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ENERGY ABSORPTION AND DISSIPATION DEVICES

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A research investigation was conducted to determine new and efficient methods to accomplish energy absorption and dissipation for high cyclic rate weapon recoil mechanisms. This study was performed by the Ohio State University under the direction of the Research Directorate, GEN Thomas J. Rodman Laboratory, Rock Island Arsenal. The main requirements of energy absorbing systems are reviewed and the relationships between impact velocity, displacement, deceleration profile and time are discussed in detail. A

20. (continued)

literature search on energy devices was conducted from which the following two mechanisms were selected for further evaluation: a rolling toroid mechanism and a reverse wound flat coil spring. Mathematical analyses and models were prepared for the selected mechanisms.

FOREWORD

This report was prepared by Dr Lynn L. Faulkner of the Ohio State University at Columbus, Ohio under Basic Agreement DAHC04-72-A-0001, Task Order 72-472.

The contract was part of a general spring study authorized and funded by the U. S. Army Small Arms Systems Agency, DA Project 1W562604A607. The work was conducted under the direction of the Research Directorate, GEN Thomas J. Rodman Laboratory, Rock Island Arsenal, Rock Island, Illinois with Henry Swieskowski as project engineer.

CONTENTS

ENERGY ABSORPTION AND DISSIPATION DEVICES

Introduction

The purpose of this investigation was to perform a literature search for energy absorbing and energy dissipating devices. From the literature search, a device was to be selected which could be classified as "new" and "efficient" for a detailed analysis and evaluation.

Summary

A literature search was conducted at The Ohio State University Libraries, at the Ohio State University Mechanized Information Center, and at the Battelle Memorial Institute library for energy absorption and energy dissipation devices. This search yielded a number of devices, some covered by unused NASA Patents and some covered by private patents and some unrestricted.

From the literature search, two devices were selected for analysis: a rolling toroid device and a reverse-wound coil spring. The rolling toroid device is an efficient energy dissipation device and can be designed to meet the specifications for this project. However, it does have some disadvantages which were not specified for the device--it is not self-returning to its original unstroked position when the load is removed; it absorbs little energy compared to the dissipated energy; it requires a minimum force to achieve deflection; the working material must be loaded such that plastic strain exists for energy to be dissipated; and it depends upon friction for the toroids to be actuated. The conclusion is that this device is not satisfactory alone as an energy absorption and dissipation device. It may have some advantages if it were used in connection with some other device such as a ring spring, such that the combination would exhibit desirable properties.

The reverse-wound coil spring is a high energy dissipation spring device which also can be designed to meet the specifications given for this project. Its energy absorption characteristics can be designed in a typical coil spring from spring rate and deflection. The energy dissipation is achieved by the coils being forced together in the radial direction when deflected. It appears that with proper design, the device has some advantages over common spring designs. This device is recommended for experimental evaluation.

TECHNICAL DISCUSSION

Design Requirements for the Energy Absorbing Device

The following design specifications were prescribed for the energy absorbing device:

*Nominal amount of energy absorption: 30 ft. lb

*Amount of energy dissipation: 0-25 ft. lb, variable

*Nominal working stroke: .50 in.

*Temperature stability: Spring properties shall not vary more than 10% within a range of temperatures from -65°F to 125°F.

*Frequency of cycling: Device shall satisfactorily function at a frequency of 300 cycles applied intermittently within a one-minute interval and at a loading rate of 5000 cycles a minute.

*Operating life: Minimum operating life shall be 10,000 cycles at specified frequency.

*Physical size: The mechanism size shall be minimal consistent with sound engineering design. Maximum allowable size is 12" length and 1.5" diameter.

Review of Energy Absorber Designs

The classical approach in the design of an energy absorbing system is to consider the desired relationship between the deceleration force, time, and displacement. For a given initial impact velocity when the mass contacts the absorbing device, different deceleration profiles will stop the mass in different times and over different displacements. Figure 1 illustrates these relations for a fixed stopping distance and a fixed impact velocity, using three different deceleration profiles. From the

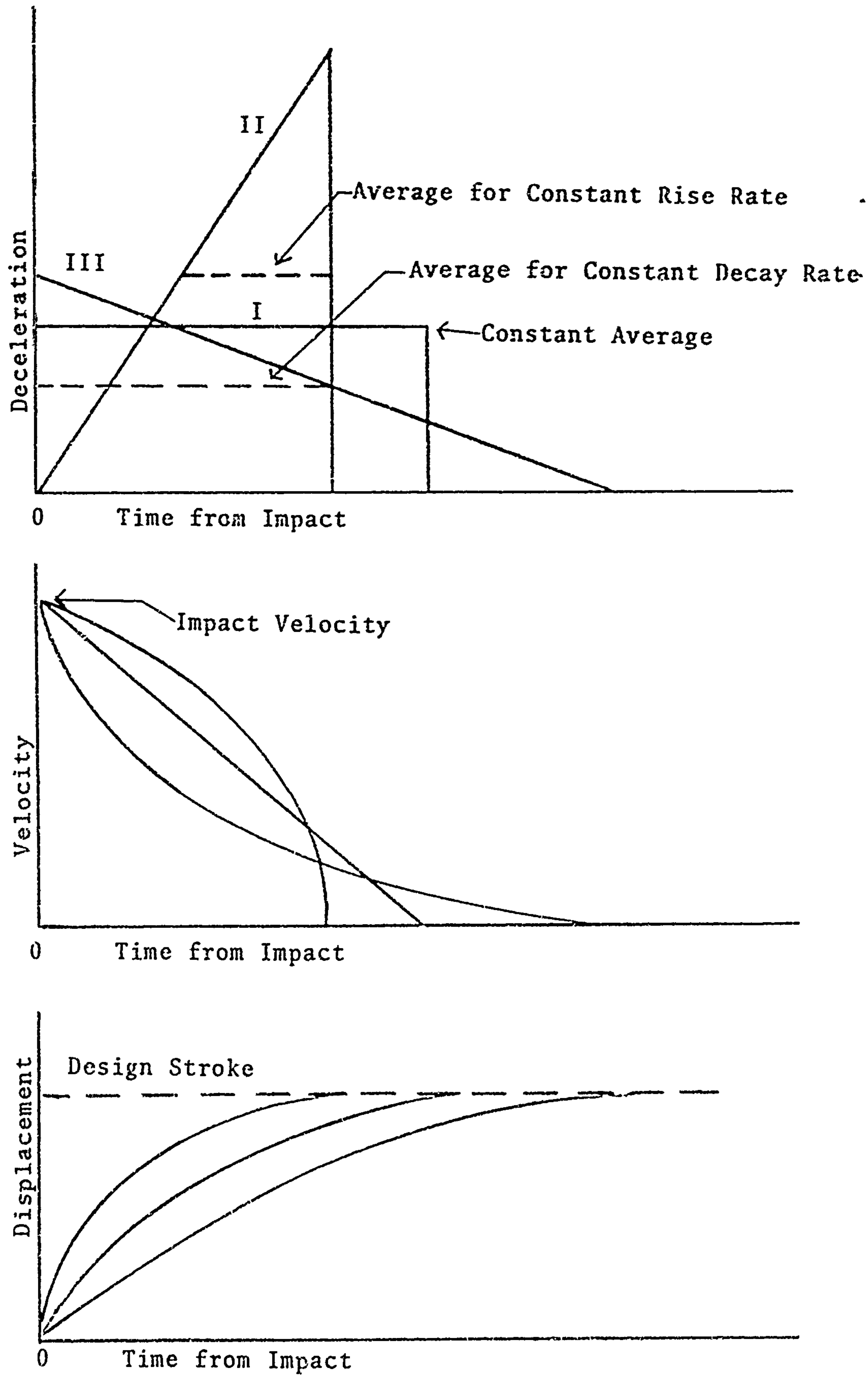


Figure 1. Motion-Time History for Different Deceleration Profiles Using a Fixed Stopping Distance

top set of curves it can be seen that

a) For a specified stopping distance (stroke), a constant-decay-rate deceleration results in the minimum average deceleration.

b) A constant-rise deceleration results in the maximum average deceleration.

c) The constant deceleration profile has the lowest peak deceleration but has an average deceleration higher than the average for the constant-decay-rate profile.

Optimization analyses can be applied to the evaluation of profiles for stroke, total energy dissipation, and a selected severity index for tolerance to deceleration. Conclusions on these subjects given in a Battelle Memorial Institute report (Reference 1) were that profiles other than a constant deceleration versus stroke offer some modest advantages.

Having developed requirements for an ideal energy absorber, it is necessary to consider the effect of the onset rate of acceleration as shown in Figure 2. The effects of onset rate are discussed in Reference 2, with the principal results given by the relation between stopping distance or stroke S, maximum deceleration g_m, impact velocity V, and maximum onset rate $\dot{g}_m$.

$$S = \frac{V^2}{2g_m} (1+\lambda-\lambda^2/12) \qquad (1)$$

where λ gives the effect of the finite onset rate ($\lambda=0$ for an ideal constant deceleration where $\dot{g}_m=\infty$).

$$\lambda = g_m^2/(V\dot{g}_m) \qquad (2)$$

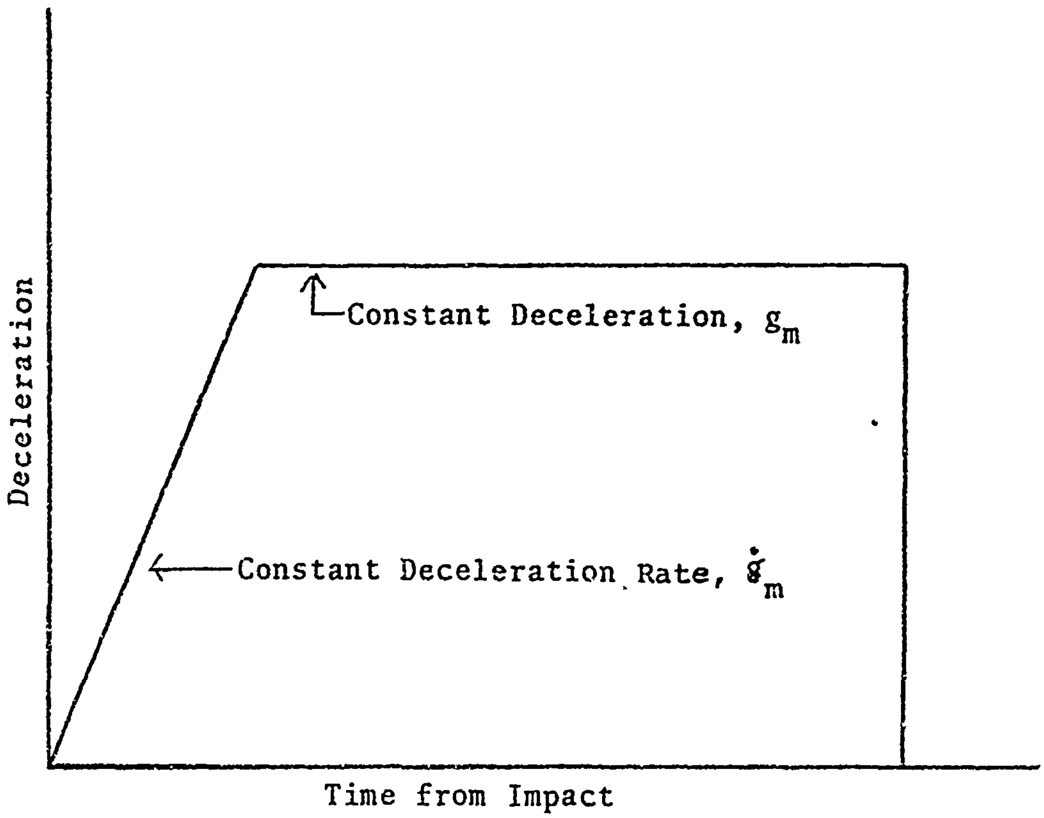


Figure 2. Deceleration Profile with Finite Onset Rate Limit

Calculations show that as long as the deceleration levels are low, the effect of the initial onset rate limit is less than 14 percent and can be neglected for most preliminary design feasibility studies. However, if the maximum allowable deceleration rate is large, λ could be large and the effect of finite onset limit would be to increase the stroke for the same energy dissipation.

Another factor for consideration is that there is an advantage offered by an absorber that is velocity sensitive: low-velocity impacts can be dissipated at lower force levels, using more available stroke. Thus, some degree of velocity dependence may be desirable.

Literature Search for Energy Absorption Devices

A number of patents which deal with energy dissipation devices exist. These patents are listed below in total without assessment of feasibility (from Reference 3):

Patent Categories

Group 1 — Tube and Mandrel

3,143,321 - Frangible Tube Patent (basic) (NASA)

3,181,821 - Spacecraft Safe Landing System (NASA)

3,236,333 - Energy Absorber
Lockheed - Split and roll the tube

3,339,674 - Energy Absorbing Device
General Motors Tube & Mandrel - ductile material - Tube merely becomes a bigger tube.

3,381,778 - Energy Absorbing Device
NASA property - very much like Lockheed 3,236,333. Tube-mandrel split and roll the tube.

<u>Group 2</u> <u>Bending Bars</u>

2,835,348 - Retarder Device
Longitudinal displacement taken up in bending of two bars beyond elastic limit.

3,283,857 - Beam Energy Absorbing Device
Cyclic bending with rectilinear motion. ABA Rube Goldberg mechanism

3,361,475 - Safety Belt with Shock Absorbing Device
Corrugated strip of ductile metal straightens out under impact load.

<u>Group 3</u> <u>Shear Pins</u>

2,837,176 - Safety Device for Automobiles
Bumper mount. Contains pins which shear one after the other.

2,845,144 - Shear Pin Brake for Auto Bumper
Complex frame arrangement with shear pins to promote glancing blow. Also linkage to close valve in front brake lines.

<u>Group 4</u> <u>Folding Tube</u>

2,870,871 - Shock Absorber
Folding Tube

3,240,676 - Nuclear Reactor Including Energy Dissipation Device for Falling Bodies

Straight tube preformed at one end, folding like an accordian.

<u>Group 5</u> <u>Rolling Mill</u>

2,953,189 - Shock Absorber
Rolling mill - bar between rollers

<u>Group 6</u> <u>Machining Metal</u>

2,961,204 - Deceleration Device
Metal machining with cutters as in key seater

<u>Group 7</u> <u>Hydraulic Cylinder</u>

2,959,251 - Auxiliary Bumper Type Impact Absorber
Hydraulic cylinder

3,330-549 - Shock Absorber (NASA Patent)
Hydraulic shock absorber to function over wide range of impact loadings in space vehicle docking

<u>Group 8</u> <u>Belleville Spring Assembly</u>

3,127,157 - Multiple Belleville Spring Assembly (NASA)

3,313,367 - Belleville Spring Biased Bumper
16 claims - dissipates some energy in tube friction.
Locks to avoid rebound.

<u>Group 9</u> <u>Friction Between Sheets</u>

3,164,222 - Nonreusable Kinetic Energy Absorber
Wound sheets - friction mode of absorption (NASA)

<u>Group 10</u> <u>Extrusion</u>

3,209,854 - Single Sheet Energy Dissipater
Extrudable material confined behind a piston. The piston has the extrusion hole in it.

3,380,557 - Variable Kinetic Energy Absorber
Various plastic materials forced by a piston to extrude through an orifice.

2,997,325 - Piston and Cylinder
Absorbs energy by extruding plastic material through apertures at end of cylinder.

<u>Group 11</u> <u>Pulling Strip Between Pins</u> - Wires through Holes

3,211,260 - Energy Absorption Device
Pulls metal strip between pins (Same inventor as 3,366,353).

3,280,942 - Energy Absorber
To anchor airplane seats.
Wires absorb energy as they weave in and out through a series of holes in a metal strip.

3,308,908 - Energy Absorber
Two tubes with helical convolutions. Soft tube screwed into hard tube. Absorbs energy under tensile load by deformation of softer tube.

3,337,004 - Impact Energy Absorber (NASA)
Very low energy absorption. Soft aluminum tape.
Plastically deformed when it is wound onto a spool.
Is plastically deformed again and absorbs energy as it is unwound from spool.

3,366,353 - Energy Absorbing Device (Variation of 3,211,260)
Pulls metal strip between staggered pins.

3,372,773 - Load Limiter Device
"A plurality of wire strands are looped in and out of longitudinal apertures in platten." Pulling wires alternately bends them.

3,377,044 - Cargo Tie-Down Apparatus
Another application of pulling a metal strip between staggered pins as 3,211,260 and 3,366,353.

Group 12 Annealed, Stranded Cable

3,217,838 - Energy Absorbing Device
Annealing steel cable after stranding & means of anchoring cable.

3,353,768 - Energy Absorbing System
Extensible Cable System with ability to handle light impacts gently and heavier ones more firmly.

Group 13 Crushable Material Instead of Hydraulic Fluid in a Similar Configuration

3,228,492 - Double Acting Shock Absorber (NASA)
Crushable material instead of fluid in a double-acting cylinder. (Same inventor as 3,175,789).

3,175,789 - Landing Pad Assembly for Aerospace Vehicles (A Structure). A system to take up misalignments and absorbers which are tube-mandrel combinations acting in tension. (NASA)

3,252,548 - Shock Absorber Cartridge
Cylindrical honeycomb to protect an airplane that gets tipped up on its tail during landing or take off.

3,339,673 - Volumetrically Expandable Energy Absorbing Material
Hexcel Corp.

2,966,200 - Shock Absorbent Fitting
Squash a lead or brass washer (Aircraft seats)

3,160,950 - Method of Apparatus for Shock Protection (NASA)
Shock protection of instruments by embedding in a material which can be sublimated.

Group 14 Torus Action

3,231,049 - Energy Absorbing Device
Aerospace Research Associates (See 3,360,080 and 3,360,081) Torus being twisted inside out - Deformation in torsion.

3,360,080 - Energy Absorbing Device
ARA One of the devices shown in 3,231,049.

3,360,061 - Energy Absorbing Device
ARA One of the devices shown in 3,231,049.

3,369,634 - Energy Absorbing Device
ARA Wire coiled into a helix between two tubes.

Group 15 Shearing Sheet Metal

3,232,383 - Energy Absorbing Means (Developed in Sweden)
Shearing of sheet metal.

3,289,792 - Apparatus for Absorption of Energy from a Moving Load
Virtually identical to 3,232,383 (from Sweden) Looks like a maneuver for legal reasons.

Group 16 Overload Relief in Drive Shaft

3,236,066 - Energy Absorption Device (NASA)
Really just an overload relief in a drive shaft.

Group 17 Rolling Tubes

3,301,351 - Energy Absorbing Device
Rolling tubes between plates with friction

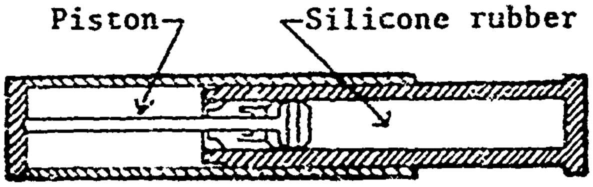
Sketches of devices considered are given in Figure 3. Many other commercial "dampers" are available which were not considered in this study.

Preliminary Evaluation

The requirement of repeatable use eliminates the so-called "mechanical fuse" devices which are a one-time energy absorbing device. The requirement of reliability and repeatability, and the ability to withstand exposure to weather eliminate all devices which rely on friction alone. This is because the friction co-efficient can vary over a wide range under varying operational conditions.

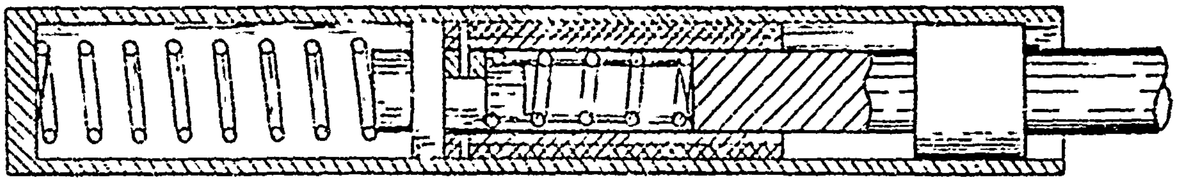
At the end of the preliminary screening process, the following devices remained under consideration:

Bending Bars
Belleville Spring Assembly
Torus Action



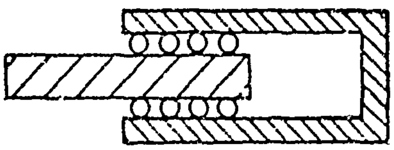
Menasco Shock Absorber

(References 5 + 6)



Dry-Friction Shock Absorber

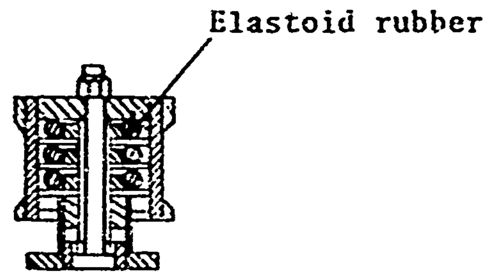
(Reference 7)



Rolling Toroids

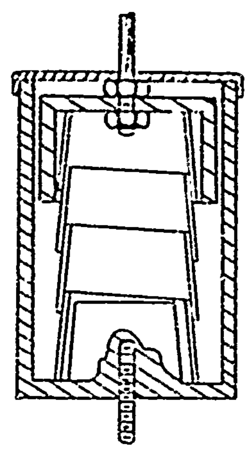
(Reference 8)

Figure 3. Shock Absorbing Devices



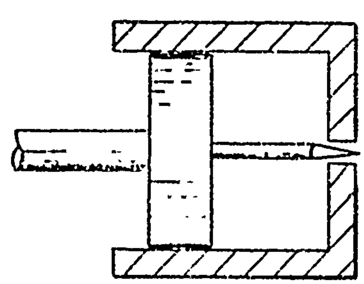
Rubber Snubber

(Reference 9)



Reverse-Wound Coil Spring

(Reference 6)



Metering-Pin Damper

(Reference 10)

Figure 3. (continued)

Menasco Shock Absorber
Rubber Snubber
Reverse Wound Flat Spring
Metering Pin Damper

Of these devices, only torus action and the reverse wound flat spring pass the criteria of "new" and "unique." Bending of bars and for that matter shock rods have long been known as shock absorber devices. Belleville springs are widely used for shock system isolators. The Menasco shock absorber is just a variation of a damper where the working fluid is silicone material. The rubber snubber uses the compression of rubber toroids and the metering pin damper has been known for some time.

Additional Evaluation

The energy absorption and the energy dissipation were two separate design parameters for the device. For this reason, consideration of the energy absorption (without dissipation) characteristics of various systems provide an additional criterion. Presented in Table 1 are comparisons of energy capacity per unit weight of some common systems and in Table 2 are comparisons of energy storage per unit volume. Note that typical energy absorption devices such as springs are at the bottom of both tables. Steel under uniform compression is better for these comparisons than typical springs.

Table 1. Optimum Weight-Modulus Comparisons[a]

(from Reference 4)

System	Ft lb/lb
Gasoline (fuel alone)	14500000
Solid propellant	1050000
Gasoline engine	660000
Ag-Zn electrochemical cell	146000
Lead-acid electrochemical cell	38000
Edison electrochemical cell	35000
Uniform-stress disk with rim	26700
Compressed gas (spherical container)	22600
Compressed gas (cylindrical container)	18400
Cylindrical flywheel (t/r = 0.2)	18000
Rim-arm flywheel	2450
Compressed liquid (ether)	300
Compressed solid (steel)	49
Compressed solid (torsion spring) ($d_i/d_o=1$)	29
Compressed solid (spiral-wound spring)	16
Compressed solid (coil spring)	15
Compressed solid (Belleville spring)	11

[a]Common values of constants which were used in the evaluations of Tables 1 and 2 are $\rho = 7.31 \times 10^{-4}$ lb sec^2/in^4, $G=11.5 \times 10^6$ psi, $\mu=0.288$, $E=30.0 \times 10^6$ psi, $S_s=50{,}000$ psi, $Sm=100{,}000$ psi, and $\rho g=0.283$ pci.

Table 2. Optimum Volume-Modulus Comparisons[a]

(from Reference 4)

System	$\frac{\text{Ft lb}}{\text{ft}^3} \times 10^{-3}$
Gasoline (fuel alone)	682000
Solid propellant	105000
Gasoline engine	99000
Ag-Zn electrochemical cell	14800
Uniform-stress disk with rim	8780
Cylindrical flywheel	8750
Lead-acid electrochemical cell	5000
Edison electrochemical cell	4800
Compressed gas (spherical container)	2500
Compressed gas (cylindrical container)	2500
Rim-arm flywheel	1070
Electrostatic capacitor	200
Compressed liquid (ether)	90
Compressed solid (steel)	24
Compressed solid (torsion spring)(d_i/d_o=1)	8
Compressed solid (coil spring) (D/d = 2)	5
Compressed solid (spiral-wound spring)(r_a/r_c=0.1)	4
Compressed solid (Belleville spring)	2

Rolling Torus

A device which passes the criteria of "new" and "efficient" is the rolling torus system, which after the literature search appeared to be worthy of additional consideration. The mathematical model for this device is given in Appendix I. The energy dissipation of the device as described in the appendix is

$$T = 4/9\ R_o \ell S \sigma_y$$

where

R_o = coil radius

ℓ = length of device

S = stroke

σ_y = yield stress of material.

The device can be designed for a one-stroke operation in which the material fails plastically because the plastic strain is exceeded in the one-stroke operation. There are some materials such as Austenitic steel which can be subjected to considerable plastic strain in every cycle and still be subjected to as many as 400,000 cycles of stress before failure (Reference 12).

For the following values:

r_s = 0.7 in. (radius of toroid)

ℓ = 12 in. (1200 toroids)

S = 0.50 in.

σ_y = 28,000 psi

the resulting energy absorption is 25 ft.lb. for a design life of 20,000 cycles using Austenitic steel with a maximum strain of 0.005 in/in. The radius of the toroid wire would be 0.004 in.

The force required for actuation of the device is given by:

$$F = TS \qquad = 12.5 \text{ lbs.}$$

For forces less than this, the device will not stroke.

Although the device is efficient in dissipation of energy, it does not self-return to its original undeflected position. The rolling toroid concept may be useful in addition to other devices such as ring springs.

Reverse-Wound Coil Springs

The reverse-wound coil spring also appears to have properties which may be desirable and also can be classified as "new" and "efficient." It has the advantage of absorbing energy and at the same time dissipating energy. The ratio of dissipated to absorbed energy appears to be controllable by design.

The mathematical analysis for this device is given in Appendix II with the following form for energy dissipation:

$$T = \frac{2\pi h^2 E\mu F^2}{NR}$$

where

T = energy dissipated

h = thickness of section (smaller dimension of coil)

E = modulus of elasticity

μ = coefficient of friction

F = deflection of spring

N = number of coils

R = mean radius of the spring

For the following parameters,

length = 10 inches

h = 1/32 inch

μ = 0.10

F = 1/2 inch

N = 20 coils (1/32 in x 1/2 in cross sect.)

R = 3/4 inch

the energy dissipated is 25 ft.lb. and nominal absorption is 40 ft.lb.

This device appears to have some advantages for energy absorption and energy dissipation. The energy absorption can be determined as for a typical coil spring from the spring rate and the deflection. Energy is stored in the coils of the device both by shear stress and hoop stress action as compared to shear stress only for coil springs and hoop stress only for ring springs. It is recommended that an experimental investigation of this device be utilized to completely evaluate the parameters involved for the specific requirements desired.

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APPENDIX I

Math Model for Toroidal Action Energy Absorber

The following derivation is based on the following assumptions:

i. Infinitely rigid cylinder and shaft

ii. Elastic perfectly plastic material

iii. No slipping between walls and torus

iv. Only nonzero deformations are circumferential.

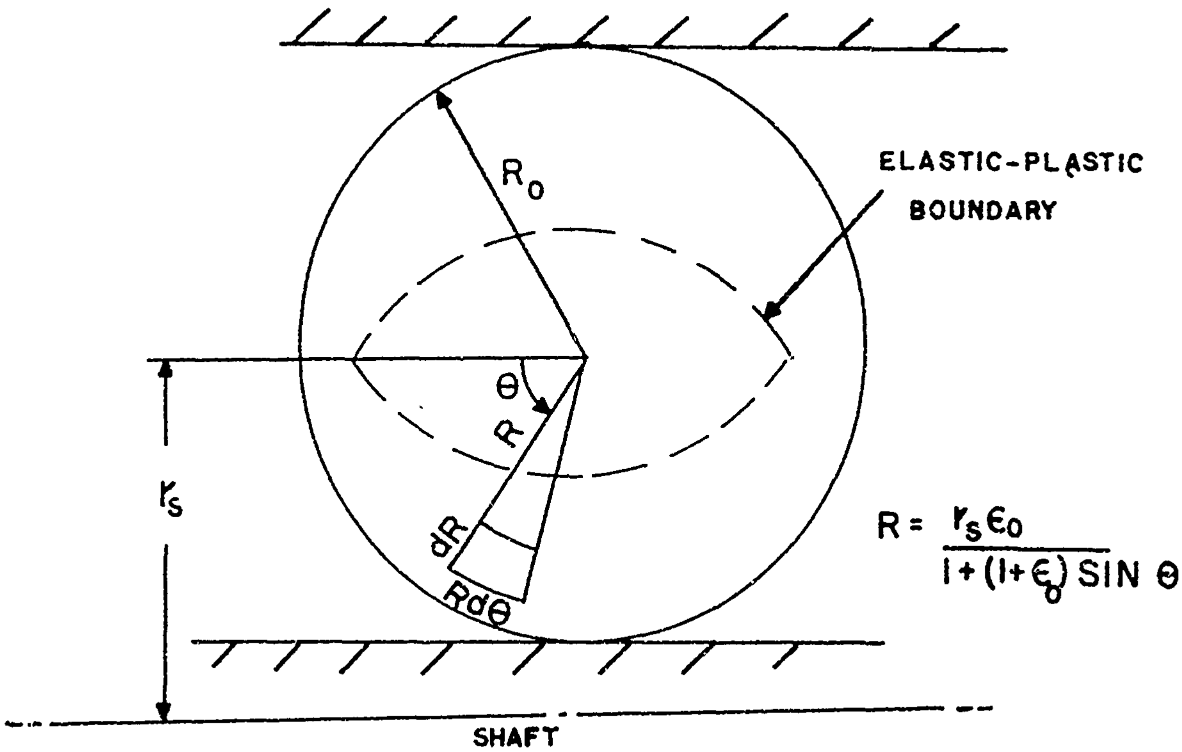

Figure I.1.1. Deformation Analysis of Rolling Toroid

As Figure I.1.1 indicates, the total strain undergone by a differential chord during one turn of the toroid is given by a ratio of the circumferences of the stretched and relaxed chords.

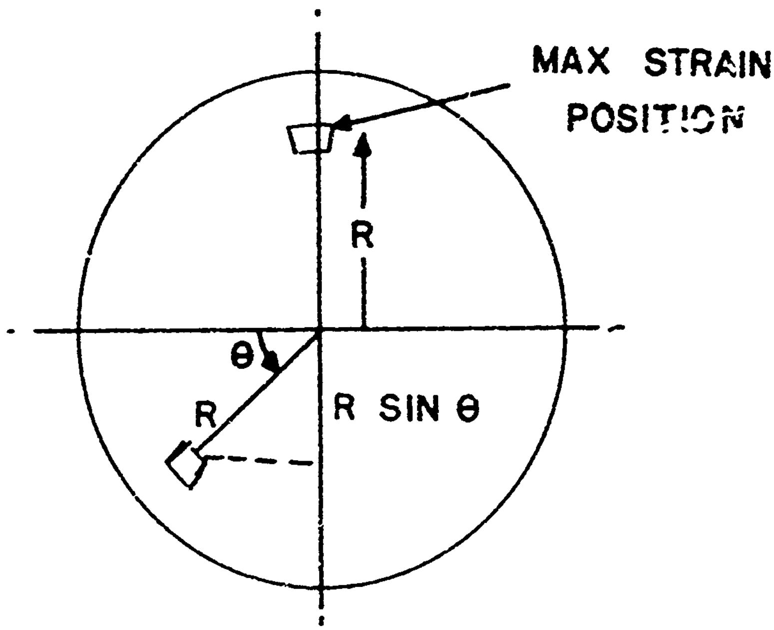

Figure I.1.2. Strain Analysis

$$\text{Max strain} = \frac{2\pi(r_s+R) - 2\pi(r_s-R\sin\theta)}{2\pi(r_s-R\sin\theta)} = \frac{R + R\sin\theta}{r_s-R\sin\theta} \qquad \text{(I.1.1)}$$

If the maximum strain (which is a function of R and θ) is less than ε_0, where ε_0=strain at yield point of the toroid material, then deformation remains within the elastic range and no energy is absorbed. Therefore:

$$R = \frac{r_s\ \varepsilon_0}{[1 + (1 + \varepsilon_0)\ \sin\theta]} \qquad \text{(I.1.2)}$$

is a boundary condition for minimum plastic strain. Equation (I.1.2) is the polar equation for a hyperbola, and all material within the hyperbola is useless for circumferential energy absorption.

To make maximum use of the plastic portion of the stress strain law, the plastic region should include all points of the circumference. Thus,

$$R_o > r_s \varepsilon_o \qquad (I.1.3)$$

Furthermore, the torus must be such that the strain never exceeds the strain at rupture of the material, designated by ε_{ult}. Thus,

$$\varepsilon_{ult} > \frac{2R_o}{r_s - R_o}$$

and

$$R_o < \frac{r_s \varepsilon_{ult}}{(2+\varepsilon_{ult})} \qquad (I.1.4)$$

The ratio of R_o over r_s consequently should fall between these two extremes.

$$\varepsilon_o < \frac{R_o}{r_s} < \frac{\varepsilon_{ult}}{(2+\varepsilon_{ult})} \qquad (I.1.5)$$

This then is a design criterion for the torus. For maximum efficiency this leads to a relation between R_o and r_s:

$$R_o = \frac{r_s \varepsilon_{ult}}{(2+\varepsilon_{ult})} \qquad (I.1.6)$$

We wish to calculate the amount of energy absorbed by a differential volume element of unit length. Consider the loading path of a typical element of material shown in Figure I.1.3. During one cycle of the torus the energy absorbed is given by the area enclosed within the parallelogram.

$$\beta = \frac{R[1 - \sin\theta]}{r_s - R\sin\theta}$$

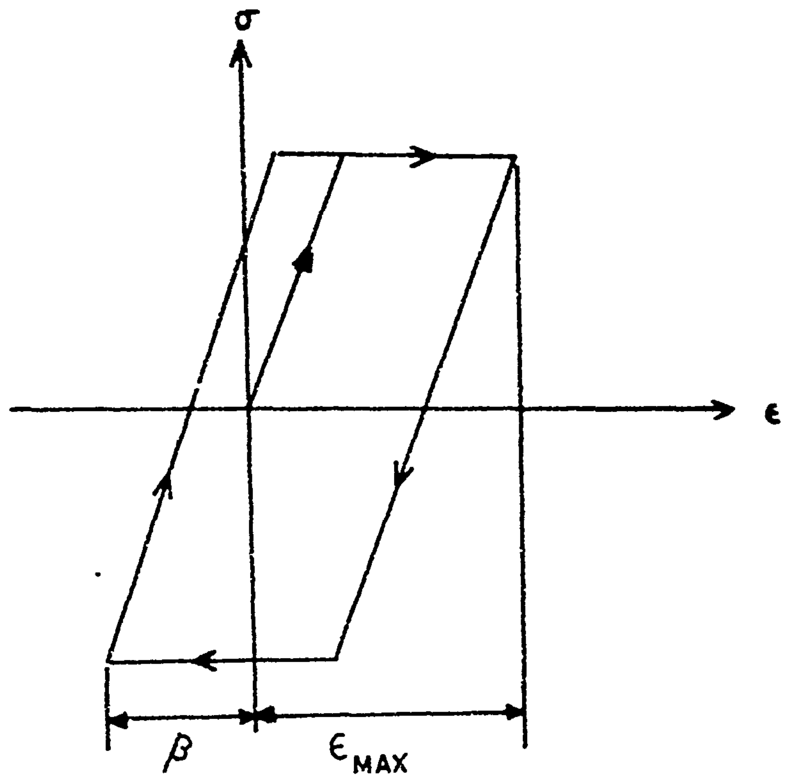

Figure I.1.3. Loading Path of Material

Therefore

$$W = 2(\sigma_y) \quad [\varepsilon_{max} + \beta - 2\varepsilon_o]$$

$$W = 2(\sigma_y) \left[\frac{2R}{r_s - R\sin\theta} - 2\varepsilon_o \right] \qquad \text{(I.1.7)}$$

This has to be integrated over the volume of the torus. The integral for total energy absorbed in one revolution is given by

$$W_T = 4(\sigma_y) \int_0^{\pi} \int_{\frac{\varepsilon_o r_s}{[1+(1+\varepsilon_o)\sin\theta]}}^{R_o} W\,(r - R\sin\theta)\, 2\pi R dR d\theta \qquad \text{(I.1.8)}$$

The integration can be simplified by approximating the lower limit of R by the average value of R to the elastic-plastic value. This then is given by

$$a_o = \frac{3}{4} \varepsilon_o r_s \tag{I.1.9}$$

Integration of equation (7) results in

$$W_T = (\sigma_y) \left\{ 8\pi\left(\frac{2\pi}{3} + \varepsilon_o\right) \left[R_o^3 - \frac{27}{64} \varepsilon_o^3 r_s^3\right] - \frac{64}{3} \varepsilon_o r_s \pi^2 \left(R_o^2 - \frac{9}{16} \varepsilon_o^2 r_s^2\right)\right\} \tag{I.1.10}$$

As the material approaches a rigid plastic material this equation becomes greatly simplified. As

$$\varepsilon_o \to 0$$

$$W_T = \frac{16}{3} \pi^2 R_o^3 (\sigma_y) \tag{I.1.11}$$

Note that for this type of behavior the dependence on r_s drops out.

Consider now how the toroids function in the device. If the overall length of the device is ℓ, only a fraction of the length will be filled with toroids. A reasonable figure would be $\ell/3$. Each torus would go through $(\ell/6\pi R_o)$ cycles. If the toroids are spaced evenly at one diameter intervals, then the number of toroids is given by $\ell/2R_o$. The total energy available would then be

$$T = W_T \frac{\ell^2}{12\pi R_o^2} \tag{I.i.12}$$

For a rigid-plastic material this gives

$$T = \frac{4}{9} \pi R_o \ell^2 (\sigma_y) \tag{I.1.13}$$

For a shorter stroke, S, each torus will go through $(S/6\pi R_o)$ cycles. The total energy would then be

$$T = \frac{4}{9} R_o \ell S \sigma_y \tag{I.1.14}$$

The maximum energy dissipation results when ε_{ult} is used in equation I.1.5. Values of ε between ε_o and ε_{ult} will result in differing amounts of energy dissipated.

Example:

If $r_s = 0.7$,

$\varepsilon_{ult} = 0.20$,

$S = 0.50$,

$\sigma_y = 20{,}000$ psi,

$\ell = 12.0$

then $T = 3{,}700$ in.lb.

This result is due to failure of the material. Lower stresses in the material will absorb less energy.

APPENDIX II

Math Model for Reverse-Wound Flat Coil Spring

Reverse-wound coil springs absorb energy both in the torsional action of the rectangular sections and the hoop stresses which exist when the coils are forced together. Dissipation of energy occurs due to hysteritic action and due to friction between the coils.

The formula for deflection is from Reference 11:

$$F = \frac{PND^3}{K_2 Gbh^3} \qquad \text{(II.1.1)}$$

where

D = mean diameter of the spring (inches)

F = deflection (inches)

N = number of coils

G = torsional modulus (psi)

P = load (pounds)

b = width of section (larger dimension inches)

h = thickness of section (smaller dimension inches)

K_2 = constant that varies with b/h.

b/h	2.0	3.0	4.0	6.0	8.0	10.0
K_2	.292	.335	.358	.381	.391	.399

For the expansion of one coil over the other during compression of the spring, the coils also develop hoop stresses:

$$\text{hoop stress, } \sigma = \frac{pR}{h} \qquad \text{(II.1.2)}$$

$$\text{radial displacement} = \frac{R\sigma}{E} \qquad \text{(II.1.3)}$$

where p = pressure on coil surfaces (pounds/inch)

R = radius of spring coils (inches)

h = thickness of coils (inches)

σ = stress (psi)

E = modulus of elasticity (psi)

If the coils are wound such that one half of the coil width overlaps the previous coil, the average radial displacement becomes:

$$\text{radial deflection} = \frac{2hF}{Nb} \qquad (II.1.4)$$

with F, N, and b given under equation II.1.3. Therefore upon substitution into equation II.1.3,

$$\frac{2hF}{Nb} = \frac{R\sigma}{E} \qquad (II.1.5)$$

and

$$\sigma = \frac{2hEF}{NbR} \qquad (II.1.6)$$

From II.1.2,

$$p = \frac{2h^2EF}{NbR^2} \qquad (II.1.7)$$

The resulting radial force on the coils is

$$\text{radial force} = \frac{4h^2\pi EF}{NR} \qquad (II.1.8)$$

From Reference 6, the limiting value of frictions between the coils is such that the coils remain compressed when the spring is deflected; therefore the limiting axial force will be

$$P = \frac{4\pi h^2EF}{NR}\,\mu \qquad (II.1.9)$$

where μ is the coefficient of friction between the coils. The work done in deflecting the coils is PdF, and the total energy dissipated is

$$T = PdF \tag{II.1.10}$$

or

$$T = \frac{2\pi h^2 E\mu F^2}{NR} \tag{II.1.11}$$

For coils overlapped b/2, the total deflection, F = b/2 (no. of coils - 1).

Example: $h = 1/32$,

$b = 1/2$,

$E = 30x10^6$,

$R = 0.75$,

$N = 10$,

$\mu = 0.10$,

total deflection of the spring = 1/4 total length.

T = 80 ft.lb.

APPENDIX III

New Technology Report Shock Absorber

CARD (CC 1-3) 2 3

NATIONAL AERONAUTICS AND SPACE ADMINIS ATION

NEW TECHNOLOGY TRANSMITTAL
(See Instructions on Reverse)

Approved Through January 1970
Budget Bureau No. 104-R0016

NT CONTROL NUMBER (CC 4-11): N P O - 1 1 1 4 6

1. TITLE
SHOCK ABSORBER

2. INNOVATOR(S) (Name and Social Security No.)
Wallis M. Tener (230-14-9032), 370 E. Pentagon, Altadena, Calif. ·.102

3. EMPLOYER (Organization and division)
Caltech/JPL

4. ADDRESS (Place of performance)
4800 Oak Grove Drive
Pasadena, California - 911ↄ

5. DOCUMENTATION (Full and complete disclosure must be enclosed, the contents of which are discussed in NHB 2170.3, Documentation Guidelines for New Technology Reporting. Place an "X" to the left of those items of documentation which are available but NOT enclosed with this transmittal)

	ENGINEERING SPECIFICATIONS		OPERATING MANUALS		COMPUTER TAPES/CARDS
	ASSEMBLY/MFG DRAWINGS		TEST DATA		
	PARTS OR INGREDIENTS LIST		ASSEMBLY/MFG PROCEDURES		

6. PREVIOUS PUBLICATION OR PUBLIC DISCLOSURE

PUBLICATION
TYPE: ☐ JOURNAL ☐ REPORT ☐ CONFERENCE OR SEMINAR
BY: ☐ NASA ☐ OTHER GOVT. ☐ CONTRACTOR ☐

VOLUME NO.	PAGE	DATE	STATUTORY BAR ESTABLISHED (Date)

TITLE

PATENT
STATUS: ☐ APPLICATION FILED ☐ ISSUED
NO.
DATE

7. STATE OF DEVELOPMENT
☐ CONCEPT ONLY ☐ DESIGN ☐ PROTOTYPE ☐ MODIFICATION ☐ PRODUCTION MODEL ☐ USED IN CURRENT WORK

8. ORIGIN (CC 12): P

9. NASA PRIME CONTRACT NO. (CC 13-23): N A S 7 - 1 0 0

10. SUBCONTRACT TIER (CC 24):

11. CONTRACTOR REPORTABLE ITEM NO. (CC 25-33): 3 0 - 1 1 1 4 6

For Internal Use Only

12. CONTRACTOR/GRANTEE NEW TECHNOLOGY NOT SUBMITTED PURSUANT TO NT/PRI CLAUSE PROVISION (CC 34) ☐

13. SUBCONTRACTOR CIC (CC 35-41):

14. NT RECEIPT DATE (CC 42-47): MO 0 8 DAY 1 4 YR · 8

15. PROJ. NO. (CC 48-51): 9 4 0

16. EVALUATION ORGANIZATION (CC 52-54): N P O

17. NT FORWARDED FOR EVALUATION (Date) (CC 55-60): MO DAY YR

18. COMMENTS
Technical Support packags not needed.

19. PREPARED BY
NAME AND TITLE: Robert A. Miller, Sr. T. U. Representative
SIGNATURE: Robert A. Miller
DATE: 5/12/6ᶜ

20. APPROVED (Center TUO)
NAME: John C. Drane
SIGNATURE:
DATE:

NASA FORM 666 SEP 68 PREVIOUS EDITIONS ARE OBSOLETE.

CARD		
2	6	0
1	2	3

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NT ISSUABILITY AND DISPOSITION TRANSCRIPT

Item	Entry	Card columns
1. NT CONTROL NUMBER	NPO 11146	4–11
2. DISPOSITION	T	12
3. DISPOSITION DATE (MO DAY YR)	05 12 69	13–18
4. PRIMARY TECHNICAL CATEGORY	5. 1 A	19–21
5. SECONDARY TECHNICAL CATEGORY		22–24
6. EVALUATION ORGANIZATION	NPO	25–27
7. ISSUABILITY CLASSIFICATION	1	28
8. CLASSIFICATION DATE (MO DAY YR)	05 12 69	29–34
9. INTERIM EVALUATION INSTITUTE		35–37
10. INTERIM EVALUATION START DATE (MO DAY YR)		38–43

11. REASON FOR ISSUABILITY CLASSIFICATION

This item should be published as a Tech Brief due to its potential military and industrial usage. This device could be the basis of a recoil-limiter in firearms and artillery.

There are many industrial uses for a lightweight, inexpensive, re-usable shock absorber -- the protection of goods in shipment, a coupling shock absorber in trains, vehicle bumper shock absorbers, elevator stops, etc.

NASA FORM 1370 (Proposed revision) JAN 69

SHOCK ABSORBER

1. Novelty

An inexpensive, reusable shock absorber and impact limiter employs a flat coiled spring with reversed, overlapping coils which provide increasing frictional resistance under compression.

2. The Disclosure

The Problem

A requirement existed for an inexpensive, repeatable shock absorbing device to protect rocket engines during shipment. Existing shock absorbers are usually expensive hydraulic devices, or they employ various crushable materials to absorb impact forces. The collapsible or crushable devices can be used only once, and are highly susceptible to damage by forces other than shock.

The Solution

This shock absorber employs a flat coiled spring with reversed, overlapping coils which provide increasing frictional resistance as the spring is compressed. The large end of the reversed spring (outer coil) is loosely fitted within a cylindrical cap attached to an actuating rod. The small end of the spring (inner coil) fits loosely around a centered boss located in the bottom of a cylindrical housing. When the actuating rod is depressed, compressing the reversed spring, the coils are forced over each other in a downward skewing motion. As the spring is compressed, frictional resistance between coils increases progressively. The large coil (outer coil) is subsequently forced into contact with the wall of the cylindrical cap, and the small coil (inner coil) contacts the centered boss, stopping spring travel. When the spring is fully compressed, the amount of friction between coils is greater than the amount of spring compression, so the spring remains locked in that position. The unit is reset by pulling the actuator rod and cap back, releasing the spring for further use.

SHOCK ABSORBER

Description and Operation

Figure 1 is a partial sectional view showing the placement of coils when winding the spring. Spring 11 is wound on straight rod-shaped mandrel 12 so that succeeding coils overlap preceding coils. The first coil A is overlapped by coil B. Coil B is then overlapped by coil C, etc.

Figure 2 is a partial sectional view of the impact limiter 10. As shown, the spring coils have been reversed, with the first coils overlapping and constricting the succeeding coils. Coil A now overlaps and constricts coil B. Coil B now overlaps and constricts coil C, etc.

The large end 13 of reversed spring 14 is provided with a loosely fitting essentially cylindrical cap 15, open at one end. Closed end 17 of cap 15 is rigidly attached to actuator rod 18 by nuts 19.

Reversed spring 14 with cap 15 is located in a substantially cylindrical housing 20 open at one end 22, and closed at the other end 24. Open end 22 of housing 20 is provided with a closure cap 23. Closure cap 23 is provided with a hole 25 to allow passage of actuator rod 18. Small end 28 of reversed spring 14 fits loosely around a boss 30 located in the bottom of housing 20. Boss 30 is essentially a slightly tapered truncated cone. Closed end 24 of housing 20 is also provided with mounting means 31.

Figure 3 is a partial sectional view of the device 10 showing the reversed spring 14 in the fully compressed stage.

In operation, actuator rod 18 of impact limiter 10 is depressed, compressing the reversed spring 14. As spring 14 is compressed, overlapping coils A, B, C and D are forced over each other, progressively increasing the area of contact. As the spring is compressed, large end 13 (outer coil) of spring 14 expands radially into contact with inner wall 16 of spring cap 15. At the same time, small end 28 (inner coil) of spring 14 is forced to contract radially around surface 32 of boss 30.

As shown in Figure 3, the point is reached when there is maximum frictional contact between inner wall 16 of spring cap 15, the coils A, B, C, D and surface 32 of boss 30. The total amount of friction at this stage is greater than the amount of spring compression, therefore the spring will remain locked in this position until reset.

To reset the spring, actuator rod 18 with spring cap 15 is pulled back, partially uncovering the nested coils. This allows the coils to expand radially, reducing the friction between coils. The spring then expands axially, returning to the starting position shown in Figure 2. To eliminate locking, spring travel is limited, preventing full spring compression.

TITLE

SHOCK ABSORBER

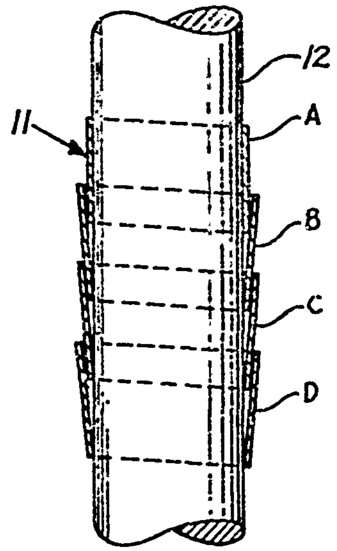

FIG. 1

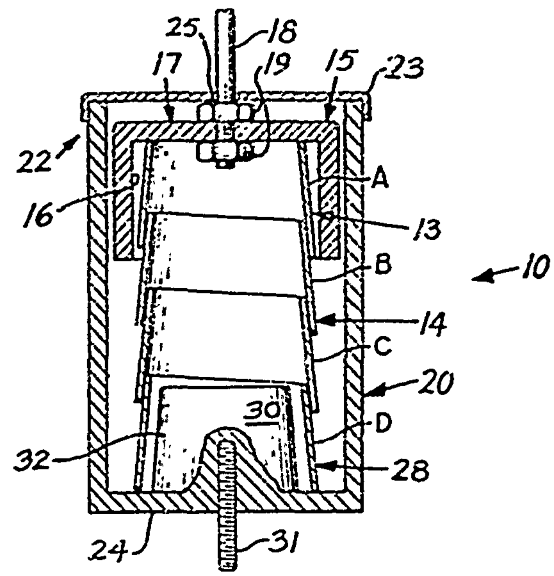

FIG. 2

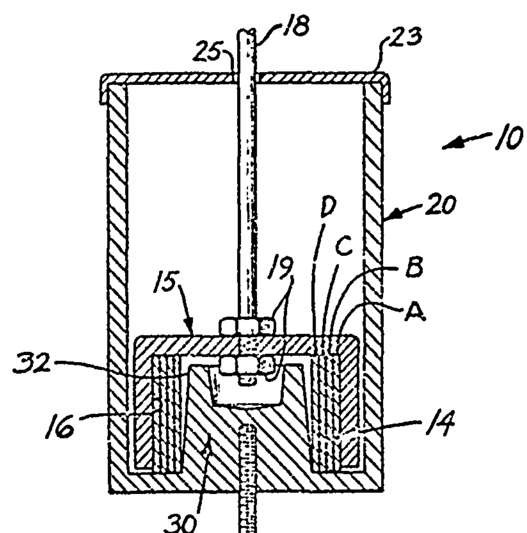

FIG. 3