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Considerations in the Experimental Determination of Constitutive Parameters for Finite Strain Plasticity

Norris J. Huffington, Jr.

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Various experimental procedures for measurement of quasi-static constitutive parameters for finite strain elastoplasticity are discussed in relation to counterpart finite element analyses of specimen deformation. It is concluded that the usefulness of such analyses is limited by the assumptions made in the formulations of current computer codes.

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1. INTRODUCTION

There remain significant difficulties in the measurement of elastoplastic parameters for use in analysis of finite straining of relatively ductile materials. Ideally, one would prefer use of tests in which a single stress component (at a time) could be varied as a function of an associated strain component during which measurements of applied loads and corresponding deformations could be made on a test section of reasonable size in which a state of homogeneous stress exists. However, conventional uniaxial tests have shortcomings which limit their usefulness. Tension tests are limited to relatively modest strains by the inception of necking. Compression testing involves overcoming friction problems on end surfaces in order to obtain uniform axial stresses on these surfaces and avoid "barreling" (or interrupted testing of re-machined specimens). The torsion test is attractive in that shearing strains of 600% and greater have been reported for thin-walled tube specimens but also presents experimental and interpretational problems which will be discussed.

It may be recalled that Poynting (1909) studied finite torsion of wires while Swift (1947) performed tests on solid and hollow rods. Both reported an elongation of their specimens under finite twisting. Subsequently, Lindholm et al. (1980) (Johnson et al. 1983) employed a torsion specimen of the form shown in Figure 1, in particular for determination of material parameters for use with the Johnson-Cook (1983) constitutive model. White (1992) recently published a report in which the limitations on use of elementary analysis for interpretation of torsion test results were assessed by comparison with finite element calculations. It was found necessary to apply a correction factor to the rotation of the grips to allow for the deformation which occurs in the shoulder section of the Lindholm-type specimen. Unfortunately, this factor is a function of the specimen geometry and the flow stress function. Also, finite element calculations have revealed a tendency for tubes to decrease in diameter as the twist increases. When this is inhibited by the massive shoulder regions of the Lindholm specimen, longitudinal bending develops. Another concern with torsion testing of thin-walled tubes is the possibility of torsional buckling. To mitigate these problems, the gauge length of the Lindholm specimen is made quite short, making accurate optical measurements of strains almost impossible. Perhaps a more serious drawback is that there is essentially no portion of the gauge section which is in a homogeneous stress state.

In an effort to circumvent at least some of the problems cited previously, the author has studied designs of torsion specimens in which a longer gauge section can be employed but which would have to

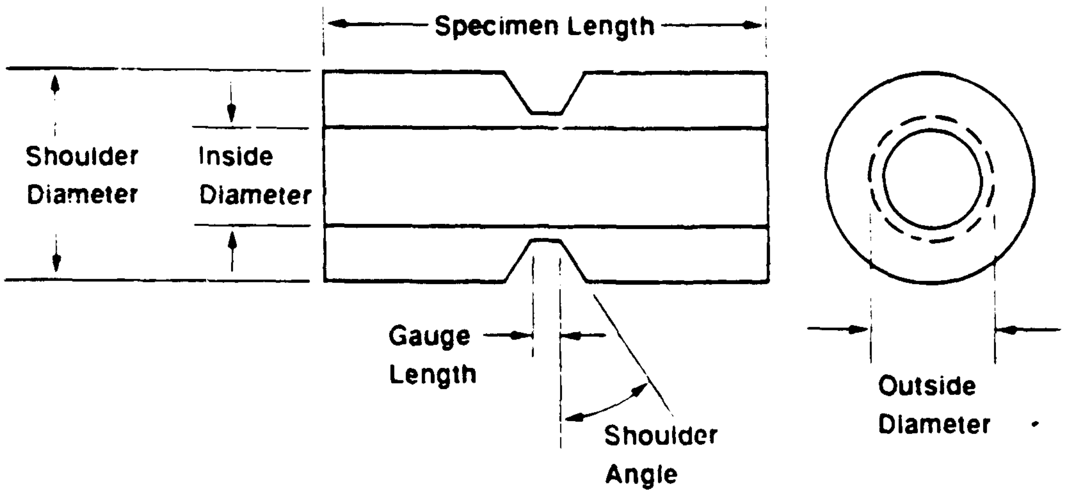

Figure 1. Geometry of the thin-walled torsion specimen.

be thick-walled or even solid to avoid buckling. There is a severe penalty associated with this approach. Whereas for the thin-walled tube, a mean shearing stress can be related to the applied torque by equilibrium considerations, it now becomes necessary to calculate the elastoplastic variation of stresses with the radius and this requires selection of a specific plasticity model. It was decided to perform the necessary calculations using rate-independent isothermal elastoplasticity, the von Mises yield function, and the associated flow rule.

The widely employed Lagrangian hydrocode DYNA3D (Hallquist 1983) provides these features in several of its material models. In particular, Model 10 accepts input of discrete data pairs representing points on an effective stress vs. effective plastic strain curve and interpolates for intermediate values as needed. This model originally only provided for isotropic work hardening, but the author has modified it to feature mixed isotropic/kinematic hardening as suggested by Hodge (1957). Also, the DYNA3D code has been altered to offer a choice between use of the Jaumann (1905) stress rate or the Green-Naghdi (1965) (Green and McInnis 1967) rate (polar decomposition of the deformation gradient). In the following, this code is employed to treat several boundary value problems pertaining to the torsion of hollow tubes and solid rods.

2. HOLLOW CYLINDERS

Consider the problem of a moderately thick ring composed of "brick" elements: 5 elements in the radial direction, 72 in the circumferential direction, and 1 in the axial direction (see Figure 2). The

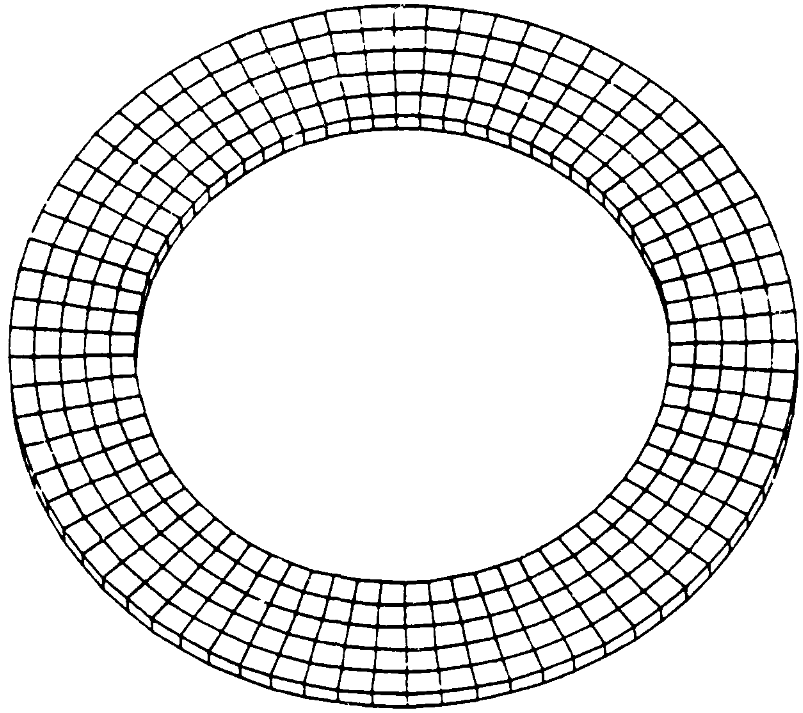

Figure 2. Ring-torsion problem gridding.

undeformed inner and outer radii of the ring are 0.315 in and 0.465 in, respectively, and the axial dimension is 0.030 in. The radial dimensions of each element are initially equal. The material data to be employed in Model 10 were derived from the quasi-static tests on annealed OFHC copper reported by Weerasooriya and Swanson (1991), 16 points on the effective stress vs. effective plastic strain curve being used as input. The density was taken to be 0.000837 lb s/in^4. The nodes are constrained to not move in the axial direction but are free to move radially. The two z = constant faces rotate in contrary directions at 1 rad/s and are given appropriate initial velocities to avoid a starting transient. Clearly, the solution of this idealized problem also applies to an infinitely long cylinder made of many such rings all subjected to the same loading. It also applies to the central portion of a finite fixed-ended cylinder sufficiently removed from the ends where torques are applied that a homogeneous state of stress exists. Except when it is desired to analyze the possibility of torsional buckling of the cylinder, it is possible to focus on the stresses and deformation of a single "wedge" of five radial elements, since all such wedges have the same deformation history (see Figure 3). Since the DYNA3D code does not have input options suitable for modeling the wedge problem, a special subroutine, T5RFIX, was introduced to apply the appropriate nodal constraints to duplicate the results of the ring calculations. Consequently, the rather voluminous results for the ring problem will not be shown but were used to check the validity of the wedge constraints.

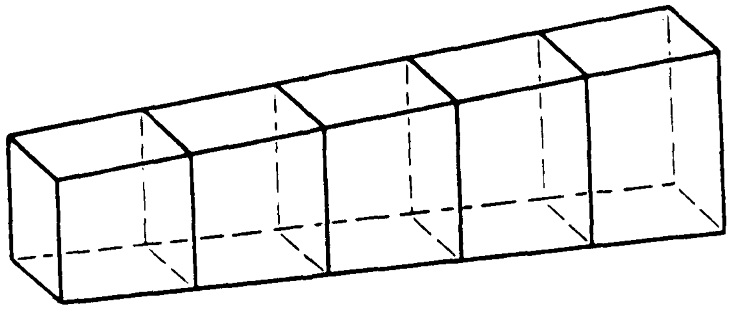

Figure 3. Wedge of five radial elements.

3. WEDGE PROBLEMS

A matrix of fixed-ended wedge problems was then studied for the possible combinations of isotropic and kinematic hardening and the Jaumann and Green-Naghdi stress rates, all run to a final torsional shear strain of $\varepsilon_{z\theta} \approx 2.0$ (tensor component). In the course of a convergence study, it was found that the major stress $\sigma_{z\theta}$ is insensitive to the size of the wedge angle, but that computed values of the circumferential stresses $\sigma_{\theta\theta}$ in the five elements were inconsistent with the requirement that the hoop force on any radial section should be zero in a statics problem. This cast doubt on the validity of all predicted normal stresses induced by the torsional loading. The difficulty appears to be associated with the brick element employed by DYNA3D. This element uses a single integration point located at its center; when the element experiences large shearing and warping, the stresses computed at the integration point are inappropriate for evaluating nodal forces since the actual stresses in the neighborhood of the nodes would vary significantly from those at the center of the element. This difficulty can be somewhat alleviated by reducing the thickness of the elements in the z-direction (which reduces the amount of circumferential stretch required to reach the desired shearing strain). Some effort was made to optimize the element thickness to minimize the hoop force and the results which follow are based on this concept.

Results from DYNA3D calculations for the fixed-ended wedge using the Jaumann stress rate for both isotropic and kinematic hardening are shown in Figure 4 for the middle element of the wedge. The isotropic hardening curve for the shearing stress is in good agreement with experimental data (Weerasooriya and Swanson 1991) and the induced normal stresses, while not zero, are too small to be visible with the scale employed. For the pure kinematic hardening case, the shear stress exhibits the widely noted sinusoidal behavior associated with the Jaumann rate, as do the induced normal stresses.

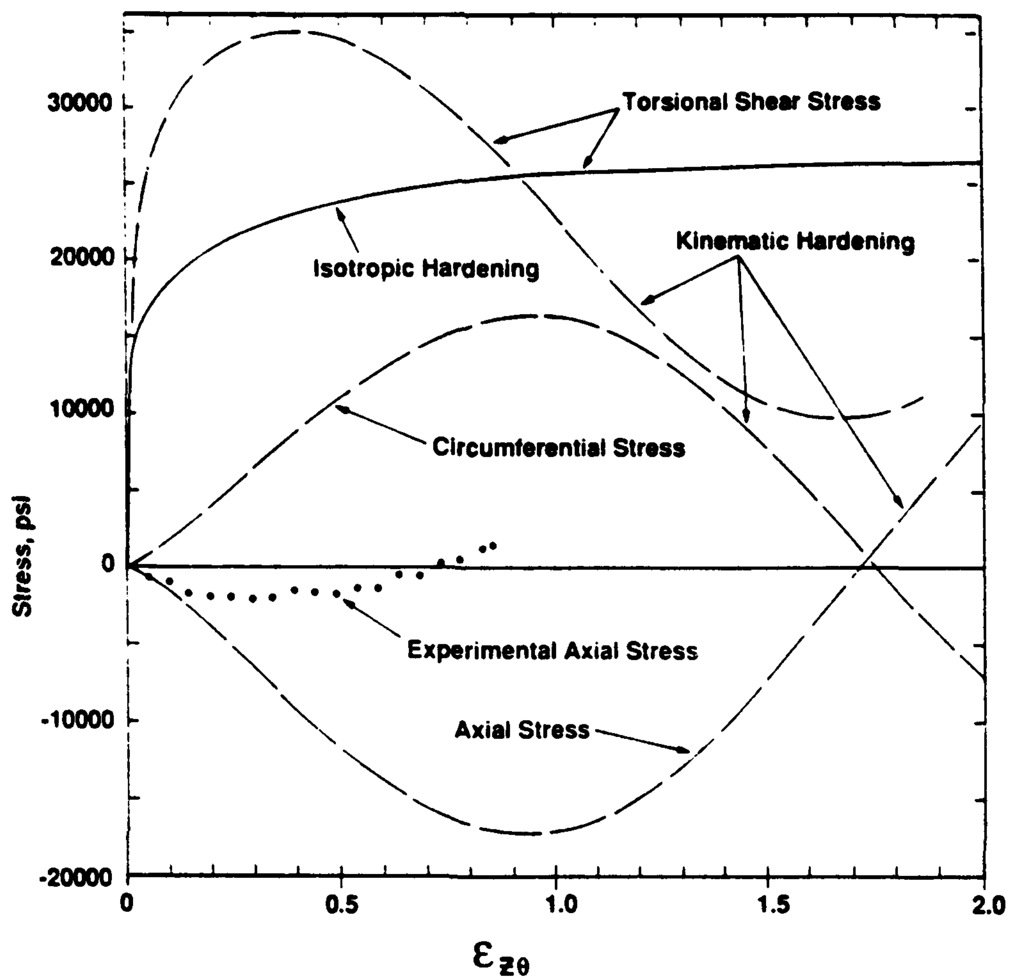

Figure 4. Stresses computed using the Jaumann stress rate.

The magnitudes of the latter stresses are unrealistically large and these stresses would significantly affect the effective stress function if actually present. The experimental curve for the induced axial stress is also shown in this figure. Calculations for a free-ended wedge were also made using the Jaumann stress rate for the isotropic case; the results were indistinguishable from the isotropic curves shown in Figure 4. Of course, there was an axial extension of the wedge and the magnitudes of the axial stresses were further reduced.

Calculations similar to those described previously were also performed using the Green-Naghdi stress rate and the results for a fixed-ended wedge are shown in Figure 5. For the isotropic case, the curves shown in this figure are essentially the same as those obtained using the Jaumann rate. In the kinematic hardening case, the early oscillatory behavior was avoided but the magnitudes of the induced normal stresses are still large.

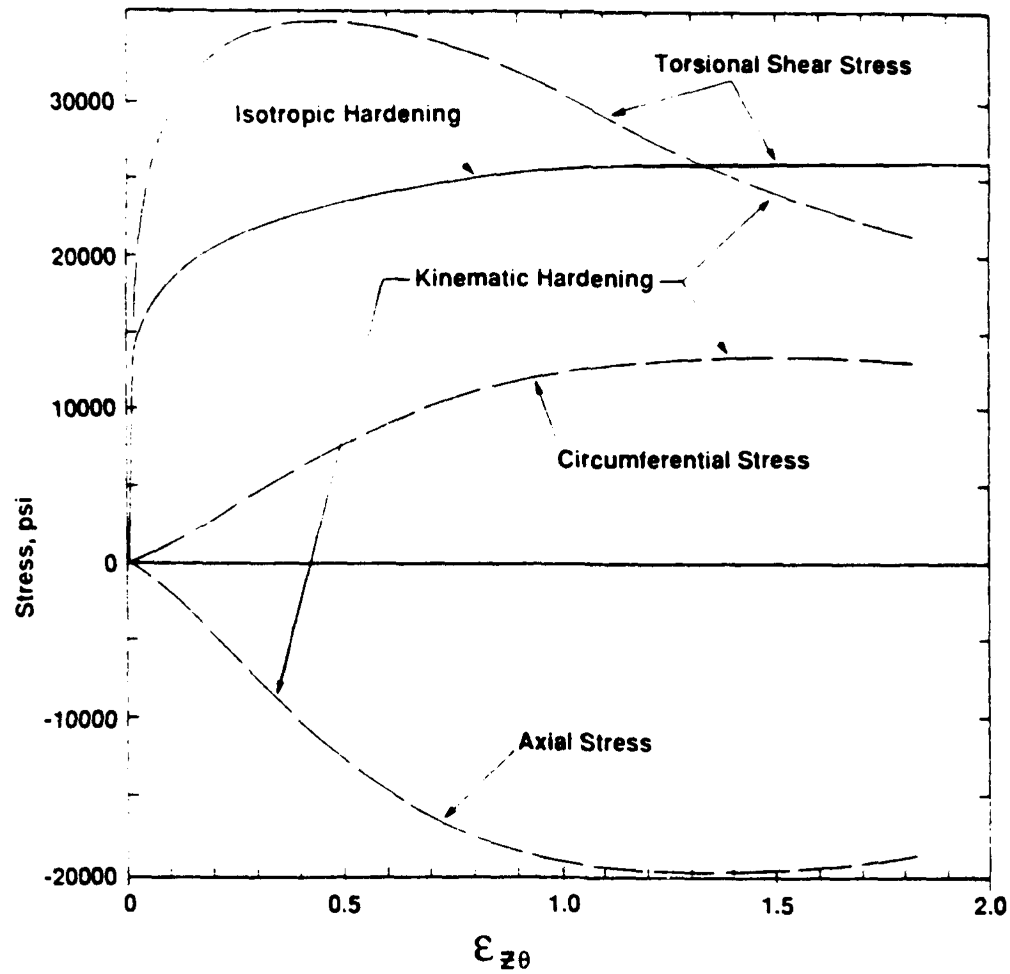

Figure 5. Stresses computed using the Green-Naghdi stress rate.

4. SOLID SPECIMENS

In anticipation that torsional buckling of hollow, cylindrical specimens might preclude successful material characterization tests at large shear strains, a study of the feasibility of using DYNA3D calculations for test data interpretations (up to incipient buckling) was conducted. Again, it is not necessary to model the entire cross section, but only a "pie-shaped" wedge with appropriate constraints. To accomplish this, the DYNA3D code was modified to include subroutines TWED and TWED2, which apply to the geometry indicated in Figure 6.

A series of calculations were performed in which the nodes on the outer surface in the "grip" region were inhibited from moving in the axial direction and constrained to rotate about the Z-axis at specified angular velocities. The results of these calculations are too complex to cover in this report. However, it is worth noting certain new phenomena which arise in these calculations.

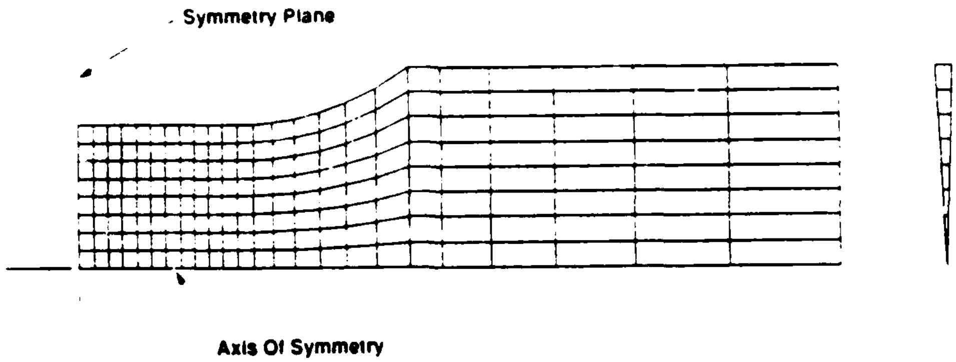

Figure 6. Geometry for the solid wedge calculations.

One of these is what may be termed "isothermal shear banding," which entails a spontaneous, rapid increase in plastic strain in an element or in all the elements at some axial location. This phenomenon is unrelated to thermal softening of materials since the mathematical model has no provision for thermal effects. Although this behavior is observed to a very limited extent during calculations using isotropic hardening, it is a serious destabilizing effect when kinematic hardening is employed. This banding is triggered in the most critically loaded element when the sinusoidally varying shear stress decreases from its first peak. Figure 7 shows end views of the twisting wedge before and after the appearance of the first band. Unlike adiabatic shear bands which progress to extreme localization, these isothermal bands tend to broaden as the banding spreads to adjacent elements.

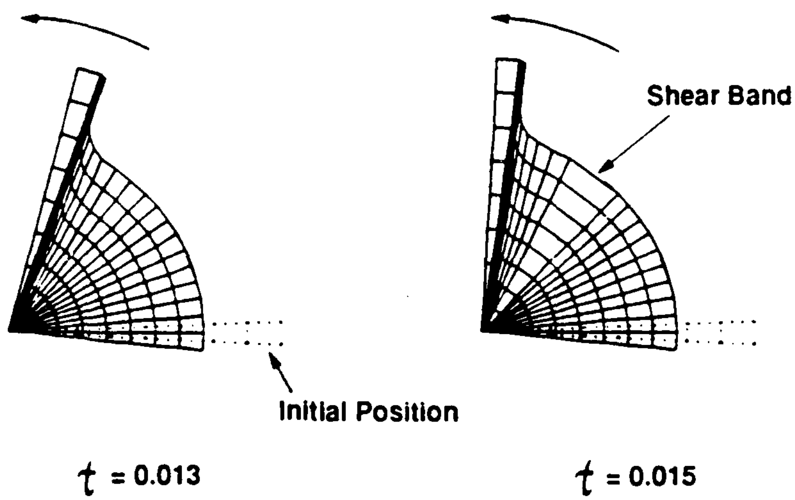

Figure 7. End views of wedge before and after shear banding.

Another phenomenon occurred during a mixed isotropic/kinematic calculation using the Green-Naghdi stress rate (made to assess the Bauschinger effect) in which the "grip" end was programmed to twist through 280° and then twist back to 120°. During the latter part of the reversed loading, the wedge was observed to buckle (computationally, but this may also occur in a physical experiment).

5. CONCLUDING REMARKS

It is the uncertainties regarding modeling plastic flow, work hardening, evolution of anisotropy, and objective stress rates which impede successful finite element modeling of experimental specimen configurations and motivate experimentalists to adopt simple shapes such as the thin-walled tube for which stress can be related to strain through equilibrium and geometric considerations.

The feasibility of modeling the torsion of hollow cylinder and solid rod specimens has been demonstrated in this report, but the results are conditioned by material modeling decisions. In view of this, the author does not feel that the tedious and expensive calculations required for a converged solution for the solid rod can be justified. Further study of modeling the hollow tube using various material representations and alternate finite elements may be worthwhile.

Where it is desired to use the thin-walled tube specimen, the configuration shown in Figure 8 may be considered. This configuration, which is very similar to that employed by Professor Swift (1947), consists of a straight, cylindrical tube with snugly fitted plugs of a high modulus material inserted in each end. The grips of a torsion tester would be applied in the region of the plugs. The gauge section of the tube must be relatively short to inhibit torsional buckling. Swift attempted to resist buckling by introducing a small clearance solid rod into the gauge section as part of one of the end plugs but had problems with binding between the rod and specimen. It would appear preferable to introduce a "free floating" solid rod and use today's super lubricants. Another method for delaying the onset of buckling would be to apply a uniform axial tension to the test specimen.

It should be remarked that elastoplastic parameters obtained by finite shearing or compression tests may no longer pertain to an isotropic material. It would be extremely valuable to be able to map the current yield surface to assess induced anisotropy, preferably in the same experimental apparatus.

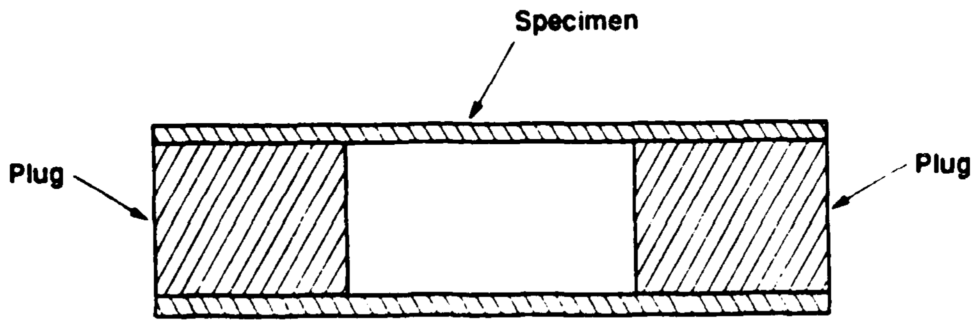

Figure 8. Suggested torsion test configuration.

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