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LOAD-LIMITING LANDING GEAR FOOTPAD ENERGY ABSORPTION SYSTEM

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ABSTRACT

As a precursor to future manned missions to the moon, an inexpensive, unmanned vehicle that could carry small, scientific payloads to the lunar surface was studied by NASA. The vehicle, called the Common Lunar Lander, required extremely optimized structural systems to increase the potential payload mass. A lightweight energy-absorbing system (LAGFEAS), which also acts as a landing load-limiter was designed to help achieve this optimized structure. Since the versatile and easily tailored system is a load-limiter, it allowed for the structure to be designed independently of the ever-changing landing energy predictions. This paper describes the LAGFEAS system and preliminary verification testing performed at NASA's Johnson Space Center for the Common Lunar Lander program.

INTRODUCTION

As NASA looks toward the future and the goal of a permanent manned presence on the moon, several smaller steps must be taken to achieve this goal. A scientific survey of the lunar surface, more detailed than the Apollo missions could accomplish, must be undertaken. One proposed way of performing this task is with a group of small, unmanned landing vehicles that could carry various scientific payloads to the lunar surface. A telescope, a soil sampler, or a small remote rover could be delivered and could carry out the necessary exploration. NASA imagined a common vehicle, capable of multiple tasks, and dubbed this vehicle the Common Lunar Lander (see Figure 1). This lander had to be inexpensive, which meant the use of a small, commercial launch vehicle such as McDonnell-Douglas' Delta rocket was necessary. The lander must also be extremely efficient to deliver the greatest payload mass possible to the moon. In 1992, the Structures and Mechanics Division at NASA's Johnson Space Center underwent a six-month design study to explore the feasibility of designing and flying such a vehicle.

As the study progressed, it was found that the structure of the vehicle became a major driving force toward the vehicle's efficiency. All of the vehicle's components were attached to the structure, and therefore had to be integrated into the structure. Any inefficiencies in the structural design would be amplified in effect by all of the other system components. In addition to these difficulties, the landing environment for which the structure had to be designed was very difficult to quantify early in the project. The loads induced into the lander and its payload are a function of the mass of the lander, the impact velocity of the lander, the surface properties, the vehicle's inertia, the radar quality, and many other factors. To make a truly optimized structure, this load environment must be well understood, and early in the design process this is not the case. Since the load environment is a strong function of the design itself, a very intensive, time-consuming iterative design process must be used to achieve the most optimum vehicle design.

Since much of the problem was centered on quantifying the landing load environment, a unique solution to the problem was created. The solution was called the Load-limiting Landing Gear Footpad Energy Absorption System, or LAGFEAS. Typical landing shock absorbers are velocity dependent and heavy. The LAGFEAS provides a simple, easily modified, and load-limiting mechanism to absorb the landing shock. Because the system is load-limiting, the maximum load input into the structure is not dependent on the landing velocity or vehicle mass, only on the parameters of the energy absorbing system itself. This offers a great advantage to the designer. A maximum load value can be chosen and the rest of the structure can be designed. As long as enough stroke is allowed, the G-levels seen by the structure and the payload can be controlled. With the maximum loads known, the rest of the structure can be designed long before landing velocities or vehicle weight can be accurately determined. With this system in place, the Common Lunar Lander design was initiated.

Landing Requirements

The expected mass and landing velocities of the lander were conservatively evaluated. These velocities corresponded to the lander's ability to land with a functioning radar device and the energy absorption system must remove all the energy present in the lander. This energy is in the form of kinetic energy, which is a function of the velocity in two directions; a vertical direction, and a horizontal direction. The energy absorption system must remove all the energy present in the system for it to complete its function under all reasonable landing conditions. During the early phase of the design, two stringent requirements were placed on the landing system:

1. One footpad must be capable of absorbing all of the energy present in the lander system.

2. The g-loading experienced during the landing phase must remain at or below the same g-loading experienced by the lander during the launch phase.

Requirement number 1 above is placed on the landing gear system to account for any unexpected landing conditions. If a large rock or ditch is hit, then one landing gear could conceivably be expected to absorb a majority of the energy. The lander has three legs, and all three landing pads would nominally act in absorbing energy, but by assuming that one energy absorbing pad can absorb the entire landing energy, a degree of safety is obtained for many varied and unexpected landing conditions.

Requirement number 2 is placed on the landing system in order to allow the future payloads and the lander itself to be designed. Since some of those future payloads are expected to be highly sensitive and potentially fragile, such as a telescope, the loading on those payloads must be kept as low as possible. The loads experienced by the payloads during launch are well known and pre-determined by the launch vehicle choice, therefore, by assuming that the landing loads must be equal to or lower than the loads seen at launch, a reasonable requirement for the landing loads is obtained.

The g-load seen by the structure is a function of the force applied to the vehicle to slow it down, and the stroke over which that force is applied. Therefore, since the maximum g-loading on the lander is determined by parameters of the landing energy absorption system alone and is relatively independent of other environmental conditions. Because both the energy and the strength requirements are defined, it is possible to design a landing gear energy absorption system. Due to the relatively small stroking distances needed, based on landing velocities, the approach taken on designing a system was to place all energy absorption in the footpads. This precludes the use of shock absorbers in the leg members which can be heavy and in some cases more complicated. This is a new approach for lunar-type landers which have used shock absorbers as the main energy dissipater. However, the design team felt that a significant weight savings could be obtained by utilizing the approach of placing all of the energy absorption in the footpads.

Energy Absorption System

A unique system has been designed that satisfies all the energy absorption requirements. Figure 1 shows the proposed Common Lunar Lander and the location of the Energy Absorption System. Figure 2 shows the major components of the system, to be described in detail in the following section.

Friction Rod and Washers

The main component of the system uses the friction between a traveling rod and press-fit washers to absorb the vertical energy. As shown in Figure 3, the washers are pressed onto the rod and spaced some distance apart. The kinetic energy of the applied loading can be resisted at the desired load level and dissipated through heat generated by the traveling friction. The washers are initially picked up one at a time, by varying the spacing and the number of the washers, the onset rate and the total friction load can be controlled. The friction rod/washer system was originally developed by NASA for use in the Apollo command-module couch struts. The design used a series of small washers placed on a 9.5 mm (3/8 inch) rod as the energy absorber and provided an acceptable g level and onset rate to the crewmen. The right materials are crucial to the success of the system. Material compatibility, especially the relative hardness between the rod and washers, is important. Various materials and lubricants were considered and tested. The best material combination found was 718 Inconel rods (heat treated to Rc 40) and fully annealed 416 stainless steel (SST) washers (R_B 83). The two materials have relatively equal Young's modulus but the yielding strength of the rod is three times that of the washers. Drill rod, 17-4 PH stainless steel rod and 304 stainless steel washers were all tested and discarded because of galling and thermal effects. To achieve the desirable frictional coefficient, a boundary (thin-film) lubrication was applied. For the thin-film, boundary lubrication, the friction coefficient falls within the range of 0.05 to 0.15. The highest friction coefficient occurs when the interface pressure becomes so great that the lubricant film can no longer support the load. Some wear will occur, however, the wear should not be visible to the eye and severe wear is abnormal and visible. Various lubricants including high-quality oils and greases were tested without success. Finally, the Miller Stephenson dry-film lubricant MS-122 successfully produced the desired results and proved to be highly repeatable.

For the application of the landing gear energy absorber, the sizes of the rod and washers have to be drastically increased compared to the Apollo tests. To meet the strength (bending and buckling) and deformation requirements, the diameter of rod was increased to more than 2.8 cm (1.1 in). To optimize the weight of the system, the rod was also hollowed. The outer diameter of the washers were designed to be twice their inner diameter and the inner diameter of the washers were manufactured to 3% less than the outer diameter of the rod. The elastic limit on the strain of the SST washer material is 0.1%. Based on analysis, the 3% interference will result in plastic yielding of the entire washer. Using the Von Mises yielding criteria, the normal compression between the washer and rod is approximately 80% of the yielding strength while the entire washer has already been plastically deformed. The washer works like a stiff but elastic rubber band which provides a constant normal force (grasping force) between the washer and rod. The plastic strain will not reach the point of rupture because the fully annealed washer has an ultimate strain of 30%. The elastic springback of the washer is also an important factor for consideration. Because of the high stiffness of the washer material, the load will vanish quickly if the rod diameter (nominal 284.5 mm) decreases 0.03 mm. The maximum allowable variation of rod diameter in manufacturing is defined to be 0.005 mm (0.0002 in) (15% of the elastic springback). Due to this small tolerance and the fact that the inconel rod has a higher thermal expansion coefficient than the SST washer, thermal effects may not be ignored. The interface compression will be significantly reduced or completely released if the system is exposed to a relatively low temperature. A light weight insulation cover or heaters can be added to resolve the problem.

The inner edge of the washers are also rounded (0.25 mm radius) to help in the installation and prevent galling. When a washer is stroked, the applied loading has to overcome the initial static friction of the washer. Afterwards, the washer will move at a lower and constant sliding friction. The thickness of the washer shall not be too great to induce a high static friction. A thicker washer also requires higher installation load. However, the washer may buckle or warp if it is too thin. The washer thickness for the Common Lunar Lander energy absorber will be between 3.8 to 5.1 mm. The washer tested for the design concept was 5.1 mm. A 3.8-mm-thick washer may be even more suited for the system, but has never been tested due to lack of resources. As stated previously, boundary lubrication generally yields coefficients of friction in the range of $\mu = 0.05$ to 0.15. The friction coefficient is dependent on various parameters including the stroking velocity. After the basic design of the energy absorber is established, the best way of obtaining and verifying the friction load of the design is to perform the actual hardware testing.

Because of a resources constraint, the testing program was performed in a very limited fashion. Nevertheless, the tests were considered very successful. By using the available materials in the shop, four test specimens were manufactured, assembled and tested. The sizes of the test specimens are tabulated in Table 1 below.

Table 1. Sizes of Washers and Rod Test Specimen

	Outer Radius, Ro, mm	Inner Radius, Ri, mm	Thickness, Plate, mm		
Washer	27.559	13.780	5.080		
			Thickness, Tube, mm	Length, cm	Washer-Rod Interference
Rod #1	14.133	6.350	7.772	26.67	2.50 %
Rod #2	14.133	7.938	6.198	26.67	2.50 %
Rod #3	14.282	7.938	6.350	26.67	3.50 %
Rod #4	14.282	9.119	5.156	26.67	3.51 %

The purpose of the test was to verify the concept and to evaluate the maximum load and the total energy absorption of the design. Because of the flexibility of the design, the sizing of the washers-and-rod system can be easily modified and tailored to meet the final design requirements. Four 5.08-mm-thick washers were installed on each rod, the 2.50% interference sufficient to induce a full plastic deformation of the washer. The washers were spaced 5.08 mm apart using the installation procedure shown in Figure 4. A static stroking test (0.38 $^{mm}/_{s}$ rate) was also performed to record the static and sliding friction loads during the installation of the washers. A typical load vs. displacement curve for a single washer (rod #4) is shown in Figure 5. Under the low stroking speed, the static friction load was 8563 N (1925 lb) and sliding friction was 2224 N (500 lb) for the washer. A dynamic weight-drop test was consequently performed. As shown in Figure 6, the total weight (2847 N (640 lb)) and the drop height (39.80 cm) were determined based on the kinetic energy and landing velocity. Because the available stroking distance of the rods was limited, each specimen was tested to half of the design landing energy (1133 N•m) with the same landing velocity (279.4 $^{cm}/_{s}$). A typical result (load vs. displacement) of the drop test is also shown in Figure 6 (for rod #4). The maximum load of the washer stack was about 17.79 kN (4000 lb) with additional spikes of a single washer. A specimen was tested at a higher energy level (50.8 cm drop height) and the maximum stroking distance was 12.7 cm for the washer on the bottom of the stack. The design capability for the legs of Common Lunar Lander was 12 kN (2700 lb), however, by using thinner washers, the system should easily meet the specific design requirements. No additional tests were performed.

Horizontal Energy Absorption

To absorb the energy in the horizontal direction, a material deformation system is used. Several materials were considered, but the material selected would be a honeycomb-type material. This type of material crushes in a way that also creates a load-limiting system. Once the honeycomb crushes to a certain load value, it continues to crush up to 70 percent of its volume at a constant load. The load will never go above this value until 70 percent of the material has been crushed. As long as enough material has been used, this system will act as a predictable load-limiter in the same fashion as the friction washers. In the system, the friction rod is enclosed within the footpad. Inside the footpad, a block of honeycomb material surrounds the rod. The rod is free to move inside of this material, being able to freely slide along the top of a stiff, honeycomb plate. When a side load is placed on the footpad from impact, the rod will crush through the honeycomb sections, absorbing the necessary energy. The bottom of the footpad will also be shaped to allow the pad to slide along the surface as much as possible. This sliding helps dissipate energy and gives the system more capability. To the bottom of the friction rod is attached a small sliding plate that is allowed to rotate with a ball joint. This allows the sliding plate to remain in contact with the honeycomb top plate at all times. The honeycomb is contained in a restraining cylinder. A very good material candidate is available commercially under the brand name DUOCEL®, and is an isotropic foam metal. The parent material can be selected and heat treated from a variety of materials and processes, including most forms of aluminum. The foam metal behaves in the same fashion as honeycomb under compressive loads. This load-limiting behavior retains the unique nature of the energy absorption system. The foam metal may be cut into sections to avoid placing portions of the metal into tension as the rod crushes through. The general mechanical properties under compression of the DUOCEL® were evaluated with three tests. Each specimen was 10.16 cm by 3.81 cm by 1.91 cm in size made of an aluminum alloy. The porosity of the specimen was 3.937, 7.874 and 15.748 pores per cm and the density was 8%, 12% and 12% respectively. Each specimen was compressed with the Instron machine with a rate of 0.2 $^{mm}/_{s}$. The results showed a constant stress portion extending over a 50% strain range for all three tests. A typical stress-strain curve of the test results is shown in Figure 7. The DUOCEL® is certainly the leading candidate for the horizontal energy absorber.

Yielding Rod Energy Absorption System

Another method of absorbing the horizontal energy has also been proposed. The **yielding rod** energy absorption system is shown in Figure 8. The system also uses the friction washers and rod to absorb the vertical energy. However, instead of letting the rod stroke through the washers, this system uses a stationary rod with traveling washers. The horizontal landing energy is absorbed by the plastic bending of a yielding rod. Crushable honeycomb materials are also used for the additional cushion and energy absorption. The solid circular section of the rod provides a high capability for plastic bending. The yielding rod basically replaces the metal foam of the previous system as the horizontal load-limiter and the energy absorber. Located at the root of a cantilever beam, the rod will carry the maximum bending moment from the lateral loading. The rod is sized based on the load and the energy requirement. The strong (F_{ty} = 1165 MPa) and tough (e_{ult}=16%) Inconel 718 was selected for the yielding rod. Based on the loading and energy requirements, and the moment capability at full plasticity (plastic bending moment) of the circular rod, the size of the rod was determined. For the design environment of the Common Lunar Lander, the radius of the rod was analyzed to be 11.68 mm. The rod would be required to bend by only 45 degrees to absorb the entire horizontal energy for the worst case of landing. The outer diameter of the tubular friction rod was 16.51 mm, which provided far stronger sectional properties than the yielding rod. The horizontal loading on the structure is limited by the plastic bending strength of the yielding rod and the minimum length of the footpad (moment arm) during and after stroking. A bend guide is also included in the system. Based on the radius of the bend guide and the size of the yielding rod, the maximum/ minimum elongation of the rod can be easily evaluated. Many tough materials can also be considered for the yielding rod application, depending on the loading and energy environments. When the horizontal landing energy becomes significantly high, this system is advantageous in weight saving.

CONCLUSIONS

In 1992, The Structures and Mechanics Division at NASA's Johnson Space Center underwent a design study to determine the feasibility of building a Common Lunar Lander which met certain, strict requirements. These requirements meant that innovative solutions had to be found to increase the lander's structural and operational efficiencies and decrease the system's mass. The LAGFEAS was designed with this goal in mind. To prove that the system is workable, and ultimately a benefit to the lander, a significant amount of testing would be necessary. Due to funding problems and finally a cancellation of the project, this testing was never completed. However, it was felt by the authors that this system was a unique approach to the problem of absorbing impact energy from landing systems and therefore deserved to be presented to the mechanisms community.

Using the plastically deformed washers to control the frictional force between the friction rod and the washers creates a system that is by design very tolerant of temperature fluctuations. In some extreme cases, a low temperature environment can affect the friction force between the rod and the washers. In these cases, a lightweight thermal insulation would solve this problem. In addition, it is felt that the coefficient of friction between the rod and the washer can also be carefully controlled in a vacuum environment. The problem of cold-welding of the materials can be avoided through the use of a captured dry-lubricant between the rod and the washers, also part of the original design. Further thermal-vacuum chamber tests would be performed to verify the system's functionality during future development tests.

REFERENCES

1. Common Lunar Lander Detailed Design Study, Structural Design and Integration Report to the Artemis Engineering Team, NASA document number JSC-26094.

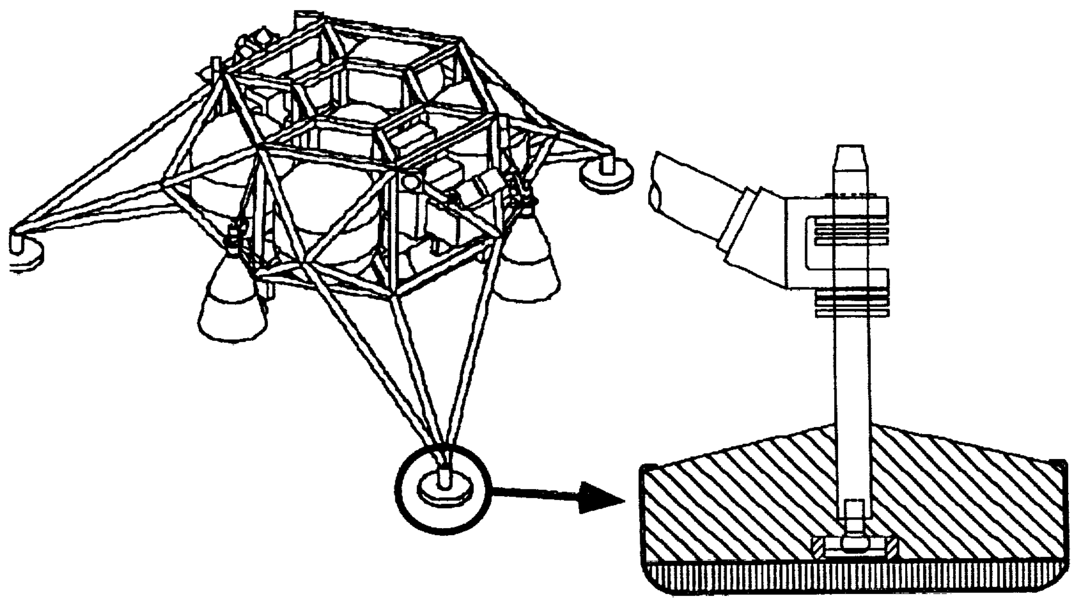


Figure 1: Common Lunar lander with Energy Absorption System at Inset.

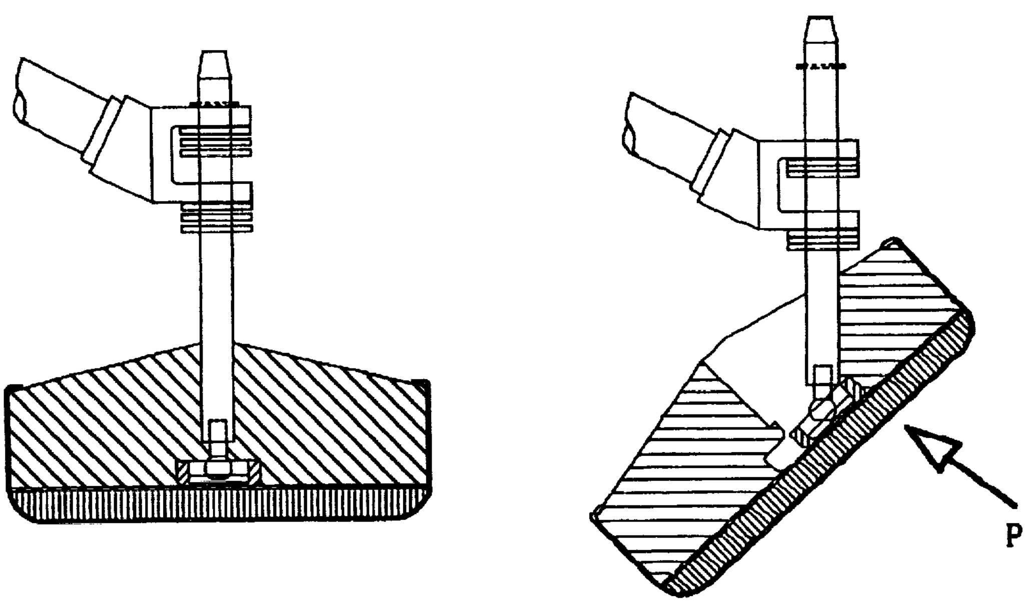
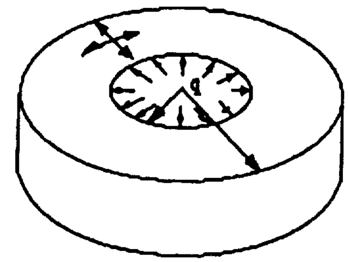


Figure 2: Landing Gear Footpad Energy Absorption System

Apply interference fit beyond the material elastic limit to induce a full over-strain of the washer; The ductile material will not rupture with sufficient ultimate strain



Yielding Criteria (Von Mises)

$$\tau_{max\ shear} = \frac{\sigma_1 - \sigma_2}{2} = \frac{\sigma_\theta - \sigma_r}{2} = \frac{\sigma_{yield}}{\sqrt{3}}$$

The interference normal compression q becomes

$$q = \frac{\sigma_{yield}}{\sqrt{3}}\left(2\ ln\frac{R_o}{R_i}\right)$$

The interface normal force N is

$$N = 2\pi R_0\, t\, q$$

The ideal washer friction F becomes

$$F_{static\ or\ sliding} = N\ \mu_{static\ or\ sliding}$$

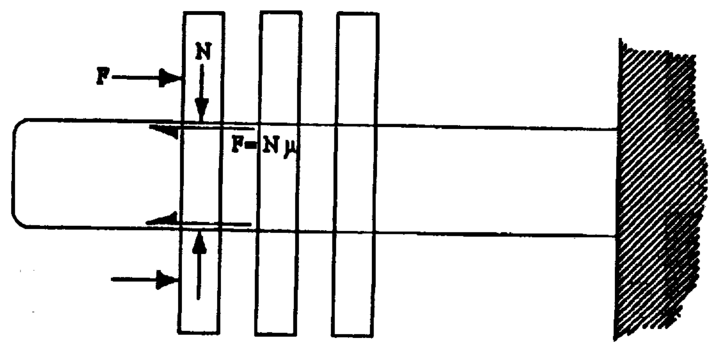


Figure 3: Washers and Rod Energy Absorption System

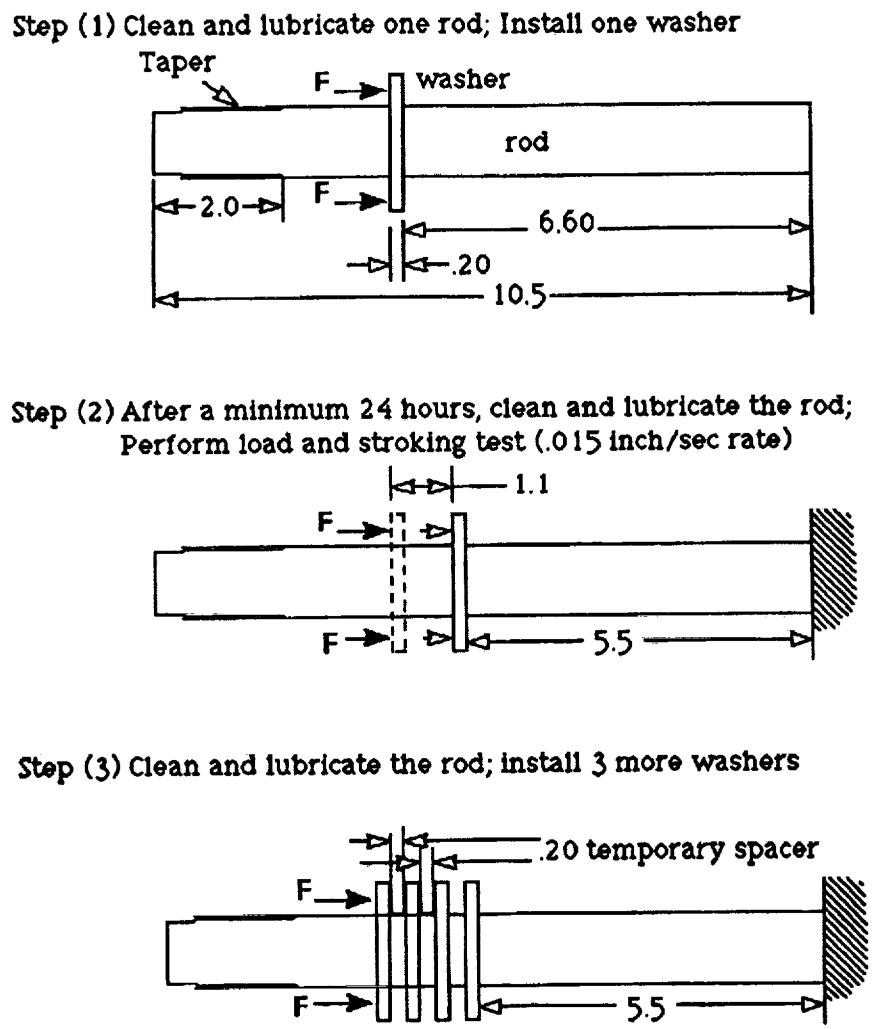


Figure 4: Configuration of Washers and Rod Test Specimen

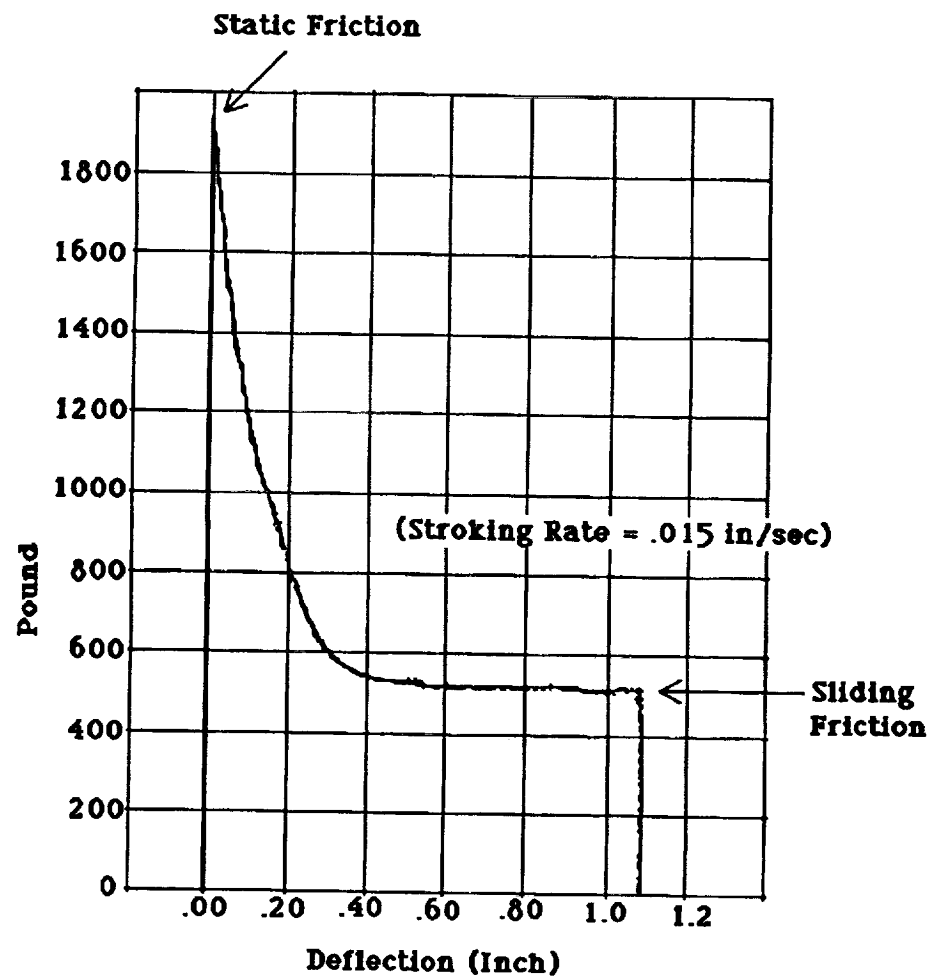


Figure 5: Typical Load Vs. Displacement Curve for a Single Washer

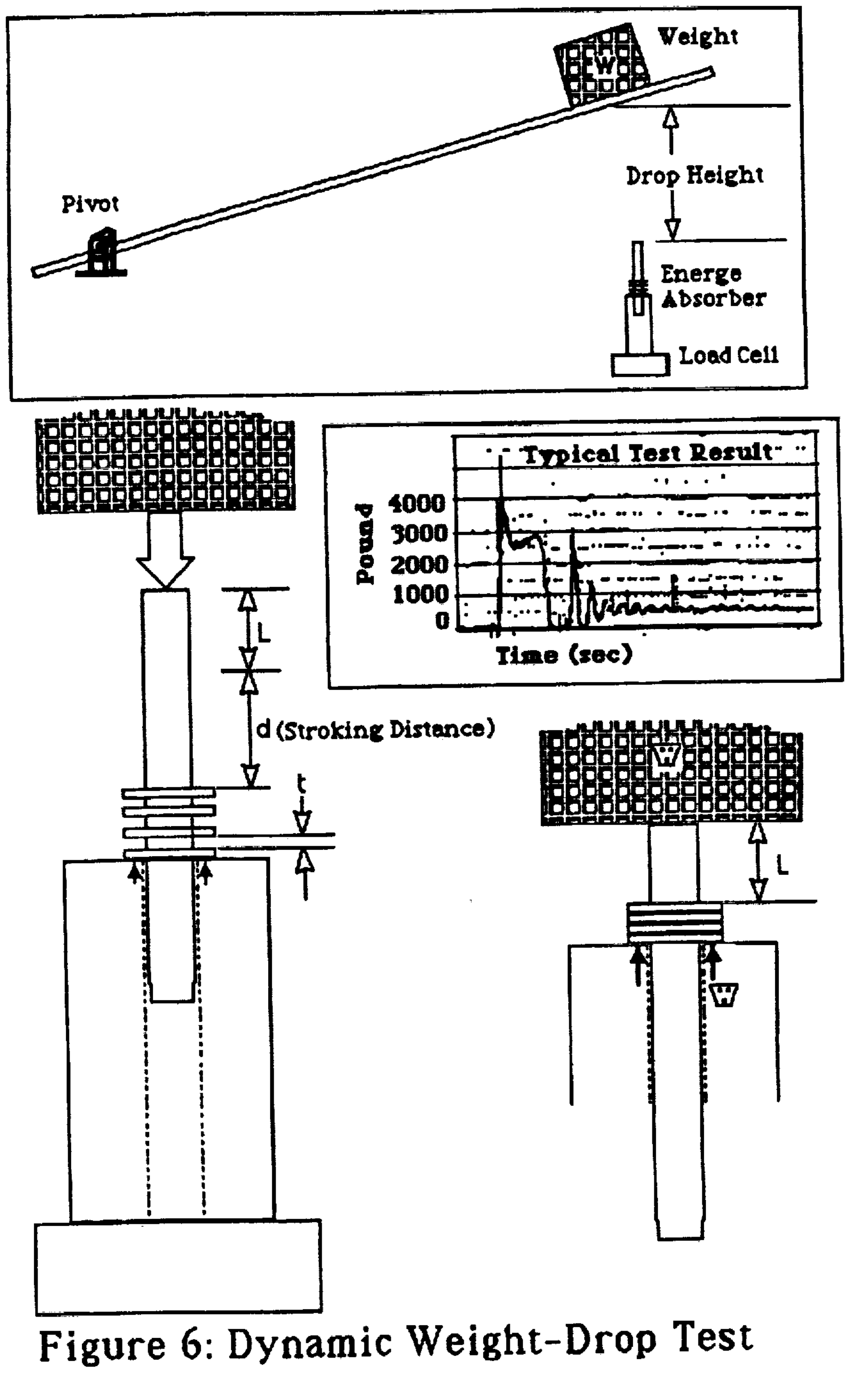
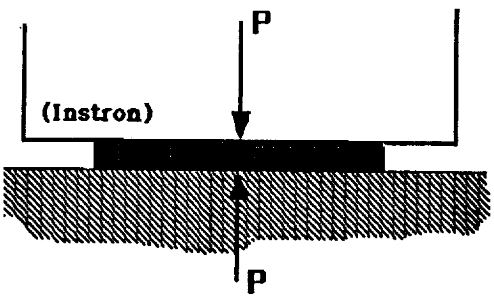


Figure 6: Dynamic Weight-Drop Test

* Rigid, highly porous and permeable and has a controlled density of metal per unit volume
  * Independently variable porosity from 10 to 40 pores per inch
  * Independently variable density from 3 to 20 percent of Aluminum
* Completely isotropic load response
* High strength to weight ratio
* Impact energy absorption application with constant stress portion extends over a 50 percent strain range



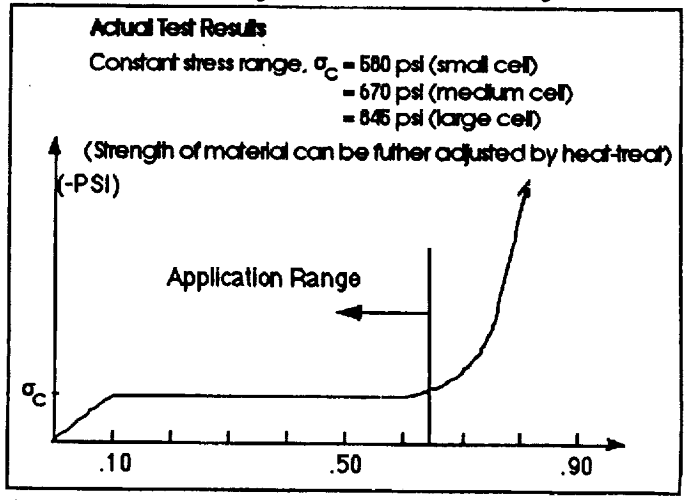


Figure 7: Typical Properties of Isotropic Open-Cell Foam Metal

* The primary vertical landing energy is absorbed by friction washers and rod
* The horizontal landing energy is absorbed by crushable honeycomb and plastic bending of a yielding rod

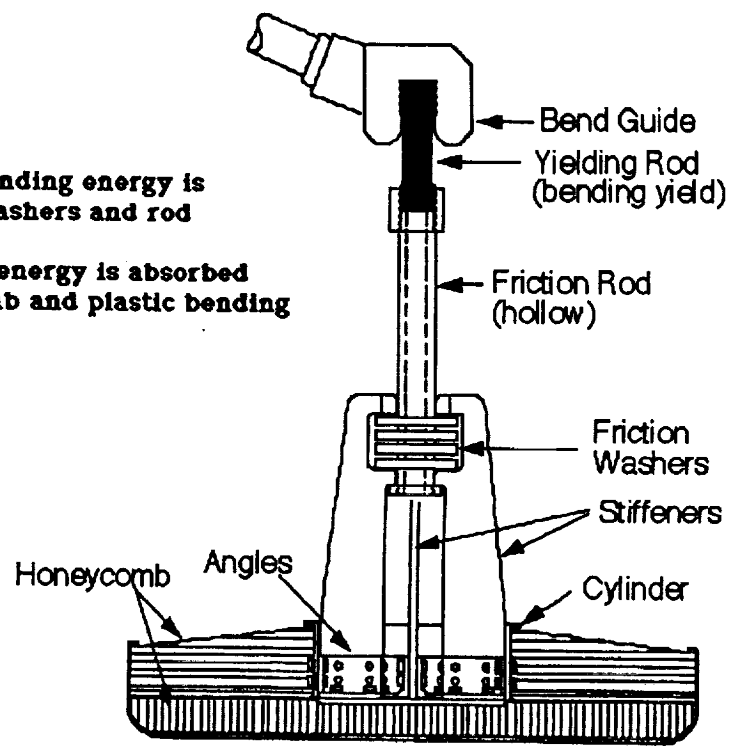


Figure 8: Yielding Rod Energy Absorption System

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The proceedings of the 28th Aerospace Mechanisms Symposium, which was hosted by the NASA Lewis Research Center and held at the Cleveland Marriott Society Center on May 18, 19, and 20, 1994, are reported. Technological areas covered include actuators, aerospace mechanism applications for ground support equipment, lubricants, pointing mechanisms joints, bearings, release devices, booms, robotic mechanisms, and other mechanisms for spacecraft.

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