This is a preprint of a paper intended for publication in a journal or proceedings. Since changes may be made before publication, this preprint is made available with the understanding that it will not be cited or reproduced without the permission of the author.

RECEIVED BY JIC FEB 1 1972

UCRL - 13568

PREPRINT

CONF-711208--6

MASTER

Lawrence Radiation Laboratory
UNIVERSITY OF CALIFORNIA
LIVERMORE

VTAB: A PASSIVE IMPULSE GAGE

W. J. COMFORT

November 24, 1971

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

THIS PAPER WAS PREPARED FOR SUBMISSION TO
THE UNDERGROUND NUCLEAR TEST MEASUREMENTS SYMPOSIUM I
SANDIA LABORATORIES ALBUQUERQUE, DECEMBER 7-9, 1971

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

W. J. COMFORT, Lawrence Livermore Laboratory, Livermore, California 94550

ABSTRACT

A passive gage has been developed for impulse measurement in a nuclear environment. It can be designed to be unaffected by ground motion for typical amplitudes of 30 to 50 g. Design tradeoffs can be made to extend this level to 1,000 g or more, depending upon the sample and impulse levels to be measured. A single gage can be designed so that its measuring range is 15 to 20 times its threshold value. The configuration of the gage is a cantilever beam with a sample attached to the free end. The fixed end bends over a curved anvil of known radius which serves two purposes: (1) it defines the maximum strain in the beam so that the ultimate strain is not exceeded, and (2) it causes plastic bending to occur over some distance, rather than at a point. By increasing the width of the beam linearly, starting at the fixed end, the calibration curve of impulse versus angle of bend can be made nearly linear rather than having the parabolic shape characteristic of many passive impulse gages. An associated computer code, VTAB, is used to calculate design parameters for the gage and predict its response.

INTRODUCTION

Impulse measurements in underground nuclear tests often require specialized gages that must perform reliably. Active as well as passive gages have been designed for this purpose with varying degrees of success. The passive gage presented here is reliable, easily designed and built, analytically predictable in its response, and able to withstand the underground nuclear test environment. It is an improvement of the cantilever tab design used previously by others. The original design, Figure 1(a), required the beam to bend about a sharp corner, whereas an improved version, Figure 1(b), has a curved anvil for the beam to bend around.

One difficulty with the original design was that it was not as analytically predictable as desired, mainly because of the likelihood of exceeding the ultimate strain of the beam material due to the sharp bending, with the resultant breaking of the beam. The present design, with the beam bending over a known radius, possesses several advantages: (1) the range of the gage is increased, (2) the response curve can be predicted and can be made linear if desired (by tapering the width of the bending portion of the beam), (3) the maximum strain in the beam can be limited to a known value, eliminating beam breakage, and (4) the strain does not occur at only one point but rather over a region. The design shown in Figure 1(c) has been replaced by the design

Figure 1
Variations of the Passive Impulse Gage

(a) Original design allows beam to bend sharply.

(b) Improved design with curved anvil limits beam strain.

(c) Tapered hinge section gives linear response curve.

Side View

Top View

shown in Figure 2. The effectiveness of the tapered section is unchanged, while lateral stability of the beam is greatly enhanced.

Additional improvements have been made, compared to previous designs in which rivets were used for holding the sample. The new design, shown in Figure 2, not only holds the sample securely, but also allows full sample exposure.

Comparisons of NTS data obtained from active impulse gages and from the passive VTAB gages are quite good.

THEORY

Predictability of the impulse gage response is dependent upon the sophistication of the theoretical model and the accuracy of the material properties used in the calculations. Fortunately, because of the nature of the mechanics of the problem and the opportunity for choice of materials, the characteristics of the system can be designed with reasonable confidence with even the simplest model.

The basic approach presented here uses energy methods rather than dynamics. The two assumptions involved for all of the theoretical models are: (1) simple bending equations, with edge and end effects neglected, apply, and (2) the surface between the bending section of the gage and the cylindrical anvil around which the gage bends is lubricated; i.e., no shear develops at the surface of the beam due to contact with the anvil.

The simplest model consists of an elastic, perfectly plastic idealization of the beam material and a beam of constant width. For many problems this is a sufficiently valid approach, and where access to a computer is either difficult or unwarranted, it represents the most practical solution. This model leads to the following expression for the plastic moment M developed in a rectangular beam in simple bending:

$$M = \frac{bh^2}{4} \sigma_y \left[1 - \frac{1}{3} \left(\frac{y_e}{h/2} \right)^2 \right] \quad (1)$$

where b is the width of the beam, h its thickness, σ_y the yield stress of the material, and y_e the distance from the neutral axis to the point where the stress in the beam reaches σ_y . Since typical designs would force the material well into the plastic region, y_e becomes very small compared to $h/2$, thus allowing the second expression within the brackets to be neglected. The resulting simplification is

$$M = \frac{bh^2}{4} \sigma_y. \quad (2)$$

Figure 2
Typical VTAB Design

It is important that the material be kept from reaching or exceeding its ultimate strain in the outer fiber of the beam. For this reason, an appropriate radius R_0 is chosen for the anvil around which the beam is deformed. The expression for the relationship between the maximum strain in the outermost fiber (ϵ_D) and the radius about which the beam bends is

$$R_0 = \frac{h}{2} \left(\frac{1}{\epsilon_D} - 1 \right). \quad (3)$$

The neutral axis is halfway through the thickness of the beam, so its radius (r) is $R_0 + h/2$, or

$$r = \frac{h}{2\epsilon_D}. \quad (4)$$

In order to express in a mathematically tractable form the relationship between the energy absorbed in the plastic moment and the kinetic energy introduced into the end of the beam, some simplifying assumptions may be made for this model: (1) the mass m at the end of the beam traveling with velocity v is a point mass, (2) the beam is weightless, and (3) the bending of the beam around the anvil does not produce sufficient shortening to affect its behavior. Thus, the following expression may be derived:

$$\frac{1}{2} mv^2 = M\theta \quad (5)$$

where θ is the angle through which the beam is bent.

To take into account the elastic energy absorbed in the beam before bending, the expression may be derived for the elastic energy (E_e):

$$E_e = \sigma_y^2 \frac{bh\ell}{9E} \quad (6)$$

where ℓ is the unbent length of the hinge section and E is the elastic modulus of the material. Equation (5) may then be written

$$\frac{1}{2} mv^2 = \frac{bh^2 \sigma_y^2 \theta}{4} + \sigma_y^2 \frac{hb\ell}{9E}. \quad (7)$$

Rearranging Equation (7) in terms of momentum per unit area, where A is the area over which the specific impulse I is distributed, one can write the final expression for this simple model as

$$I = \left[\frac{2mbh\sigma}{A^2} y \left(\frac{h\theta}{4} + \frac{\sigma y \ell}{9E} \right) \right]^{1/2} \quad (8)$$

As a consideration for design, the smallest impulse measurable by the gage is one for which θ is greater than zero, i.e., one that leaves the gage bent to some small extent.

COMPUTER CODE VTAB FOR GAGE ANALYSIS

For a more precise theoretical model of the gage and a more versatile and accurate prediction of its calibration curve, a computer solution may be used. The following considerations may now be included: (1) the rotational inertia of the beam and the sample is considered, (2) the beam length shortens as a linear function of the bend angle, (3) the nonlinear stress-strain curve is used; (4) the plastic moment varies as a function of r in going from the straight beam ($r = \infty$) to the final radius required by the anvil, and (5) elastic "springback" of the gage beam is considered.

Analytical expressions for many of these effects can, of course, be developed. In this case, however, a computer solution was chosen as a simpler technique. The computer code VTAB was written in order to consider all of the above effects in predicting the calibration curve and in doing the gage design work. The code considers the case where the bending portion of the hinge widens linearly with distance along the hinge from the fixed end, as shown in Figure 1(c). The reason for this modification is that the calibration curve approaches linearity as the b/b_0 ratio increases, instead of having the parabolic shape it would have with a beam of constant width. This modification improves the resolution of the measurement at low impulse levels.

To understand how a tapered beam width gives a linear response curve, consider the following: Rearranging Equation (5) in the form

$$I = \sqrt{\frac{2m}{A^2}} \int M d\theta$$

one can see that if θ (under the radical) could be raised to the second power, the equation would be linear. In order to do this, it is possible to introduce a linearly increasing section as a function of θ ,

$$b = k\theta,$$

where k is a constant. This would linearize the equation perfectly, except that in the physical situation the width cannot start at zero; thus the equation of the width must be

$$b = b_0 + k\theta$$

where b_0 is the initial width of the beam. Furthermore, the elastic terms and various other complications reduce the linearity somewhat; but, as may be seen in Figure 3, the linearization technique is quite effective.

For most problems, strain rate effect appears to be minimal since the strain rates usually encountered are on the order of 10^3 in./in.-sec or less. Most aluminums (which happen to make good hinge materials) do not show significant strain rate effects below 10^3 in./in.-sec. In the event strain rate were a problem, the code could be modified to include strain rate effects without much difficulty.

GAGE CHARACTERISTICS

One of the greatest concerns for good design involves the requirement that the gage withstand handling and environmental loading without giving erroneous data. Consideration of the beam thickness and length, and to a lesser extent the width of the hinge portion of the beam and the sample mass, must be traded off against one another for most effective designs. This is the reason for making the beam length and sample mass as small as possible while making the hinge thickness and width as large as possible. A reasonable approximation of the maximum g-loading the beam will withstand without deformation is given by the following expression:

$$n \approx \frac{\sigma_y h^2 b}{6mgL} \quad (9)$$

where n is the number of g's and L is the distance from the anvil to the center of mass of the sample. Good design requires a tradeoff between the g-loading expected, the beam hinge proportions and gage measuring range.

The gage can be designed to withstand rather severe g environments. For example, it can be designed to withstand acceleration of 30 or more g's normal to the sample at any angle of bend with no change in the existing bend angle. The gage corresponding to this design has a range of 220 to 5,200 taps and a sample areal density of 4 g/cm^2 . Sensitivity to g-loadings is greatly reduced after impulse produced deflection of the beam about the anvil has occurred. This is because of the reduction in the moment arm and, in the case of the linearly varying width beam, the increased width of the beam at the fixed end of the cantilever configuration. Because ground motion arrival times are long compared to gage response times, the post impulse sensitivity can be used as another tradeoff possibility. For example, a gage with a sample areal density of 2 g/cm^2 and an impulse range of 34,000 to 61,000 taps can be designed to withstand loadings of 1,000 g. The reduced range is an obvious disadvantage as a trade for reduction in g-loading sensitivity; however, the example is used to show the effect of tradeoffs in this case. This philosophy, however,

Figure 3
Typical VTAD Calibration Curve

implies that the impulse must exceed 34,000 taps in order to give valid data if the ground shock environment is 1,000 g. The same gage and sample would have a range of 2,200 to 61,000 taps if the g environment were less than 200 g.

The maximum impulse, which the gage is capable of measuring, can normally be designed to be about 15 to 20 times the threshold value, which is the point at which the elastic limit of the beam is first exceeded and permanent bending results. This wide range leaves considerable margin for error in predicting the impulse to be measured.

GAGE PERFORMANCE

Experimental calibrations¹ have been run using an air gun accelerated projectile for imparting momentum, and a ballistic pendulum for measurement of momentum. The technique is highly successful and repeatable. Comparison of the VTAB code calculated curves and the calibration curve indicate that the VTAB code prediction is approximately 15% low. This error can be reduced by increasing the sophistication of the model; however, for design purposes, 15% is sufficient.

Averages of impulse measurements of samples of 2014 aluminum fielded on both VTAB's and active LDT (LLL) gages agreed to within better than 5%. The samples were fielded at Station 2B in the Mint Leaf event, and the experimentally derived calibration curve was used for the VTAB gages.

CONCLUSIONS

This passive impulse gage has several important advantages, including relatively low cost for applications in the hostile environment near an underground nuclear explosion. Either form of the gage, i.e., with constant width or tapered hinge portion, has a response that can be accurately predicted and covers a usefully large range. The gage is relatively insensitive to the extraneous shocks that it may experience in handling or during the experiment. With the tapered hinge section, the gage has a nearly linear response curve, which gives improved resolution at the lower end of its measuring range. The VTAB code makes it possible to quickly design and build an appropriate gage of this type for any specified impulse-measuring application.

REFERENCES

- 0 Work performed under the auspices of the United States Atomic Energy Commission.
1. C. W. Sullivan, Lawrence Livermore Laboratory, private communication (April 23, 1971).

DISTRIBUTION

Lawrence Livermore Laboratory Technical Information Department

15

W. J. Comfort

36

NOTICE

"This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately-owned rights."