 1/2\, VT$$

$$V = ngT$$

where

h = height at which test vehicle is released
g = acceleration of gravity, 32.2 ft./sec.
V = impact velocity
T = pulse time duration
n = number of g's
S = honeycomb core thickness

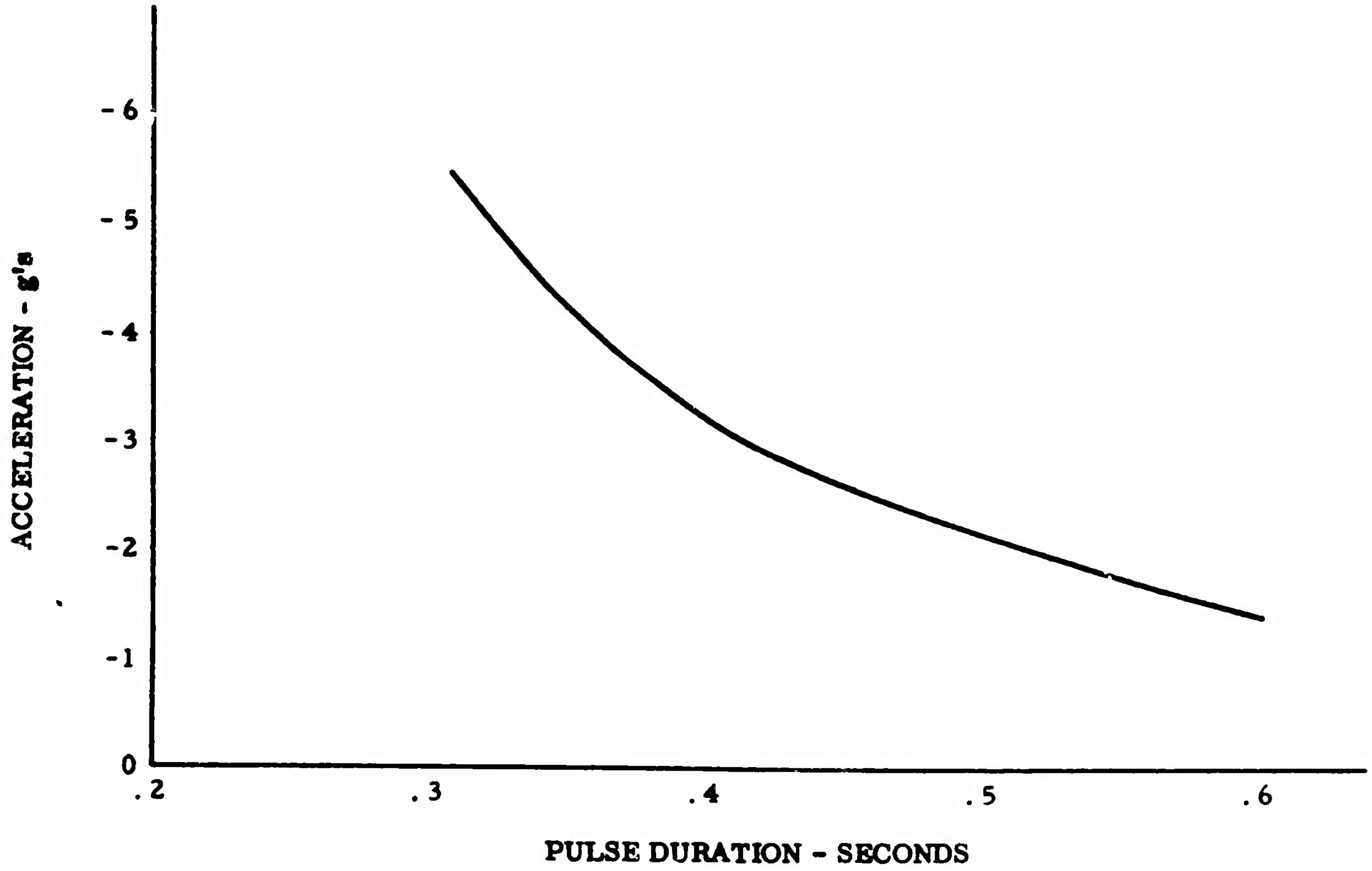


Figure 85. Limitations of Equivalent Square Acceleration Pulse Envelope for Guillotine Test Method

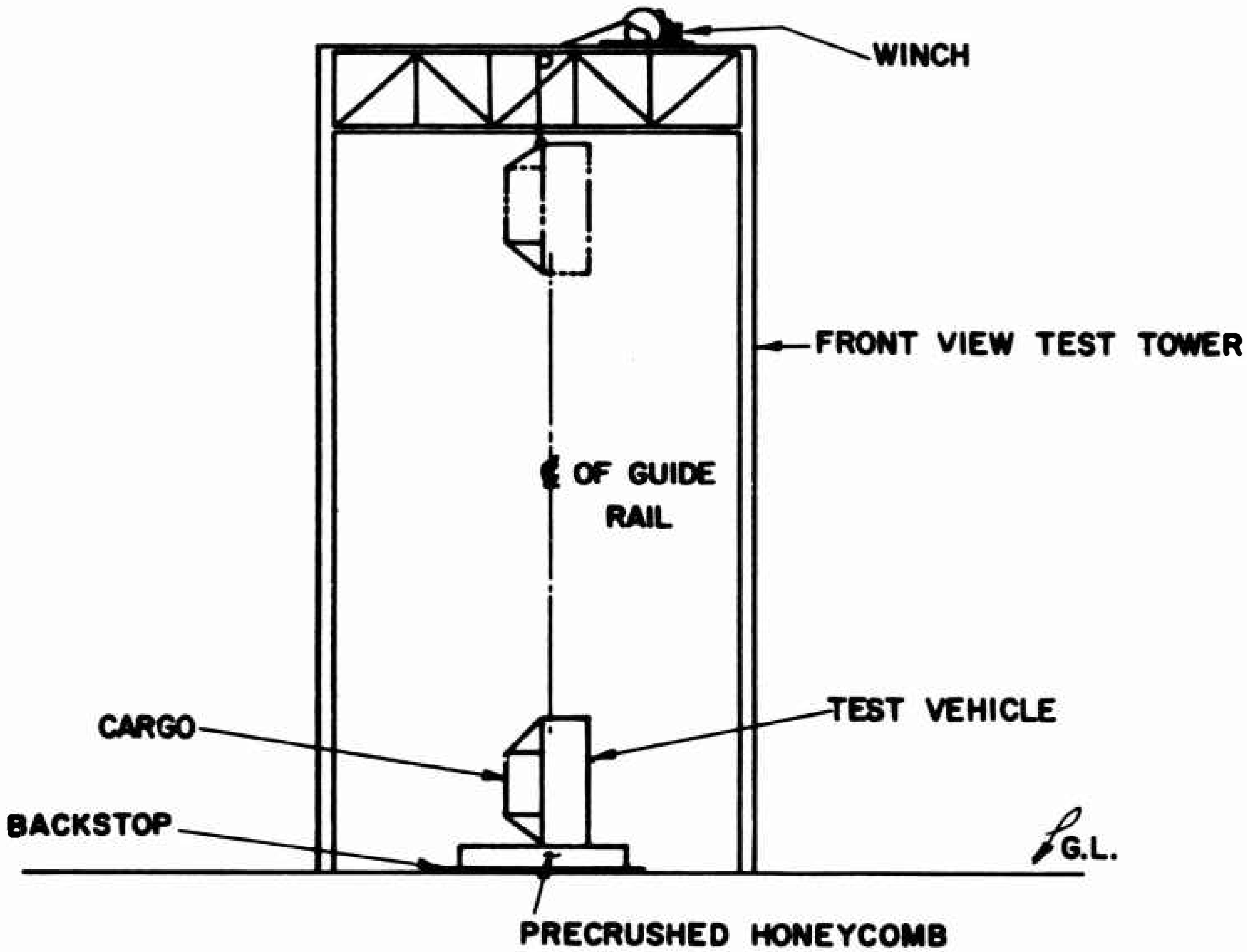


Figure 86. Guillotine

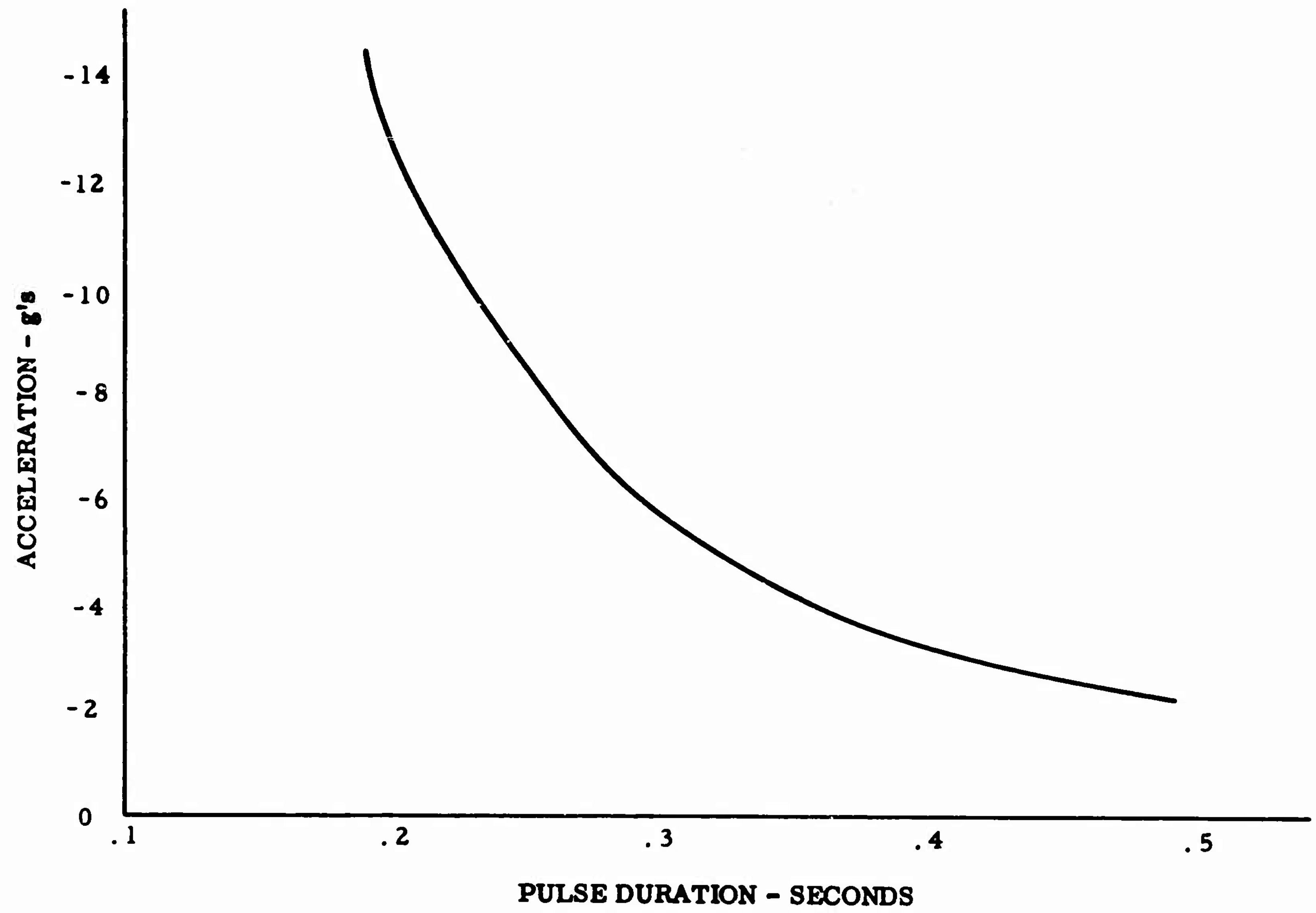


Figure 87. Limitations of Equivalent Acceleration Pulse Envelope for Pendulum Test Method

This method is capable of a maximum square pulse envelope of 5.5 g's at about 0.30 second, as depicted from the curve.

FULL-SCALE ARRESTMENT METHODS

1. Piccolo Tubes as Dynamic Water Brake

The information contained herein is obtained from bibliography reference 1 b.

The "piccolo tube" decelerator is a linear hydraulic energy absorber which dissipates the dolly energy by the displacement of water through multiple, sharp-edge orifices in the tube wall. Figure 88 shows the piccolo tubes and test vehicle during arrestment.

Depending upon the magnitude of deceleration desired, a choice of two or four decelerator tubes can be used in parallel. The dolly to be utilized with this test method will have hooks installed near the aft end of the dolly which engage cables leading into the tubes. Pistons at the ends of the cables form a tight seal with the tube walls and displace water. By programming the orifice hole sizes along the length of the tube, a square pulse curve and a constant pressure (and, thus, a constant retarding force) can be achieved throughout the arresting stroke. Also, the orifice hole sizes can be programmed to produce a triangular pulse curve throughout the arresting stroke.

The curve of Figure 89 depicts the possible limitations that can be accomplished for the equivalent square pulse curve. As an example, it is possible to obtain 8 g's for 0.36 second. The curves are based on a test dolly of 12,000 pounds, onset rate of 500 g's per second, and the following equations:

For longitudinal engagement, the equation

$$(V_o)\ \max = \frac{CF}{qE} \tag{50}$$

is applicable (see reference 1 c)

where

(V_o) max = the maximum impact velocity

C = speed of sound in cable

E = modulus of elasticity of cable

q = metallic cross section of cable

F = retarding force

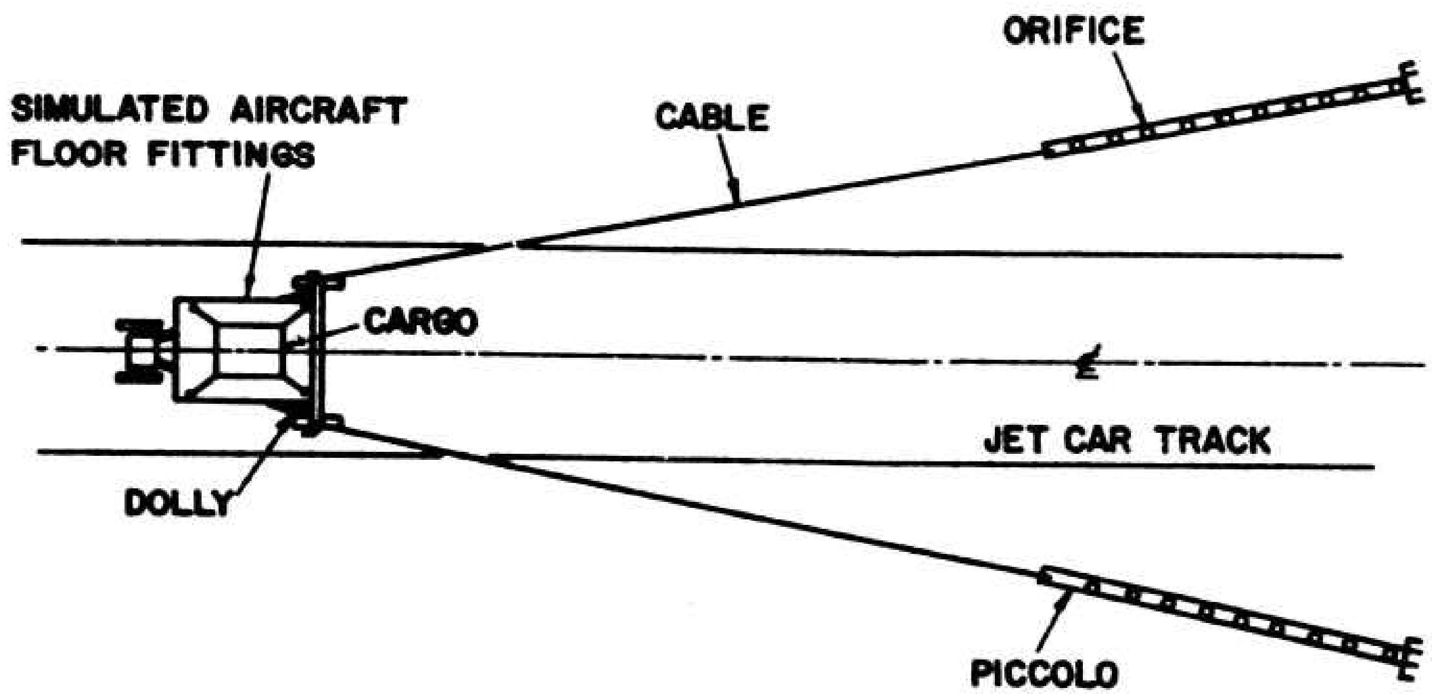


Figure 88. Piccolo Tubes as Dynamic Water Brake

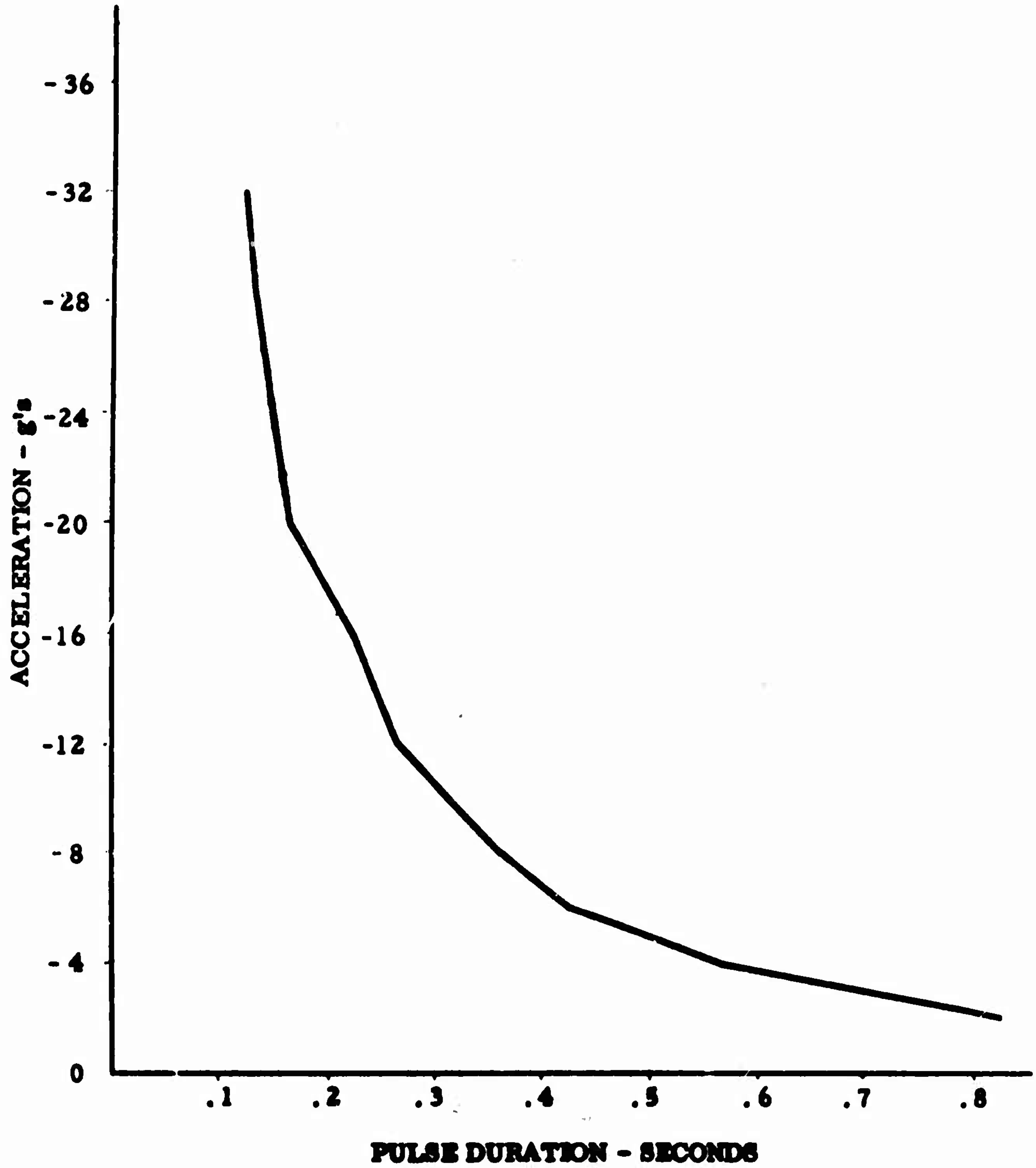


Figure 89. Limitations of Equivalent Square Acceleration Pulse Envelope for Piccolo Test Method

$$T = \frac{V_o}{ng} + \frac{n}{1,000} \tag{51}$$

$$S = 1/2\ ngT^2 \tag{52}$$

where

T = pulse time duration

n = number of g's

g = acceleration of gravity, 32.2 ft/sec.^{2}

V_o = obtained from equation (50) and $F = nW$

S = stroke in tubes

Equation (52) is an approximate equation that is utilized to check the tube stroke, which is limited to about 35 feet. The cable size will also be limited, based on the impact velocity that is capable of the limited stroke. An onset rate of 500 g's/second was considered.

The cable sizes considered were the minimum diameter sizes capable of withstanding the maximum retarding force and limiting stroke in the tubes. It is not economically feasible to use a different cable after every shot; therefore, depending on the number of test runs desired, a minimum number of cables should be used to accomplish a buildup in the incremental g load.

The decelerator is installed at All American Engineering Company's Georgetown facility. The company has conducted a crash resistant fuel system test for the Federal Aviation Agency in which increments of g were applied dynamically to a test vehicle bearing an aircraft wing section. See reference 1 b.

Engaging speed is quite critical in obtaining the proper deceleration level. The decelerator is sensitive to the function of the square of the velocity. Because of the variation in the local wind velocity, operation of the jet car within very limited velocity tolerances is difficult.

2. Stainless Steel Strap Decelerator.

The stainless steel strap decelerator is an energy absorber which dissipates the dolly energy by strain energy of the material. Figure 90 delineates the stainless steel straps and test vehicle during arrestment.

The lengths of steel straps to be used depend on the magnitude of deceleration and pulse time duration desired. An equivalent square acceleration pulse

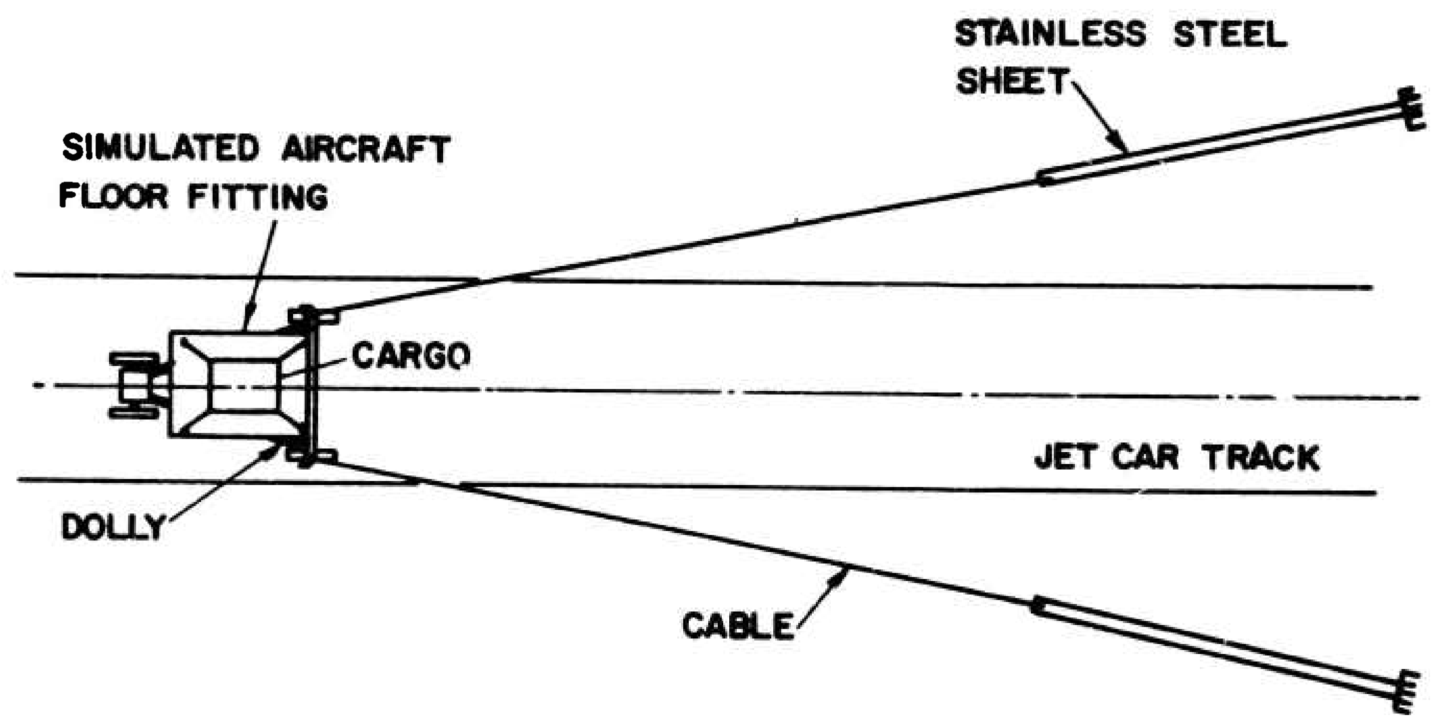


Figure 90. Stainless Steel Decelerator

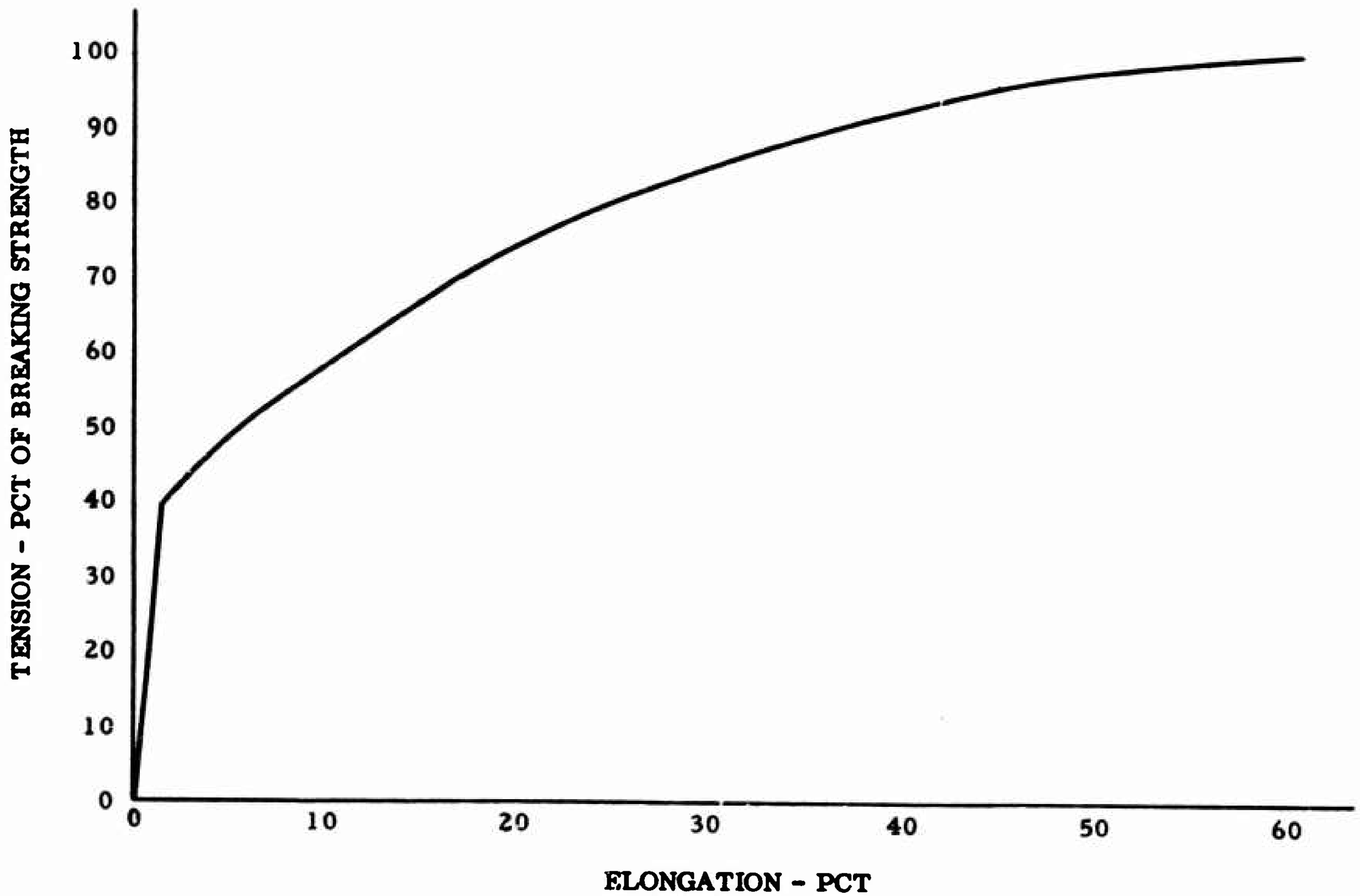


Figure 91. Stress Strain Curve for 302 Annealed Stainless Steel Sheet

envelope can be simulated by this test method. Similar to the piccolo test method, a dolly with hooks built near the aft end engages cables leading to the straps. The straps are elongated to a length not greater than 90 percent of their breaking strength, which is equivalent to 36 percent of the total elongation. Figure 91 represents the stress-strain curve for 302 annealed stainless steel sheets. The curve is in terms of percent breaking strength versus percent elongation.

The curve of Figure 92 represents the capability with respect to an equivalent square acceleration pulse curve. Also shown in the figure are the different steel lengths that are required for these limitations. As an example, a 16-g load would require a steel sheet 40 feet in length, while a 10-g load would require 60 feet. The curve is based on a test dolly of 12,000 pounds and a maximum impact velocity of 100 feet per second. The impact velocity was determined for equation (50). The curve was plotted from a computer analysis that involved the following equations:

$$T = M\frac{d^2x}{dt^2} \tag{53}$$

$$T = \left[1.122 - .722e^{\left(-3.255\frac{x}{1}\right)}\right] B.\,S. \tag{54}$$

where

T = tension in stainless steel sheet

M = mass of test vehicle

x = elongation of stainless steel strap

t = time occurrence of elongation

1 = total length of steel strap

e = log base e = 2.72

B. S. = breaking strength of steel

The second equation above is the equation of the curve of Figure 91.

The g loads presented in Figure 92 are average loads. The computer analysis shows a convex curve of positive slope (see Figure 93). The variation in g loading is more pronounced over the first 40 percent of the pulse curve, with the remainder of the curve being relatively flat. It is difficult for any test method to produce a flat acceleration pulse curve. A plot of a typical acceleration pulse curve is delineated in Figure 93. The average acceleration loads are a means of representing the desired pulse curve.

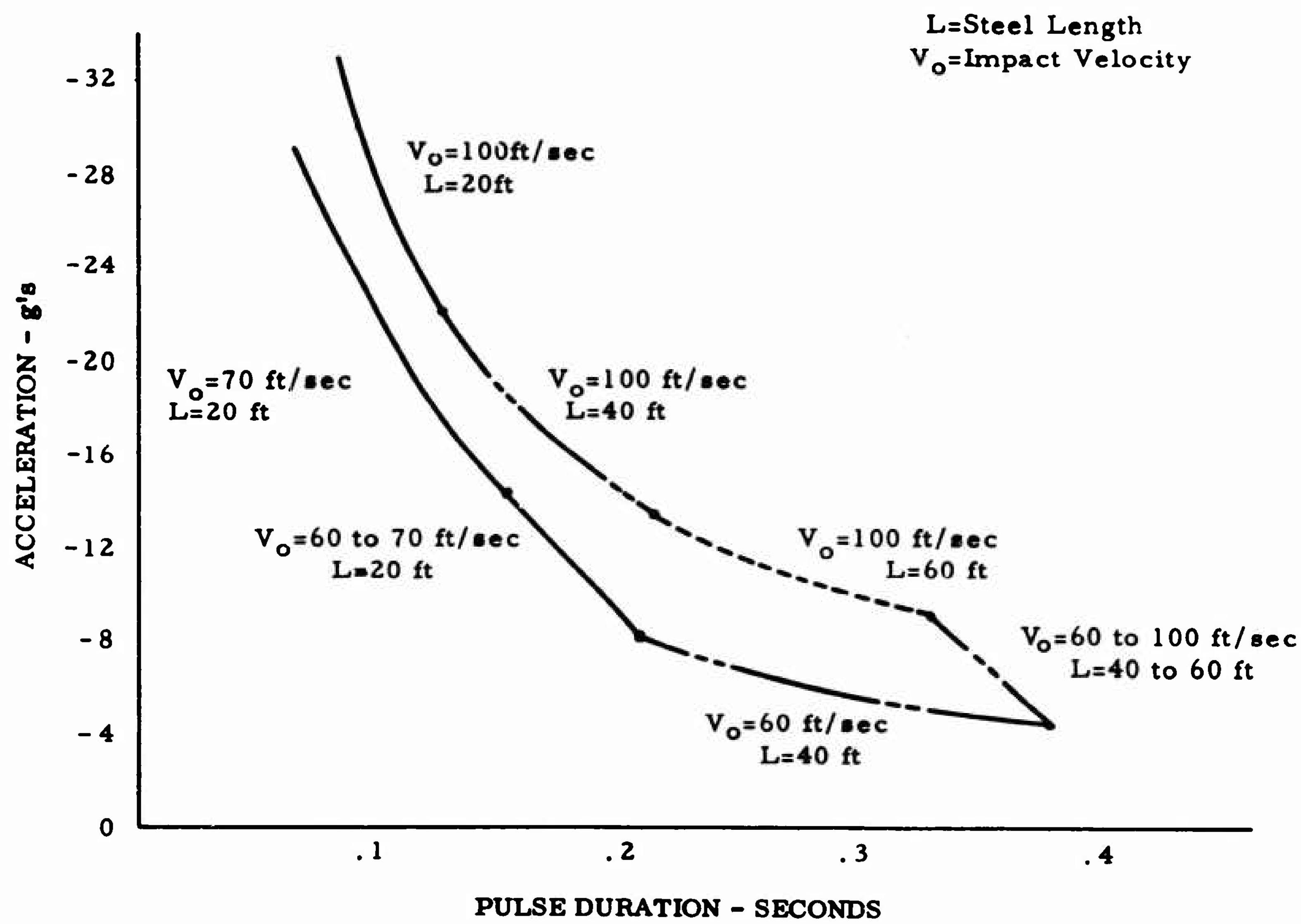


Figure 92. Limitations of Equivalent Square Acceleration Pulse Envelope for Stainless Steel Decelerator

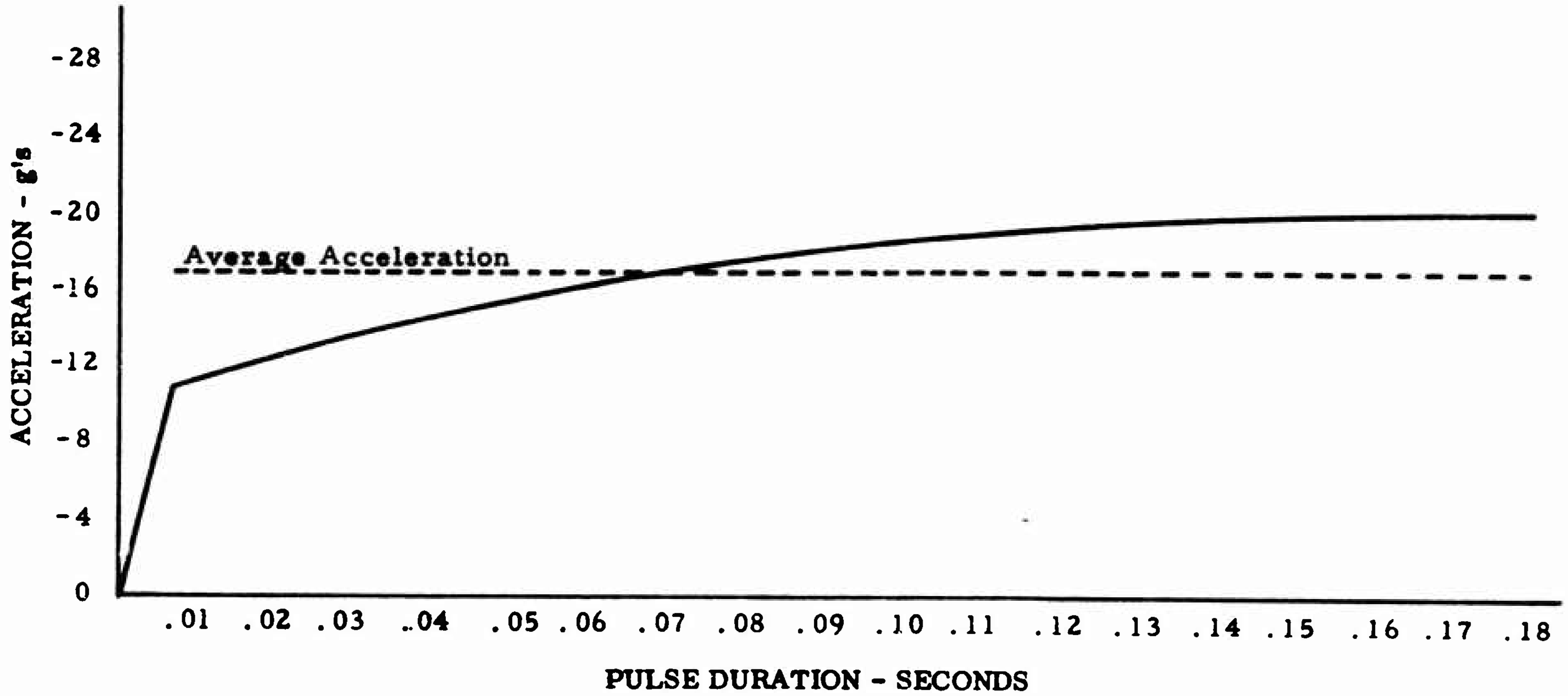


Figure 93. A Typical Acceleration Pulse Curve for Stainless Steel Decelerator

3. Friction Brakes

Friction brakes are utilized to stop the test vehicle, which is guided on a track (see Figure 94). The brakes are adjusted, prior to a run being made, to a predetermined clearance with the top flange of the guide track. At the end of the acceleration run, the guide track flange gradually thickens to a predetermined thickness, expanding the brakes. This provides the normal force necessary for braking. The brake rail and brakes are located at the Georgetown facility.

A curve, Figure 95, depicting friction coefficient versus vehicle velocity during the braking stroke, has been replotted from a previous brake test (Report N-313, bibliography reference 1 d) conducted by All American Engineering Company. From the curve, it can be seen that the friction coefficient increases from the moment the vehicle enters the brake section and peaks at the moment it comes to a stop. This follows the normal characteristic in that the coefficient of rolling friction is less than the static coefficient of friction.

The shape of the friction-velocity curve is important when the acceleration pulse curve that can be imparted to the test vehicle is established. The approximate equation of the friction-velocity curve is

$$u = 0.080 + 0.085e^{-0.0168(V)} \tag{55}$$

where u is coefficient of friction. This equation is approximate but accurate enough to be utilized analytically.

Vehicle deceleration and pulse duration can now be obtained as a function of velocity. The dynamic equation is

$$M\frac{dV}{dt} = -F_B \tag{56}$$

where

F_B = total force from brakes

M = mass of vehicle

$\frac{dV}{dt}$ = vehicle acceleration

but

$$F_B = uF_N$$

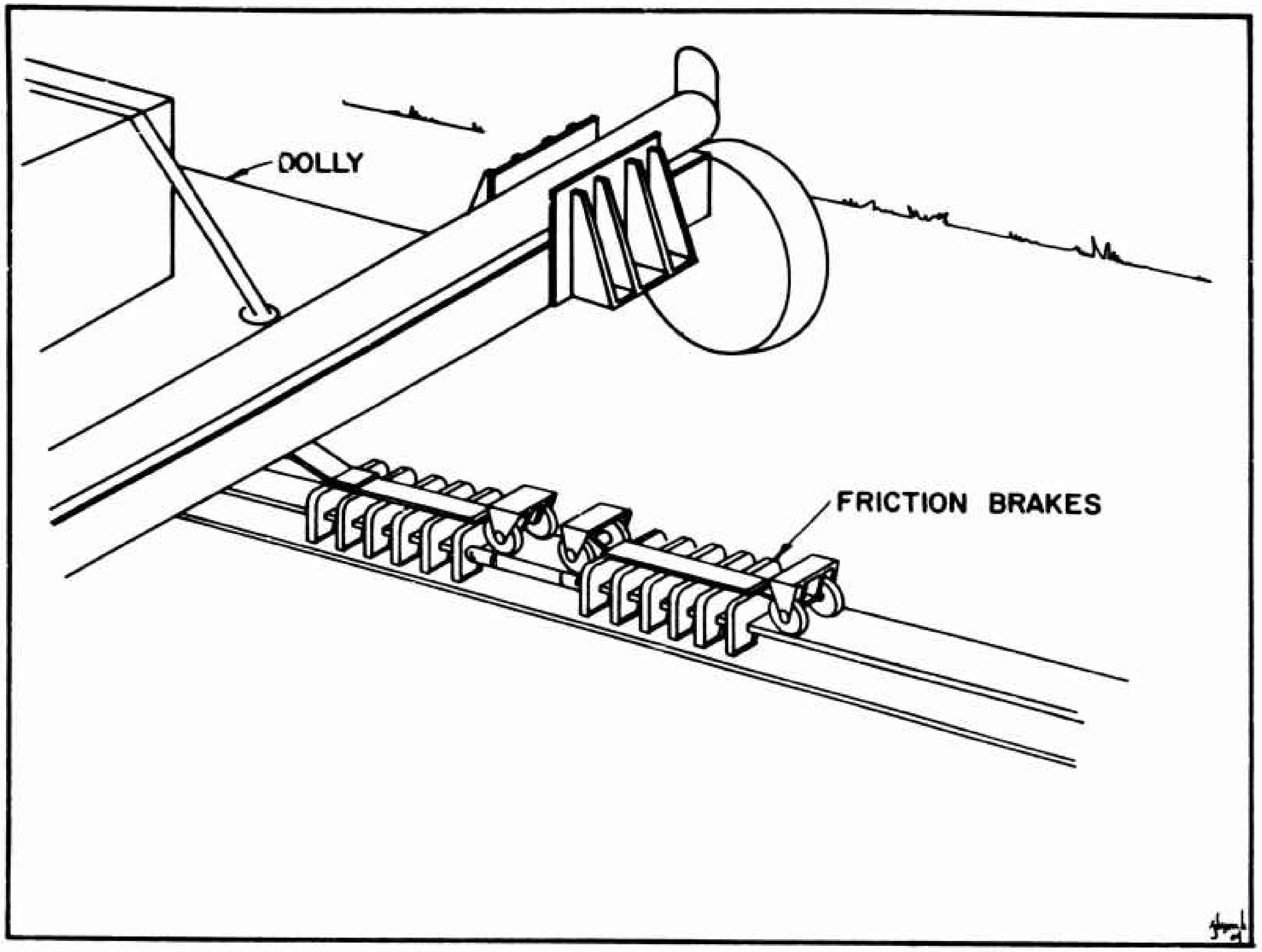


Figure 94. Friction Brakes

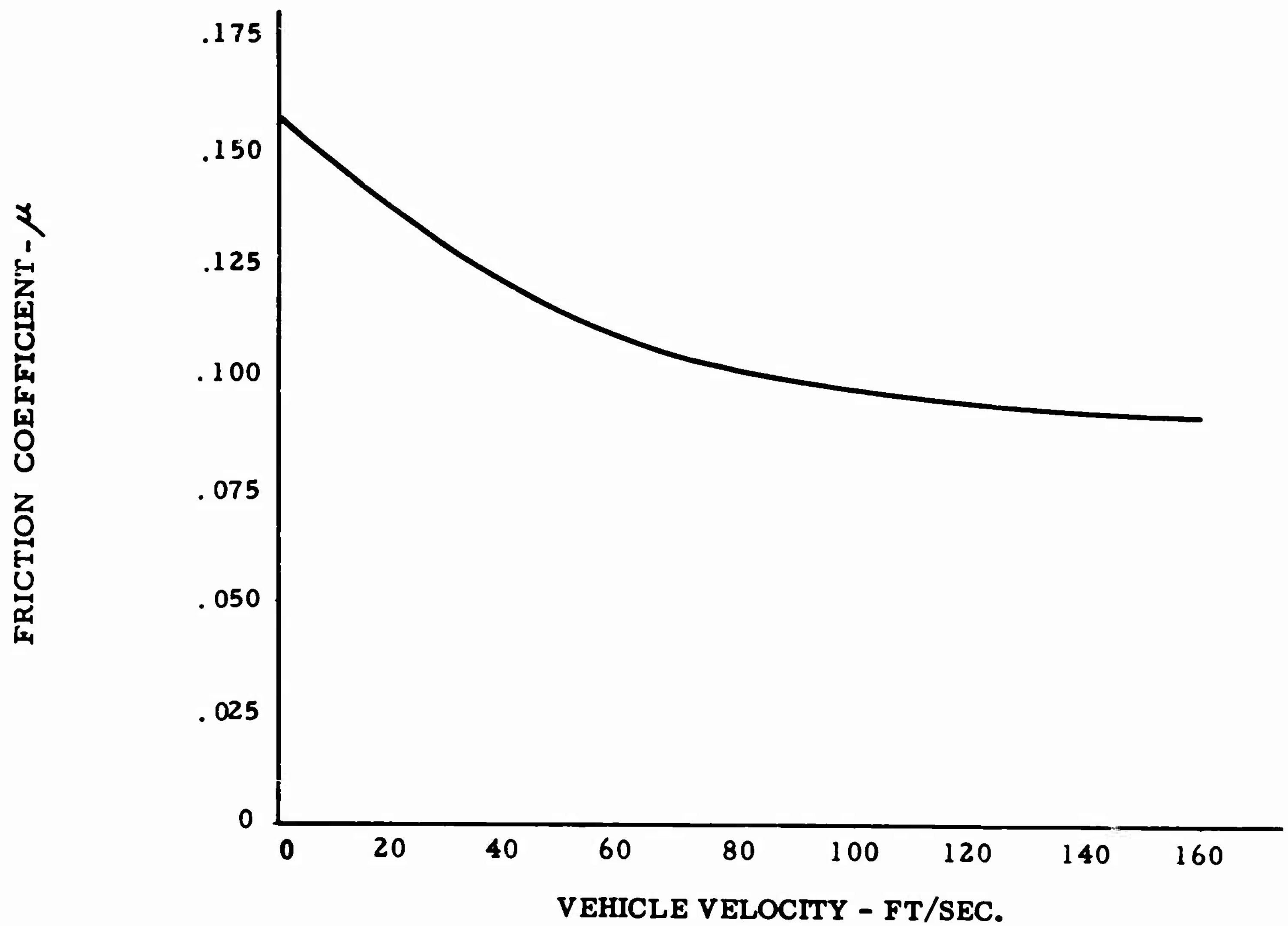


Figure 95. Friction Coefficient Versus Vehicle Velocity

where

$$F_N = \text{total normal force on brakes}$$

Then

$$M\frac{dV}{dt} = -uF_N$$

$$\frac{dV}{u} = \frac{-F_N}{m}\, dt$$

The solution to this equation is

$$t = \frac{M}{F_N} \frac{1}{.001344} \left[-.0168(V) - \ln\left(.080 + .085e^{-.0168(V)}\right)\right]_{V_o}^{V_n} \tag{57}$$

where

V_n is final velocity of vehicle

V_o is initial velocity of vehicle

Also with the vehicle's acceleration, $A = \frac{dV}{dt}$,

$$A = -u\frac{F_N}{M}$$

Using equation (55),

$$A = \frac{-F_N}{M}\left[.080 + .085e^{-.0168(V)}\right] \tag{58}$$

Equations (57) and (58) were utilized in plotting the friction brakes' limitations (Figures 96 and 97) and are based on the expected test vehicle and cargo weights. Figure 96 neglects the weight of the brake units, while Figure 97 includes the brake weights which clearly shows the effects of the increase in weight on the system. The charts are also based on an initial braking velocity of 230 feet per second, 8 brake units, and the maximum brake spring setting.

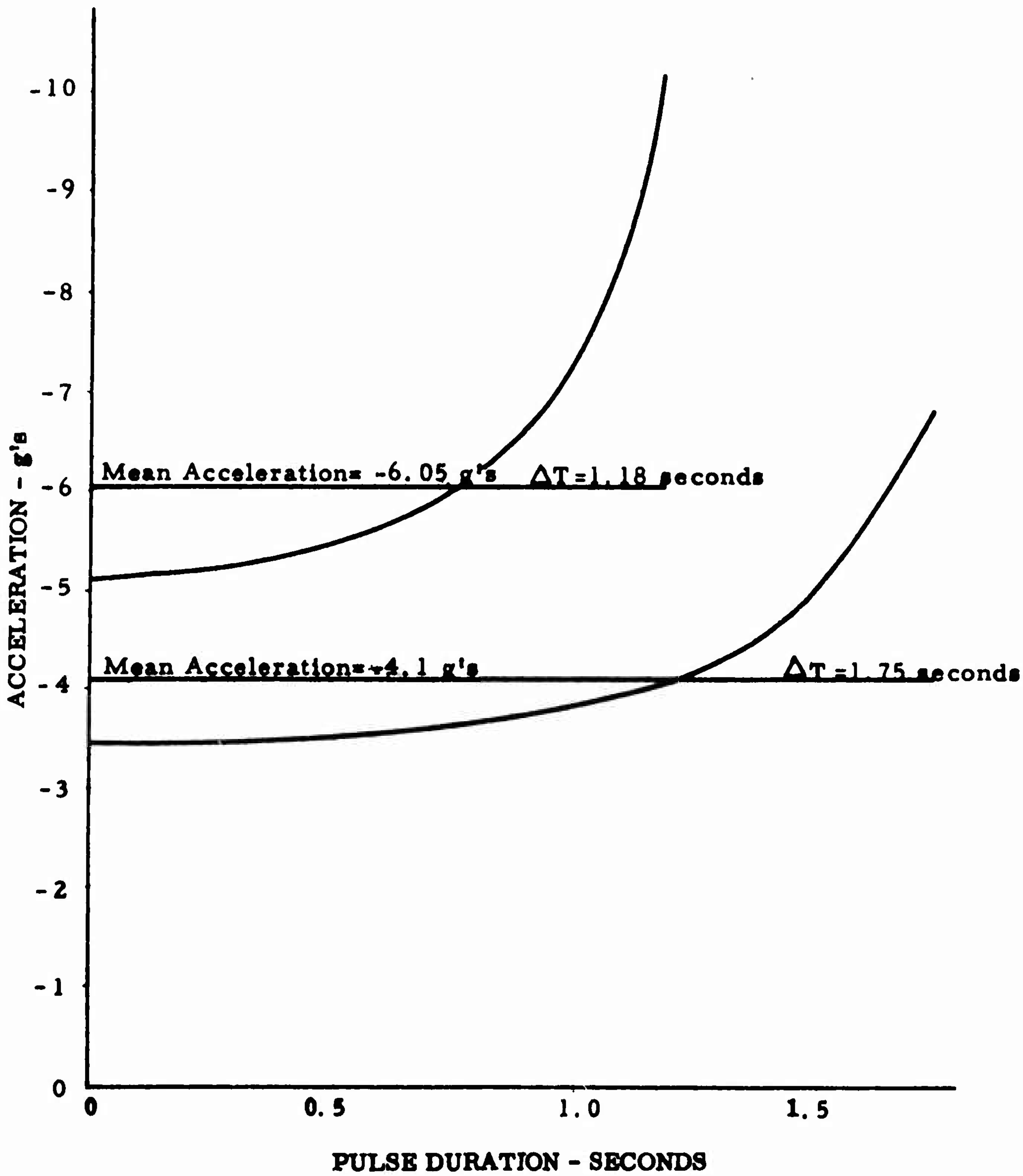


Figure 96. Pulse Duration Envelope Limitations for Friction Brakes Neglecting Weight of Brakes

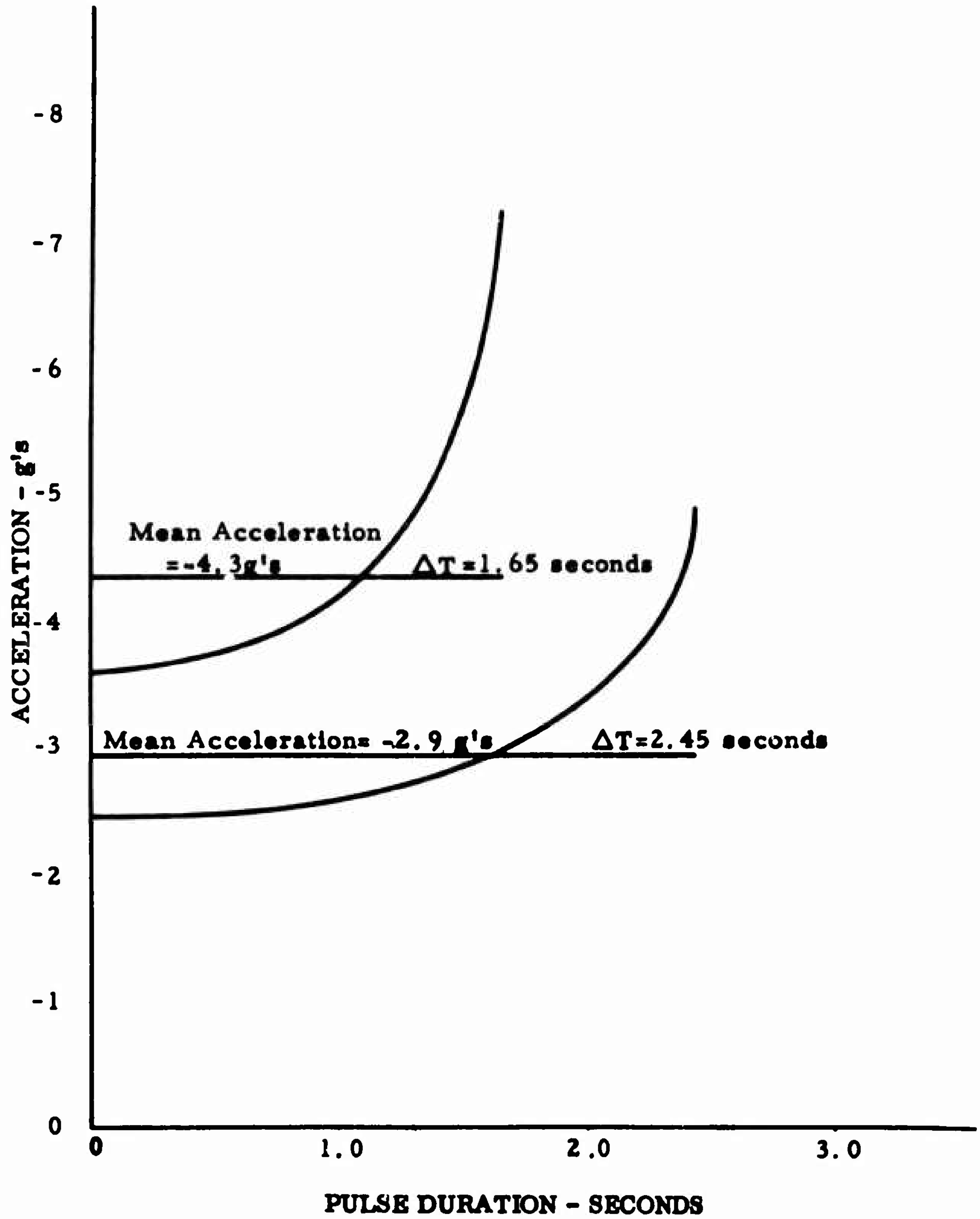


Figure 97. Pulse Duration Envelope Limitations for Friction Brakes Including Weight of Brakes and Test Vehicle

For any significant time duration, the g loads are limited to low values and are not as applicable as one might expect. The two curves represent the limits that may be obtained based on the extreme brake spring settings. Higher deceleration loads may be obtained by decreasing the initial brake speed, but the time durations decrease rapidly with increased g load as depicted from the curves. Increasing the number of brake units for higher g's is not feasible because of the resulting decrease in time duration.

As noted from the curves, an average constant g load occurs immediately after braking for a limited time interval and then increases rapidly. This maximum g load occurs at the instant the vehicle stops and is the limiting factor in the design of the test vehicle. It is possible to obtain constant g for longer time durations by varying the thickness of the lining bonded to the guide track flange. Halting the vehicle can be accomplished by the brakes at the location on the track that is outside the test area. This alternative of utilizing friction brakes requires reworking the brake lining already bonded to the guide rail. A thorough investigation of the effects of heat on the non-prismatic brake lining is also required.

Limitations of the pulse duration envelope are considered to be over the initial g load, that is, relatively constant. As can be seen from the curves, the pulse time duration that is delineated for an equivalent square pulse shape is for short time intervals.

Another test conducted and documented (reference 1 d) by All American Engineering Company delineates for a specified type brake lining a friction-velocity curve of higher friction coefficient than the curve shown in Figures 96 and 97. This curve has a small negative slope, and the test data is recorded for a maximum velocity of 50 feet per second. Because of the lack of test data at higher speeds, this limitation must be adhered to. The maximum square acceleration pulse is calculated as follows, assuming a flat friction-velocity curve:

Maximum acceleration (n):

$$n = \frac{uF_N}{W}$$

where

u = 0.21 as depicted from reference 1 d

F_N = 500,000 pounds (8 brake units)

W = 16,800 pounds (weight of vehicle and brakes)

$$n = \frac{(.21)\ (500{,}000)}{16{,}800}$$

$$n = 6.25 \text{ g's}$$

Maximum pulse time duration (T_{max}):

$$V_o = -AT_{max}$$

$$A = \frac{-uF_N}{M}$$

$$T_{max} = \frac{MV_o}{uF_N}$$

where

$$M = \frac{16,800}{32.2} = 522 \frac{\text{pounds (second)}^2}{\text{feet}}$$

V_o = impact velocity of 50 feet per second

$$T_{max} = \frac{(522)\ (50)}{(.21)\ (500,000)}$$

$$T_{max} = 0.25 \text{ second}$$

The maximum pulse duration envelope is then 6.25 g's at 0.25 second.

ITEMIZED COST ESTIMATE OF PROPOSED TEST FACILITY

GFE Equipment

The following items are considered to be Government-furnished equipment:

1. Instrumentation equipment - Borg-Warner magnetic tape recorder.
2. Dead load dolly - D.L. 100, owned by U.S. Navy: designed and built by AAE.
3. Aircraft floor tie-down fittings.

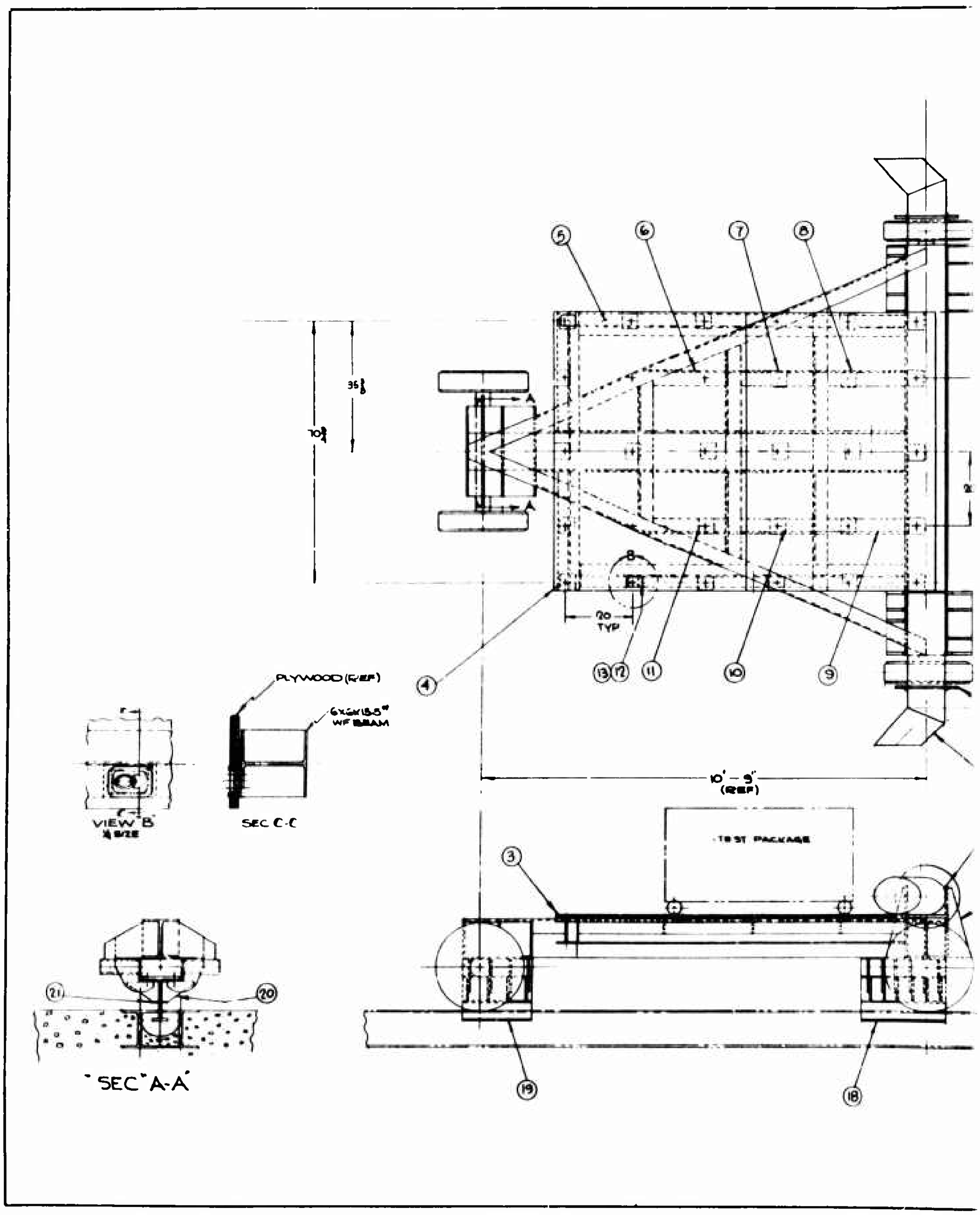
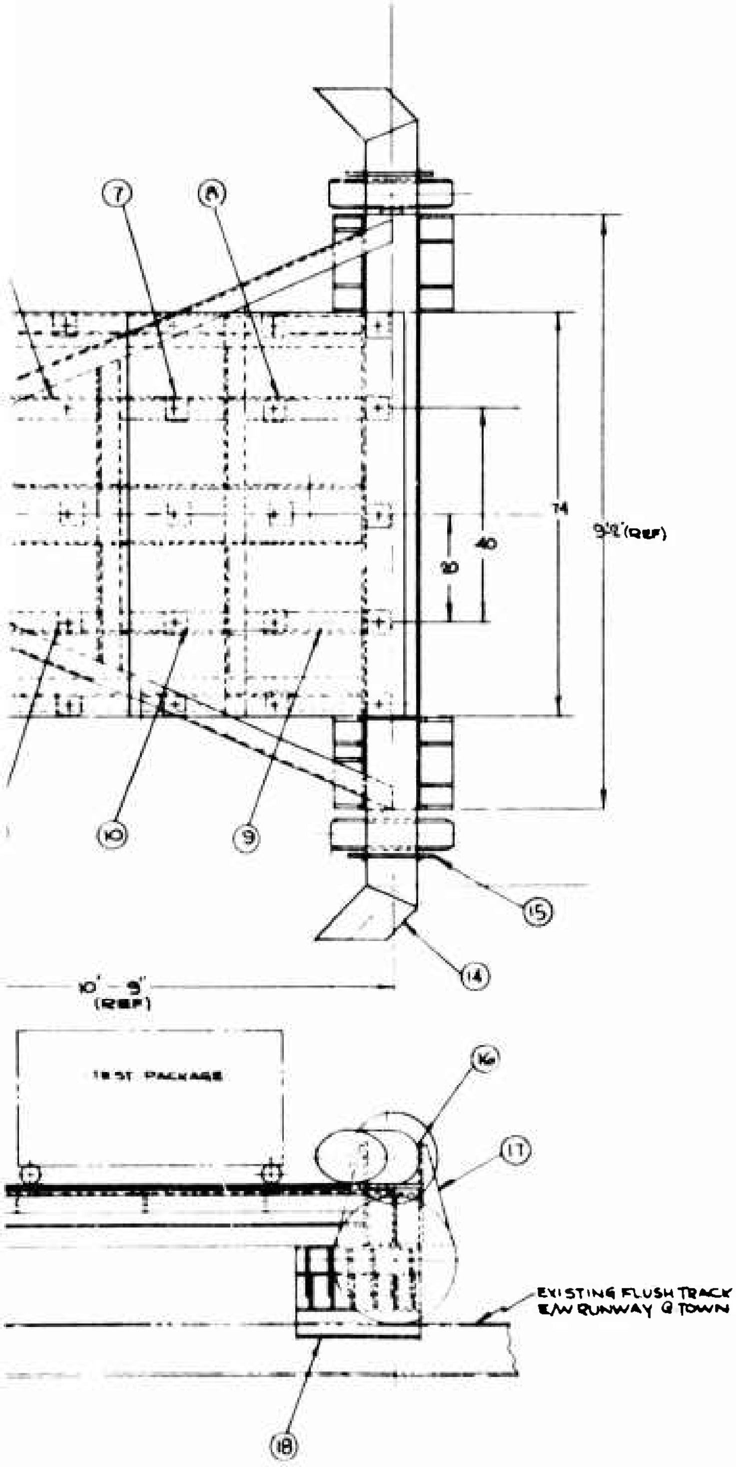


Figure 98. Dead Load Assembly Proposal

A



NOTE-

1 ALL PARTS SHOWN IN THE LIST OF MATERIALS ARE TO BE INSTALLED ON THE EXISTING DEAD LOAD

2 THE FOLLOWING PARTS ARE TO BE REMOVED FROM THE DEADLOAD (SEE DWGS 50576 & 50573)

A 50573-13 THRU-22

B 50876-36 THRU-52

QTY	ITEM	PART NO.	DESCRIPTION	MAT'L & FIN STOCK SIZE	SPEC OR CODE NO.
2	22	16SK173-22	PLATE	STL 24x2x½	
2	21	-21	PLATE	STL 19x2x½	
20	20	-20	GUSSET	STL 8x8x¼	
1	19	-19	BEAM	STL 12x10x53½x 19	
1	18	-18	BEAM	STL 12x10x53½x 24	
16	17	-17	GUSSET	STL 18x6x¼	
4	16	-16	PLATE	STL 17x9x½	
2	15	-15	PLATE	STL 18 DIA x [illegible]	
2	14	-14	TUBE	STL [illegible]	
AR	13	C4F1070-1	ADAPTER ASSY-TIE DOWN	GFE	
26	12	-12	PLATE	STL 8x4x½	
1	11	-11	ANGLE	STL 4x4x½x22	
1	10	-10	ANGLE	STL 4x4x½x23	
1	9	-9	ANGLE	STL 4x4x½x27	
1	8	-8	ANGLE	STL 4x4x½x27	
1	7	-7	ANGLE	STL 4x4x½x23	
1	6	-6	ANGLE	STL 4x4x½x22	
2	5	-5	BEAM	WF BEAM STL. [illegible]	93"
1	4	-4	BEAM	WF BEAM STL. [illegible]	74"
2	3	-3	PLYWOOD	[illegible] PLYWOOD [illegible]	
1	2	50573-1	DEAD LOAD DOLLY ASSY		
X	1	16SK173-1	DEAD LOAD ASSY		

LIST OF MATERIAL

ALL AMERICAN ENGINEERING CO.

DEAD LOAD ASSEMBLY
PROPOSAL

96157 | F | 16SK173

1" = 1'-0" | 301018 | 1

Cost Estimate

An analysis of the items that make up the cost array is as follows:

1. Preparation for Test:

a. Impact Barrier Preparation. Design and build a structural barrier for installation of shock struts capable of sustaining required impact forces.

b. Test Vehicle Preparation. Design and build structural members to improve an existing dolly so that a plywood floor and tie-down fittings can be installed under the expected tie-down loads. The proposed dead load assembly (Figure 98), as shown with the elimination of items 14, 15, 16, and 17, is adaptable to the proposed model momentum test method. Items 1 through 17 are required when utilizing arrestment-type test methods.

c. Instrumentation Preparation. Prepare instrumentation on the test dolly to include:

(1) Two channels of acceleration
(2) Two channels of strain.
(3) Three channels of position (relative motion of cargo and restraining device.)
(4) One camera fixed on runway to observe the cargo during load application.

d. Fabrication of Test Restraints. Build two load limiter tie-down assemblies for two consecutive test runs, and for the remainder of test, rebuild the load limiters by salvaging the parts not destroyed.

e. Test Plan. Write a test plan for a series of dynamic tests to conform to the formulated design criteria and the selected restraint concept.

2. Conduction of Test:

a. Test Operation Mechanics. The test will be conducted with one run below and one above the design g and two runs at the design g. The total number of test runs will be five, which includes one preliminary shot. The crew required consists of two people for the test equipment and one test engineer.

b. Instrumentation and Photographs. Development of the instrumentation channels is to be completed after each test run. The photographs are to include:

(1) Motion picture film to cover fabrication and engineering tests of the project to provide an overall pictorial record of the operations.
(2) Color shots covering the overall project.
(3) Still photographs of the cargo area before and after arrestment.

c. Data Reduction and Chart Curves. After each run, data pertinent to continuation of the test runs will be reduced. Reduction of all data will be presented in the form of tables, graphs, and charts. The basic data will be force-time history, cargo and vehicle accelerations, restraint device tensions, and position of cargo relative to vehicle.

3. Documentation:

a. Draft Test Report. At the completion of the tests, a draft report will include a verbal and graphic description of all tests conducted, including a description of test apparatus involved, test procedures, and results of the test.

b. Revised Test Report. Upon approval of the draft by the Contracting Officer, a final test report will be furnished.

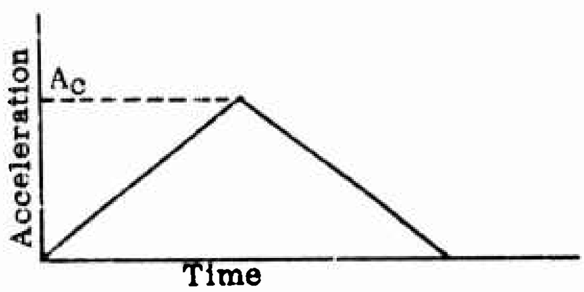
The cost sheets (page 167) depict the itemized estimate of each phase outlined above. The total estimated cost including G and A is $16,450.29.

It is also estimated that nine and one-half weeks is the total required time to complete all phases. This includes two weeks for customer approval of the draft report.

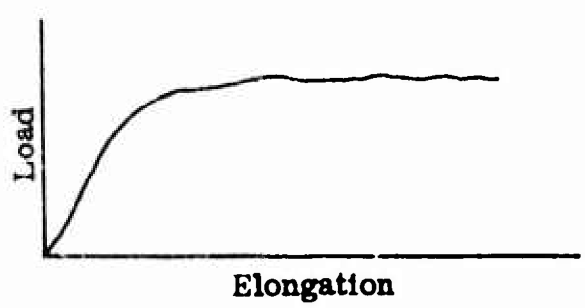
ANALYSIS

The following steps must be realized in formulating a model dynamic test of a cargo restraint system:

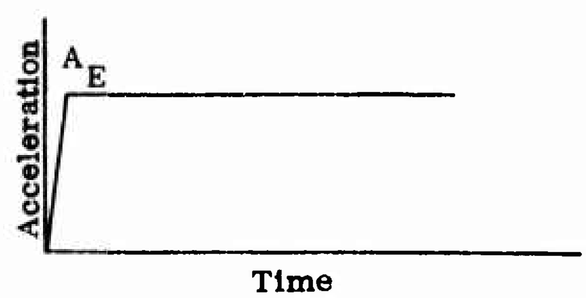
(a) The pulse duration envelope criterion is defined by a triangular pulse curve of 25 g's and 0.25 second.



(b) With item (a) established, the characteristics of the restraint system are then designed when a realistic restraint system is designed. The characteristics of the restraint system are related to the energy performance of the design restraint.



(c) Knowing the restraint characteristics, an equivalent square pulse curve can then be established.



(d) From a dimensionless analysis, the scaled factors of pertinent parameters from prototype to model are established.

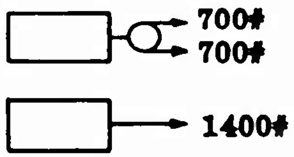
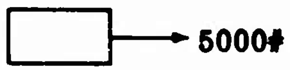
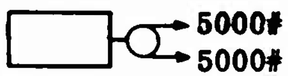
(e) Finally, a model dynamic test environment for the restraint system would then be established utilizing the above information.

The accuracy of the test is dependent on the load developed in the restraints versus the test vehicle inertia force during deceleration. This is due to the fact that the impact force equals the sum of the vehicle's inertia force and the restraint load. The smaller the restraint load is in relation to the vehicle's inertia force, the more accurate the test results. Table XIV shows the accuracy attained in utilizing a model (designed) load limiter tie-down assembly, with intensity of 14,000 pounds, to test with a model cargo weight of 100 pounds. Also included in the table are the prototype load limiter tie-down assemblies of 5,000- and 10,000-pound capacities utilized with model cargo weights of 360 and 715 pounds respectively. All other pertinent model parameters are the same for these three cases and can be found in the table.

From the cost analysis (Figure 99), the fabrication of the model restraint system cost \$2,432 (including General and Administration). By utilizing the fabricated prototype restraints as Government furnished equipment (GFE), this amount may be realized as a savings. However, the accuracy of the optimum test results utilizing GFE is decreased by about 7 and 13 percent, whereby the model tie-down assembly is contained within 2 percent. See Table XIV.

In order to realize the same test accuracy with the prototype restraints, vehicle weights of 17,650 and 35,300 pounds, as depicted in Table XIV, are required. This requirement would result in an increase in the impact force; therefore, a more sophisticated test decelerator is required. The cost savings of \$2,432 is about 14.6 percent of the total estimated cost of testing the restraint system (\$1,645, including General and Administration). Therefore, this savings is offset by the need for a heavier vehicle and cargo mass, a greater potential barrier capable of sustaining a high impact force, and a degradation in the accuracy of the test results. Therefore, it is concluded that little savings, if any, will be realized if the prototype instead of the model restraints is used for the model dynamic testing of the restraint system.

TABLE XIV
ACCURACY OF MODEL VERSUS PROTOTYPE DYNAMIC TESTING

Item	Description	Test Vehicle Weight (pounds)	Inertia Force MA_E (pounds)	Restraint Load P (pounds)	Cargo Weight (pounds)	Test Results Accuracy $\frac{MA_E - P}{MA_E} \times 100$ (percent)	Remarks
Model Tie-Down Assembly		5,000	75,600	1,400	100	98.2	
Prototype Tie-Down Assembly Without Pulley		5,000	75,600	5,000	360	93.4	A test vehicle weight of 17,650 pounds is required to realize the same test results accuracy using model restraints.
Prototype Tie-Down Assembly With Pulley		5,000	75,000	10,000	715	86.7	A test vehicle weight of 35,300 pounds is required to realize the same test results accuracy using model restraints.

ALL AMERICAN ENGINEERING CO.

TITLE CUST/S.O. PREPARED BY DATE

		GRAND TOTAL				
	EVENT NO.	TO	TO	TO	TO	TO
	REQUIRED TIME, WKS.					
	ENGINEERING HRS.	804				
	SHOP HRS. WILM.	200				
	SHOP HRS., G-TWN.	496				
LABOR COSTS	ENGINEERING (DIRECT)	2,465.00				
	SHOP-WILM. (DIRECT)	1,552.00				
	SHOP G-TWN. (DIRECT)	1,388.60				
	SUB-TOTAL	5,405.60				
	ENGINEERING BURDEN	2,412.00				
	SHOP-WILM. BURDEN	700.00				
	SHOP G-TWN. BURDEN	1,984.00				
	SUB-TOTAL	5,096.00				
DIRECT COSTS	PRIME MATERIAL	3,215.00				
	PURCHASED PARTS					
	SUB-CONTRACT					
	SUB-TOTAL	3,215.00				
	TRAVEL	460.00				
	PER-DIEM	128.00				
	AUTO-RENTAL					
	SUB-TOTAL	588.00				
	OTHER					
	TOTAL PRODUCTION COST	14,304.60				
	G & A EXPENSES					
	TOTAL ESTIMATED COST	16,450.29				
	FIX FEE/ PROFIT					
	TOTAL CONTRACT PRICE	18,095.32				

Figure 99. AAE Cost Analysis (sheet 1 of 4)

ALL AMERICAN ENGINEERING CO.

TITLE CUST/S.O. PREPARED BY DATE

	1 TEST PREP	2 CONDUCT TEST	3 DOCUMENTATION		
EVENT NO.	TO	TO	TO	TO	TO
REQUIRED TIME, WKS.					
ENGINEERING HRS.	450	182	172		
SHOP HRS. WILM.	200				
SHOP HRS., G-TWN.	264	232			
LABOR COSTS: ENGINEERING (DIRECT)	960.50	773.50	731.00		
SHOP-WILM. (DIRECT)	1,552.00				
SHOP G-TWN. (DIRECT)	739.00	649.60			
SUB-TOTAL	3,251.50	1,423.10	731.00		
ENGINEERING BURDEN	1,350.00	546.00	516.00		
SHOP-WILM. BURDEN	700.00				
SHOP G-TWN. BURDEN	1,056.00	928.00			
SUB-TOTAL	3,106.00	1,474.00	516.00		
DIRECT COSTS: PRIME MATERIAL	2,415.00	725.00	75.00		
PURCHASED PARTS					
SUB-CONTRACT					
SUB-TOTAL	2,415.00	725.00	75.00		
TRAVEL	100.00	160.00	200.00		
PER-DIEM			128.00		
AUTO-RENTAL					
SUB-TOTAL	100.00	160.00	328.00		
OTHER					
TOTAL PRODUCTION COST	8,872.50	3,782.10	1,650.00		
G & A EXPENSES					
TOTAL ESTIMATED COST	10,203.38	4,349.42	1,897.50		
FIX FEE/PROFIT					
TOTAL CONTRACT PRICE	11,223.72	4,784.36	2,087.25		

Figure 99. AAE Cost Analysis (sheet 2 of 4)

ALL AMERICAN ENGINEERING CO.

TITLE CUST/S.O. PREPARED BY DATE

		1A IMPACT BARRIER PREP	1B TEST VEHICLE PREP	1C INSTRUM. PREP	1D FAB TEST RESTRAINTS	1E TEST PLAN
	EVENT NO	TO	TO	TO	TO	TO
	REQUIRED TIME, WKS.					
	ENGINEERING HRS.	16	56	224	102	52
	SHOP HRS. WILM.				200	
	SHOP HRS., G-TWN.	48	96	120		
LABOR COSTS	ENGINEERING (DIRECT)	68.00	238.00		433.50	221.00
	SHOP-WILM. (DIRECT)			952.00	600.00	
	SHOP G-TWN. (DIRECT)	134.40	268.60	336.00		
	SUB-TOTAL	202.40	506.60	1,288.00	1,033.50	221.00
	ENGINEERING BURDEN	48.00	168.00	672.00	306.00	156.00
	SHOP-WILM. BURDEN				700.00	
	SHOP G-TWN BURDEN	192.00	384.00	480.00		
	SUB-TOTAL	240.00	552.00	1,152.00	1,006.00	156.00
DIRECT COSTS	PRIME MATERIAL	220.00	620.00	1,500.00	75.00	
	PURCHASED PARTS					
	SUB-CONTRACT					
	SUB-TOTAL	220.00	620.00	1,500.00	75.00	
	TRAVEL	40.00	60.00			
	PER-DIEM					
	AUTO-RENTAL					
	SUB-TOTAL	40.00	60.00			
	OTHER					
	TOTAL PRODUCTION COST	702.40	1,738.60	3,940.00	2,114.50	377.00
	G & A EXPENSES	15%				
	TOTAL ESTIMATED COST	807.76	1,999.39	4,531.00	2,431.68	433.55
	FIX FEE/ PROFIT					
	TOTAL CONTRACT PRICE	888.54	2,199.33	4,984.10	2,674.84	476.90

Figure 99. AAE Cost Analysis (sheet 3 of 4)

ALL AMERICAN ENGINEERING CO.

TITLE CUST/S.O. PREPARED BY DATE

	2A TEST OPN. MECHANICS	2B INSTRU. & PHOTO.	2C DATA REDUCTION	3A DRAFT TEST REPORT	3B FINAL TEST REPORT
EVENT NO.	TO	TO	TO	TO	TO
REQUIRED TIME, WKS.					
ENGINEERING HRS.	20	100	62	92	80
SHOP HRS. WILM.					
SHOP HRS., G-TWN.	72	160			
LABOR COSTS: ENGINEERING (DIRECT)	85.00	425.00	263.50	391.00	340.00
SHOP-WILM. (DIRECT)					
SHOP G-TWN. (DIRECT)	201.60	448.00			
SUB-TOTAL	286.60	873.00	263.50	391.00	340.00
ENGINEERING BURDEN	60.00	300.00	186.00	276.00	240.00
SHOP-WILM. BURDEN					
SHOP G-TWN. BURDEN	288.00	640.00			
SUB-TOTAL	348.00	940.00	186.00	276.00	240.00
DIRECT COSTS: PRIME MATERIAL	25.00	700.00		50.00	25.00
PURCHASED PARTS					
SUB-CONTRACT					
SUB-TOTAL	25.00	700.00		50.00	25.00
TRAVEL	60.00	100.00		100.00	100.00
PER-DIEM				64.00	64.00
AUTO-RENTAL					
SUB-TOTAL	60.00	100.00		164.00	164.00
OTHER					
TOTAL PRODUCTION COST	719.60	2,613.00	449.50	881.00	769.00
G & A EXPENSES					
TOTAL ESTIMATED COST	827.54	3,004.95	516.92	1,013.15	884.35
FIX FEE/PROFIT					
TOTAL CONTRACT PRICE	910.29	3,305.44	568.61	1,114.46	972.78

Figure 99. AAE Cost Analysis (sheet 4 of 4)

APPENDIX V

ITEMIZED COST ESTIMATE
OF
SELF-STORING, WIRE-BENDING ENERGY ABSORBERS

ITEM	DESCRIPTION	COST PER ITEM
10066007	Tongue Plate Bottom	1.09*
10066008	Tongue	1.79*
10066009	Tongue Plate Top	1.09*
100660011	Guide	.77
100660012	Platen	9.00
100660013	Wire	.10
100660014	Filler	.71
100660015	Hook Assembly 5^{K}	6.05
100660018	Shield	.60
10066005	Cover	5.60
Assembly		5.00
		31.80
Profit 10%		3.20
		$35.00

The cost of each energy absorber is based on 2,000 production units.

*The cost of these items is based on 6061-T6 aluminum shaped bar extrusions. The prototype units were delivered with 2024-T4 machined parts.

Unclassified

Security Classification

DOCUMENT CONTROL DATA - R&D	
(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)	
1. ORIGINATING ACTIVITY (Corporate author) All American Engineering Company, Lancaster Pike and Center Road, Wilmington, Delaware 19899	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP
3. REPORT TITLE AIRCRAFT CARGO RESTRAINT SYSTEM	
4. DESCRIPTIVE NOTES (Type of report and inclusive dates) Final	
5. AUTHOR(S) (Last name, first name, initial) RUSSO, A.	

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
September 1966	183	26

8a. CONTRACT OR GRANT NO. DA-44-177-AMC-68 (T) b. PROJECT NO. 1F121401A254 c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) USAAVLABS Technical Report 66-50 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) None
10. AVAILABILITY/LIMITATION NOTICES Distribution of this document is unlimited.	
11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY US Army Aviation Materiel Laboratories, Fort Eustis, Virginia

13. ABSTRACT

An investigation of the restraint requirements for Army cargo in Army aircraft, presenting realistic criteria for cargo restraint systems with the formulation of feasible restraint concepts. The existing static load factor criterion is shown to be insufficient. A dynamic analysis relates crash pulse criterion to restraint system deflection, leading to load limiting type energy absorber concepts.

An effectiveness analysis is given of existing Army cargo restraint methods as presently employed, the same methods applied in a correct manner, and the proposed restraint system properly applied.

DD FORM 1 JAN 64 1473

Unclassified

Security Classification

Unclassified

Security Classification

14. KEY WORDS	LINK A ROLE	LINK A WT	LINK B ROLE	LINK B WT	LINK C ROLE	LINK C WT
Aircraft Cargo Restraint						

INSTRUCTIONS

1. ORIGINATING ACTIVITY: Enter the name and address of the contractor, subcontractor, grantee, Department of Defense activity or other organization *(corporate author)* issuing the report.

2*a*. REPORT SECURITY CLASSIFICATION: Enter the overall security classification of the report. Indicate whether "Restricted Data" is included. Marking is to be in accordance with appropriate security regulations.

2*b*. GROUP: Automatic downgrading is specified in DoD Directive 5200.10 and Armed Forces Industrial Manual. Enter the group number. Also, when applicable, show that optional markings have been used for Group 3 and Group 4 as authorized.

3. REPORT TITLE: Enter the complete report title in all capital letters. Titles in all cases should be unclassified. If a meaningful title cannot be selected without classification, show title classification in all capitals in parenthesis immediately following the title.

4. DESCRIPTIVE NOTES: If appropriate, enter the type of report, e.g., interim, progress, summary, annual, or final. Give the inclusive dates when a specific reporting period is covered.

5. AUTHOR(S): Enter the name(s) of author(s) as shown on or in the report. Enter last name, first name, middle initial. If military, show rank and branch of service. The name of the principal author is an absolute minimum requirement.

6. REPORT DATE: Enter the date of the report as day, month, year; or month, year. If more than one date appears on the report, use date of publication.

7*a*. TOTAL NUMBER OF PAGES: The total page count should follow normal pagination procedures, i.e., enter the number of pages containing information.

7*b*. NUMBER OF REFERENCES: Enter the total number of references cited in the report.

8*a*. CONTRACT OR GRANT NUMBER: If appropriate, enter the applicable number of the contract or grant under which the report was written.

8*b*, 8*c*, & 8*d*. PROJECT NUMBER: Enter the appropriate military department identification, such as project number, subproject number, system numbers, task number, etc.

9*a*. ORIGINATOR'S REPORT NUMBER(S): Enter the official report number by which the document will be identified and controlled by the originating activity. This number must be unique to this report.

9*b*. OTHER REPORT NUMBER(S): If the report has been assigned any other report numbers *(either by the originator or by the sponsor)*, also enter this number(s).

10. AVAILABILITY/LIMITATION NOTICES: Enter any limitations on further dissemination of the report, other than those imposed by security classification, using standard statements such as:

(1) "Qualified requesters may obtain copies of this report from DDC."

(2) "Foreign announcement and dissemination of this report by DDC is not authorized."

(3) "U. S. Government agencies may obtain copies of this report directly from DDC. Other qualified DDC users shall request through ____________."

(4) "U. S. military agencies may obtain copies of this report directly from DDC. Other qualified users shall request through ____________."

(5) "All distribution of this report is controlled. Qualified DDC users shall request through ____________."

If the report has been furnished to the Office of Technical Services, Department of Commerce, for sale to the public, indicate this fact and enter the price, if known.

11. SUPPLEMENTARY NOTES: Use for additional explanatory notes.

12. SPONSORING MILITARY ACTIVITY: Enter the name of the departmental project office or laboratory sponsoring *(paying for)* the research and development. Include address.

13. ABSTRACT: Enter an abstract giving a brief and factual summary of the document indicative of the report, even though it may also appear elsewhere in the body of the technical report. If additional space is required, a continuation sheet shall be attached.

It is highly desirable that the abstract of classified reports be unclassified. Each paragraph of the abstract shall end with an indication of the military security classification of the information in the paragraph, represented as *(TS)*, *(S)*, *(C)*, or *(U)*.

There is no limitation on the length of the abstract. However, the suggested length is from 150 to 225 words.

14. KEY WORDS: Key words are technically meaningful terms or short phrases that characterize a report and may be used as index entries for cataloging the report. Key words must be selected so that no security classification is required. Identifiers, such as equipment model designation, trade name, military project code name, geographic location, may be used as key words but will be followed by an indication of technical context. The assignment of links, rules, and weights is optional.

Unclassified

Security Classification

6468-66