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DESIGN MANUAL FOR POTTING
ELECTRONIC ASSEMBLIES

Prepared Under PDO 6984462

Published September 1976

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MASTER

**Kansas City
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FOREWORD

This manual is intended for practicing mechanical designers. Hence, definition of engineering terms is minimized. Unless otherwise stated, the usual engineering definition of each term is intended.

This manual will give a sound background of design practice for screening potting compounds, and quick overall design methods, basic formulations for detailed design, and design features both good and bad, so that the designer can sort and incorporate good design features at a preliminary stage. In addition, use of the manual should delete bad features from existing designs as life-test failures and service failures occur.

Each section presents a state-of-the-art overview of pertinent subject matter. Each section is expected to be changed as technology advances, as corrections are required, or as useful additions become available. These additions will be included in each section as the need arises. Possible titles for several of these additions are italicized in the table of contents. Each completed section is identified by original author or later editor. Each reference within the text refers to a figure or table within the section unless otherwise noted.

The numbering scheme includes reference to the part, the chapter within that part, and the section within that chapter. For example, 2.3.2 is in the second part, the third chapter, and is the second section in that chapter. Pages, figures and tables are numbered using a similar scheme. For example Figure 3.4-6 is in the third part, the fourth chapter of that part, and is the sixth figure of that chapter.

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1. DESIGN PHILOSOPHY

1.1 Design Philosophy Concepts (F. W. LaMar and W. J. Stone)

Only an expert in the design details of electronic components and in their packaging can successfully design potted electronic modules in their totality and within an acceptable time frame. The intent of this manual is not to create such design experts. The intent is to present the philosophy, the techniques, and the material parameters necessary to make good engineering judgments during the production, potting, analysis and minor modification of a given design. It was the original intent that these recommended judgment techniques be restricted to those that could be used by the casual processing engineer with a slide rule and in the matter of a few hours; some short computer programs are now included as a modern convenience.

This part of the manual on design philosophy is of great importance. A design engineer exercises his design philosophy when he selects materials, components, and design formulas and uses the resultants to decide whether his design is good enough. The work involved in and around these decision-making processes can be performed by nondesign engineering personnel with a minimum of direction from the design engineer if they understand his design philosophy.

Design philosophy, as defined for this manual, is that collection of design laws, knowledge, and experience that governs the field of potted electronic assemblies. The design laws and design knowledge are largely proprietary in this field and are hard to acquire from available literature. The engineering judgment necessary to successfully combine them, however, is the hardest design philosophy factor to acquire, and it can be obtained usually only from experience. Fortunately, a considerable amount of basic philosophy can be transferred from other fields of engineering.

Most design engineers differ somewhat in their design philosophies, but successful ones agree on basics. No doubt the accumulation of basic design philosophy would benefit if detailed investigations could be made to correct processing and design failure problems as they arise. Most often, production schedules cause manufacturing engineers to avoid such investigations and their detailed reports. There is a tendency to use preconceived notions and surface judgments to select a design fix. This is the main tool of the production engineer, and he successfully solves many particular problems in this way. It is a tribute to him that only in recent years have increased service requirements on the product demanded more sophisticated techniques for the general solution of the potted components problem. It is this lack of public domain design philosophy and increased demand, which has increased the effort

in this field. One major objective of this manual is to aid in the accumulation of this potting design philosophy.

In generalized approaches to the design of potted electronic assemblies, it is dangerous to forget that the assembly is most often a collection of closely grouped but greatly different individual components. Such "standard commercial items" are of highly developed design and do meet rigorous specifications to assure a universality of use. Commercial components offer availability, long functional histories, proven operational parameters, and mass production economies. However, the reason for a particular design detail may be buried in the past and for a particular potting may be a liability. Cost reduction, especially for quantity commercial production, frequently has introduced catastrophic features into designs that are required to resist mechanical loads. Hence, a processing engineer can be delivered an electronic assembly of failure prone commercial components for potting to resist thermal and mechanical loadings with the admonition that the encapsulation design should not affect the electrical characteristics.

Normally only a small percentage of potted electronic assemblies fail from poor component selection at the present day level of design experience. To adequately solve these design failure problems, the designer must become intimate with the design details of the offending component. With this detailed knowledge he can apply his general laws and general knowledge to solve the particular design failure without creating a whole new set of problems.

In recognition of this essential blend of general and detailed design philosophies, the reader will find two parallel, but not exclusive, paths pursued in this manual. One path will develop one or more general solutions, and the other will explore particular effects of these solutions on specific components and the detailed design of items, such as transformers, collector leads, and resistor cores.

The two paths require the creation of certain design tools and a standard sequence of events as the design engineer applies his philosophy. He will:

- Determine component requirements,
- Define the functional environment of the components,
- Delineate the significant design features,
- Survey and select design formulations,
- Develop formulation interactions,

- Design the component model,
- Fabricate a model,
- Test the model,
- Redesign as a product,
- Place the component module in production,
- Ascertain service problems, and
- Redesign for service problems.

His working tools for this sequence at least implicitly include the following

- The Component Model.
- The Component Module Model.
- The Design Model.
- The Math Model.
- The Experimental Model.

The following definitions will show that these need not be mutually exclusive concepts and that a complete set of such tools may be required for each entity singled out for analysis. One complete set of models may be required for the assembly, one complete set for each component or sub group of components, and perhaps a complete set of models for each of several parts of the individual component. That is, a world within a world within a world to the degree of resolution required by the individual design engineer.

A Model is any useful representation of the subject of an investigation. It should contain the substance of the entity to be studied. It should contain only those attributes that are significant to the study.

A Component Model is a useful representation (sometimes only a simple narrative description) of the typical, nominal, average, or normal member of the statistical class of the entity we chose to call a component; generally, this might be a resistor, transformer, diode, or perhaps a lead wire of any of the above.

A Component Module Model is a useful representation of the minimum-sized assembly of the subject entity and its significant environment. In this work it is often the smallest assembly of potting

compound, conformal coating, and embedded components needed to study the interactions of external thermal-mechanical loads on a problem electronic component. It is sometimes called a "pilot model."

A Design Model is the specific design philosophy for an entity to be designed. Aside from the collection of significant design formulae, material and environmental requirements, functional objectives, safety factors, and so forth, it is the simplest adequate specification of what is to be accomplished. It is the total concept that will be used to judge the success of the design effort.

A Math Model is a mathematical representation of the entity. Once an adequate working math model of a concept is established, it is the cheapest, fastest, most reliable tool for probing the subject for information. These can be "slide rule" simple or "computer" complex.

An Experimental Model is a physical representation of the subject capable of manipulation, observation, and measurement while performing the function under investigation. If it conforms to the well-known laws of size and similitude, then it is a "true model", and experimental data from it applies validly to the subject under investigation. If it varies in a known manner in either size or similitude relations, then it is known as a "distorted model." The latter is the normal case, and the necessary correction must be made to the raw data from distorted models.

It is important to recognize that the total design model of a potted electronic component assembly contains all of the above models and that they interact to enforce, correct, modify, and illuminate each other in the manipulation of the design philosophy that creates the final model of the desired product, the prototype.

Here the concept of design efficiency is introduced. When the total assembly has been designed and components and materials selected, the designer must look back over his entire design and determine if it is good enough.

Four conditions may be satisfied with the "design efficiency" calculation.

- The material selected must withstand its environment (internal stresses and external loads).
- The components must withstand the potting material (potting compound stresses).

- The component assembly being potted or repotted must last through its required service life.
- The margin for error (safety factors) must be anticipated.

Design efficiency gives the designer an approximate level of confidence in his work. It is a measure of how well the models that were developed approach or represent the design requirements.

A design efficiency for a particular math model is defined

$$\text{Design Efficiency (Percent)} = \frac{\text{Math Model Stress}}{\text{Design Model Stress}} \times 100 \text{ Percent}$$

That is the ratio of the maximum stresses from that math model to the maximum stresses from a given design model. Note: There may be only one total assembly design model, but if necessary, a particular component can have its own design model and hence its design efficiency with respect to any of its complete set of models.

An example is shown in Figure 1.1-1. If we assumed that the math model consisted only of the radial pressure between the ceramic core and the epoxy covering of a resistor, a design efficiency could be established from Figure 1.1-1. A similar calculation comparing the experimental model to the same design model is a measure of the designer's success.

A design efficiency, calculated at any stage up to and including the prototype, can caution a designer that his design is in danger or can reassure him that his design is satisfactory. For example, a design efficiency (for a component not subject to fatigue) of 30 percent might be considered satisfactory and a very simple generalized math model might be satisfactory. This math model of efficiency of 30 percent might be based on ignoring stress concentration. However, if buckling is a factor, such as in a transistor, a design efficiency of 70 percent might be required by the designer and a different general math model will be used (one that contained the concept of buckling).

In the following treatment of philosophy of material properties, one relatively easy way of designing with each property will be discussed, not necessarily the best one for each property but it will illustrate the problems involved and how they can be attacked. In future work, less general and more refined approaches may be substituted if required.

It should be noted that the later chapters on design will use the different philosophies discussed to provide modified materials and design formulations. It is expected that designers who become deeply involved in screening of potting compounds will add others as required.

Figure 1.1-1. Design Efficiency

1.2 Philosophy of Materials (F. W. LaMar, W. J. Stone, and G. D. Swanson)

All models attempt to describe the functional behavior of an entity by using performance curves that describe behavior under stresses applied during controlled test conditions. Because it is impractical to use the actual performance curves, certain values or properties are taken mathematically from the curves for use in the formulae of the model. This chapter discusses the proper ways to use some of these properties and techniques for adjusting them for situations where the standard book values are inadequate.

Stress-strain curves have been used in the past to create some of the constants and variables which engineers call properties of materials. These properties are so commonplace to designers that the concepts actually represented are not often consciously perceived. Worse, the so called constants may be obtained from data having large inherent statistical scatter such as in ultimate strength or percent elongation. Some depend on specimen history and are not constant (such as elastic limit or yield strength); and some are derived only for a specific group of materials (such as elastic modulus for Hookean solids). When engineers began to modify these concepts to fit new materials, the modified properties were given special names, i. e., flexural modulus or secant modulus.

Now that polymers are used to carry low loads and composites to carry high loads, more precise design practices are needed. Designers can no longer take refuge in large safety factors. Modern metals and polymers require higher design efficiency because of more severe service conditions.

Design formulas are usually based on either allowable loads or deflections. Many formulas for loads pose no problems today. However, there is a tendency to compromise difficult designs by allowing some plastic flow in metals and using polymers for load carrying or absorbing structures. These require more accurate calculations of total deflection. Unfortunately, many engineers, when attempting to make accurate deflection calculations, continue to use, even on polymers, the usual engineering material properties.

This chapter will help guide the designer in a judicious use of engineering material properties for making good designs and for assisting in evaluating calculated answers.

1.2.1 The Stress-Strain Curve

A stress-strain curve is the relationship between unit load and unit deflection for a given material. Many engineers use factors derived from these curves (such as elastic modulus or yield

strength) but forget the basic concepts. As an example, consider a stress-strain curve for a typical low carbon mild steel (Figure 1.2-1). These steels work well for static loads, low rpm machinery, and pre-date modern mechanical stress analysis. Because deflection in such steels becomes large immediately after the metal starts to yield, the yield point was taken as being the maximum limit for load. When deflections became important (for such things as beams) the elastic modulus was used to convert calculated deflections into loads for comparison with the yield point.

Such calculations worked well until high rpm machinery required high strength, low-alloy steels. Because these steels had no obvious yield point, a new property, the yield strength, was defined as a specified offset of strain. The yield point of mild steel occurred at approximately 0.2 percent offset; therefore, 0.2 percent offset was initially used for the yield strength of alloy steels. Values ranging as low as 0.01 percent have also been used. When it was discovered that the gage length was important, a two-inch (50.8 mm) length was used unless stated otherwise.

Low modulus materials that carry loads have deflections that are difficult to calculate accurately. The elastic modulus which normally appears in most one, two, and three dimensional formulas is assumed to be that which is measured on uniaxially loaded tensile test specimens. The bulk modulus also appears occasionally but it too is often related to a uniaxial test. A comparison of Figures 1.2-1, 1.2-2, 1.2-3, and 1.2-4 shows the cause of some trouble. Only the first 1 percent of the stress-strain curve in Figure 1.2-1 is used for normal design work. A safety factor of 2 reduces the 1 percent to 0.5 percent and probably results in a stress below the elastic limit (point X, Figure 1.2-2 for the mild steel and point Y for the alloy steel). Both points are below the proportional limit and fall on the elastic modulus curve giving accurate strain, deflection, and stress calculations.

Figure 1.2-3 shows that a similar philosophy for glass microballoon filled epoxy will result in a calculated stress (point Z) on the elastic modulus (E) curve but *above* the stress-strain curve. The 200 psi or 10 percent difference is rather small, but the 2200 psi load is also small. Mechanical designers are unwilling to settle for loads of 1000 to 2000 psi (6.8 to 13.7 MPa), even in unfilled polymers for use in potting compounds. It is common to see overall stress levels of 5000 to 6000 psi (34.4 to 41.0 MPa) in polymers and 1000 to 2000 psi even in elastomers. Figure 1.2-3 shows that a 0.2 percent yield strength of 4600 psi (37.9 MPa) (point u) will appear as a stress of 5500 psi (point t) if an elastic modulus of 548,000 psi is used. This is approximately a 20 percent error. Figure 1.2-4 shows how a deflection calculation for an elastomer, which gives a result of 800 psi (5.5 MPa), would be an error of +60 percent on stress-level (point R versus S).

Figure 1.2-1. Mild Steel Tensile Stress

Figure 1.2-2. Detail in Figure 1.2-1
Magnified 20X

Figure 1.2-3. Glass Microballoon Filled Epoxy

Figure 1.2-4. EN-4 Elastomer Compression

Point Q in Figure 1.2-3 shows that the results of elastic stress calculations, when deflection formulations are not used, can be even worse because the calculated stress can be high enough to miss the stress-strain curve altogether.

Obviously, some modification of philosophy is needed to get back to a practical situation. The following subsections show how to modify various engineering material properties to overcome such a problem. First, however, a general philosophical approach will be developed.

All engineering design methods and formulations (finite differences, fracture toughness theory, and the like) require the stress-strain curve. This basic dependency will be continued with appropriate modifications for potted components. Hence the basic stress-strain curve for the material under consideration will be examined closely and its modified engineering properties determined. This can be discouraging after looking at a family of real stress-strain curves for some materials. Figure 1.2-5 shows that stress-strain curves of individual specimens vary. The solution, of course, is the use of nominal curves and confidence bands where the statistics of the data requires them.

It has been said that engineers use imprecise data to make precise calculations from which imprecise engineering judgments are rendered. Nevertheless, engineers are still correct in using such data because good materials testing will normally yield only a small spread of data points. Engineering prediction usually does very well with such data.

When a critical situation arises where the data spread is too large (for example, at high temperature) and close engineering prediction is required, minimum and maximum stress-strain curves (Figure 1.2-5) can be assumed in several trial calculations to determine their effect on the design. However, it is a mistake to attempt to assume a conservative value. So many physical interactions occur that it is not practical to guess which direction is conservative.

1.2.2 Apparent Elastic Modulus

A good solution to the elastic modulus problem is the use of a curve fit equation for the stress-strain curve. Most existing design formulas were derived using a constant value E for elastic modulus. A solution to the problem of assuming a constant elastic modulus is shown in Figure 1.2-6. A deflection calculation yields Point 1 in Figure 1.2-6. If the apparent modulus shown had been used, the stress calculated would fall on the actual stress-strain curve at Point 2. The apparent modulus E is not known but can be found by trial and error.

Figure 1.2-5. Modulus of Elasticity GMB/Epoxy

Figure 1.2-6. Calculating an Apparent Modulus

For example, after a stress calculation shows a stress at Point 1, the following procedures should be followed.

- A lower stress level (Point 3) about a third to a half of the distance between Points 1 and 2 should be selected.
- The strain (ϵ_4) for that stress level (σ_2) on the stress-strain curve (Point 4) should be determined.
- An apparent elastic modulus for Point 4 should be determined.
- The stress using the original formula and the apparent modulus calculated in the above step should be calculated.
- The new stress level should be plotted on the stress-strain curve (Point 3 in Figure 1.2-7) using the apparent modulus E .
- The above steps should be repeated until the calculated stress level coincides with the stress-strain curve (Point 5 of Figure 1.2-7).

This process usually needs only three to five trials by hand and also works well on a computer when care is taken to assure it converges to the stress-strain curve. Of course, the correct elastic and apparent moduli must be used because of the thermal changes involved. Use of the room temperature elastic modulus at -65°F (-54°C) or $+165^{\circ}\text{F}$ (74°C) can cause a large error.

Figure 1.2-7 Convergence
of an Apparent
Modulus

1.2.3 Time-Modified Modulus

An apparent elastic modulus may appear to remove all difficulties in calculating deflection. However, another large error in stress analysis will occur if thermal cycling or time under load is involved. Visco-elastic effects in polymers under significant stresses¹ can cause errors as large as 300 percent.² Essentially, visco-elasticity involves an increase in strain under constant static load at any temperature. It is much like creep in metals at high temperature² or stress relaxation in bolts stressed above the yield point. Generally, the following design approach should be used only when components are under a static load for more than one hour.² While most discussion here is based on linear visco-elasticity theory, the designer can solve some problems involving the more complex nonlinear phenomena.

The reader is assumed to have some knowledge of creep, stress relaxation, stress rupture, and the like. Figures 1.2-8 and 1.2-9 show the visco-elastic effects from various types of loading. These are creep, creep recovery, stress relaxation, and relaxation recovery.

Once the designer has examined the component and has determined which of the four reactions is occurring, the stress (or strain) formulation can be corrected by modification of the elastic modulus. The correction is time dependent and thus called a time-modified modulus.

Here is an illustration for the designer that shows how these corrections can be developed. A time-modified modulus for the hoop stress on the inside diameter of the epoxy covering of a typical resistor will be used as an example. The hoop stress for a potted resistor is calculated for a large temperature drop from room temperature. The large thermal contraction of the epoxy cover is resisted by the miniscule shrinkage of the ceramic core, and a large tensile hoop stress results (O to A, Figure 1.2-9). The inside diameter of the cover is held nearly constant, causing stress relaxation (A to B, Figure 1.2-9). This stress change must be added algebraically to the thermal hoop stress when the resistor is kept at a low temperature. When the unit strain is constant the stress is still proportional to the modulus.^{1,2}

$$E = \frac{\sigma}{\sigma_0} E_0 \quad (1)$$

σ = stress at any time t ;

σ_0 = initially applied stress;

E_0 = initial elastic modulus; and

E = time-modified modulus at time t

Figure 1.2-8. Creep Behavior

Figure 1.2-9. Relaxation Behavior

A time-modified modulus can also be used for creep.

Sometimes it will be more convenient to use a creep-strain versus time curve, in which case²

$$E = \frac{\sigma}{\epsilon_o + \epsilon_t}$$

ϵ_o = initially applied strain, and

ϵ_t = creep (visco elastic) strain at time t .

Caution must be exercised since, when using any time-modified modulus, the time-modified Poisson's ratio of section 1.2.6 should also be used.

1.2.4 Bulk Modulus

Most engineers are rarely involved with the bulk modulus of materials (also called modulus of volume expansion or compressibility modulus). Textbooks refer to it only briefly. Because the design of potted electronic components involves trapped materials, engineers will be greatly concerned with the bulk modulus. Design use of the bulk modulus must be sophisticated or large errors might arise. This discussion of design philosophy is intended to make the designer aware of what the bulk modulus actually is and when it should be used.

An engineering background and a rudimentary concept of the bulk modulus is assumed. The bulk modulus concept is required when engineers attempt to use traditional analytical methods in design situations where the material is constrained by exterior forces in such a way that the lateral Poisson's contraction is prevented from occurring (Case C, Figure 1.2-10). While the bulk modulus does not directly apply in Case B (Figure 1.2-10) several things can be done to allow for its usage (later discussion on Figure 1.2-15). The bulk modulus concept was developed for use by designers for triaxial stress analyses⁵ where the three principal stresses are equal and no shear stresses exist.³

The concept can be extended to other situations provided some limitations are not exceeded. Figure 1.2-11 show the behavior of a cube subjected to a uni-axial load when there is no side constraint.⁴ Theoretically all strains are considered constant across each face of the cube, and Poisson's ratio, μ , is assumed to be isotropic. There are two Poisson's ratio effects because there are four cube faces parallel to the load axis.

Figure 1.2-10. Application of Bulk Modulus

Figure 1.2-11. Axial Load Effect on Cube

Many design formulas contain terms such as $1-\mu$, $1-2\mu$, and $1-\mu^2$. In such cases the designer should be aware that biaxial or tri-axial stresses are being considered by the stress analysis. When engineering accuracy (5 percent or better) is of interest, it should be remembered that these terms compensate for the restraining effect of Poisson's ratio.

A rectangular parallelepiped having arbitrary normal stresses applied to the surfaces illustrates the extension of the bulk modulus concept (Figure 1.2-12). By the classical theory of linear elasticity applied to homogeneous, isotropic solids, the strains in any direction can be found by linear superposition of the strains resulting from each individual stress (σ) (Table 1.2-1). Poisson's ratio is μ and Young's elastic modulus is E .

By applying linear superposition, the material is shown to have strains (ϵ) in the various directions.

$$-\epsilon_x = \frac{1}{E} [\sigma_x - \mu(\sigma_y + \sigma_z)] \quad (3)$$

$$-\epsilon_y = \frac{1}{E} [\sigma_y - \mu(\sigma_x + \sigma_z)] \quad (4)$$

Figure 1.2-12. Parallelepiped Under Three Normal Stresses

Table 1.2-1. Determining the Strain in Each Direction

Direction	Stress Applied	X Strain	Y Strain	Z Strain
X	$-\sigma_x$	$-\frac{\sigma_x}{E}$	$+\mu\frac{\sigma_x}{E}$	$+\mu\frac{\sigma_x}{E}$
Y	$-\sigma_y$	$+\mu\frac{\sigma_y}{E}$	$-\frac{\sigma_y}{E}$	$+\mu\frac{\sigma_y}{E}$
Z	$-\sigma_z$	$+\mu\frac{\sigma_z}{E}$	$+\mu\frac{\sigma_z}{E}$	$-\frac{\sigma_z}{E}$

$$-\epsilon_z = \frac{1}{E} [\sigma_z - \mu(\sigma_x + \sigma_y)] \quad (5)$$

This relation between stresses and strains is independent of the magnitudes or relative magnitudes of the stresses as long as the strains are small enough to remain in the linear elastic realm.

The bulk modulus (K) is then defined as the ratio between the mean stress (σ) and the volume strain (ϵ_v)

where

$$\epsilon_v = \epsilon_x + \epsilon_y + \epsilon_z \quad (6)$$

$$\sigma = \frac{1}{3} (\sigma_x + \sigma_y + \sigma_z) \quad (7)$$

so that

$$K = \frac{\sigma}{\epsilon_v} = \frac{E}{3(1-2\mu)} \quad (8)$$

The last equality above follows directly from the general stress-strain relations and the mean stress and volume strain definitions. The bulk modulus is hence the ratio of mean stress to volume strain even when the principal stresses are unequal.

In a pure hydrostatic loading, $\sigma_x = \sigma_y = \sigma_z = -P$ where P is the applied uniform hydrostatic pressure. The strains are then

$$\epsilon_x = \epsilon_y = \epsilon_z = \frac{-P}{E} (1-2\mu) = \epsilon, \quad \epsilon_v = 3\epsilon \quad (9)$$

and the bulk modulus is

$$K = \frac{\sigma}{\epsilon_v} = \frac{-P}{3\epsilon} = \frac{-P}{\frac{-3P}{E}(1-2\mu)} = \frac{E}{3(1-2\mu)} \quad (10)$$

as in Equation 8 above.

At no time have shear forces been involved; therefore, the above formula represents a uniform compression as in Figure 1.2-12. This is called a dilatational change and is a change of volume only.

When a shear stress is applied to a cube, it distorts as shown in Figure 1.2-13. Because the volume does not change, the resultant change is in shape only and is called a deviatoric change.

Figure 1.2-13. Shear Stresses

When normal and shear stresses are superimposed, the volume and shape will both change. This change is the sum of the dilatational normal and the deviatoric shear stress effects.

When principal stress directions are selected (x, y, and z orthogonal components) and the stresses are not equal, a cube becomes a parallelepiped as in Figure 1.2-14. Normal tension stresses (Figure 1.2-14) can be considered as a triaxial, uniform tension of 25,000 psi (172.4 MPa) with the following superimposed stress system.

Figure 1.2-14. Unequal Stresses on Cube

$$\sigma_x = 0$$

$$\sigma_y = 35,000 \text{ psi (241.3 MPa)}$$

$$\sigma_z = 10,000 \text{ psi (68.9 MPa)}$$

These additional stresses cause the additional extension in the y and z directions.

In many design situations, application of the bulk modulus does not completely solve the problem. Consider the situation which occurs during a temperature drop when the conformal coating of a resistor shrinks more than the resistor. The surface of a resistor applies a radial compressive stress to the conformal coating. The glass microballoon filled epoxy encapsulant applies a reactive radial compressive stress to the conformal coating. A stress situation similar to Figure 1.2-15 results.

Cases of this type are best solved by assuming two superimposed stress systems. The tri-axial compression portion can be treated via the bulk modulus, and the tension portion would require the elastic modulus. If a shear is also present, it should be treated using the shear modulus.

Figure 1.2-15. Stresses of Unlike Sign

1.2.5 The Elastic Modulus of a Trapped Material

To cope with the general potting problem as made less severe by the conformal coating, one must understand and appropriately apply the concept of a trapped modulus. A material is trapped when it cannot distort in a direction orthogonal to the direction of an external loading. A conformal coating trapped between a stiff component and a stiff potting material while undergoing compressive loading because of differential thermal expansions would be an example.

The object is to optimize the kinds, volumes, and locations of potting materials with respect to their interactions with the potted components. The handiest and most economical tool for this optimization is a mathematical model. From classical elasticity theory, various one, two, and three dimensional mathematical expressions have been developed, which can predict deflections in a potted entity because of external loadings or thermal gradients transmitted through multiple layers of protective coatings. Prime factors in all such valid mathematical models are the relative bulks of each material, their geometries, their relative coefficients of thermal expansion, and their load-deflection moduli. This section considers these latter moduli, that is, the relationships between load and deflection of a material.

There are more than twenty commonly used different kinds of moduli: Tangent, Secant, Off-set, Young's, Compression, Trapped, Bulk, Linear, Proportional, Elastic, Plastic, Visco-elastic, Instantaneous, Average, Effective, Thixotropic, Pseudo-plastic, Newtonian, apparent, and Pseudo, to name some of the more significant.

These many shadings of the load-deflection modulus concept come from the method of measurement, dimensional concepts, different physical conceptions, and other arbitrary interpretations of the stress-strain curve. One is better than another for a particular phenomenon. Some useful criteria follow.

Truth Versus Relevance

One method for sensing the loading of a potted component is through strategically located strain gages. If a well-executed strain gage measurement is defined as truth, then any computational model is valid only when it agrees with the strain gage.

That elastic modulus which best predicts the strain gage reading is the most appropriate and, therefore, the true modulus for that particular case. Relevance is truth by definition.

Functionality Versus Classical Concepts

Table 1.2-2 illustrates the experimentally determined moduli on Conap EN-4. While these nine cases are not exhaustive, they indicate experimental gradations in the same concept. Functionally, some of them model the various uses of EN-4 at Bendix Kansas City quite well. They vary in the relative amounts of dilational and deviatoric work allowed on the specimen during the test. A large geometrical effect variation is also apparent in one or more of these cases, and this variation is referred to by one or more of the following terms: bulk effects, surface-to-volume ratio effects, edge effects, sheet, or film effects.

For example, the flexibility of a steel measuring tape results mainly from the geometry; whereas, the behavior of an anvil made of the same material is controlled by its bulk elastic properties.

Accuracy of Available Methods

The following limitations exist in any experimental determination of a physical parameter:

- The ability to adequately define the concept,
- How well the experimental methods and specimens model the defined concept, and
- The error in the instrumentation.

Table 1.2-2. Elastic Moduli of EN-4

Case	Method
<p>Tension: Young's Modulus</p> <p>1. $E \approx 2500 \text{ psi (17.2 MPa)}$</p>
<p>Compression: Nonrestricted</p> <p>2. $E_c = 5740 \text{ psi (39.6 MPa)}$</p>
<p>Compression: Restricted</p> <p>3. $Q = 8220 \text{ psi (47.5 MPa)}$ $\dot{\epsilon} = 0.04/\text{min}$</p>
<p>4. $Q' - \text{Modulus } 11900 \text{ psi (82.9 MPa)}$ $\dot{\epsilon} = 0.04/\text{min}$</p>
<p>5. $K'' - \text{Modulus } 442,000 \text{ psi (3.0 GPa)}$ $\dot{\epsilon} = 0.03/\text{min at } 1000 \text{ psi (6.8 MPa)}$</p>
<p>6. $K - \text{Bulk Modulus } 260,000 \text{ psi}$ $(1.5 \text{ GPa}) \text{ at } 500 \text{ to } 1,000 \text{ psi}$ $(3.4 \text{ to } 6.8 \text{ MPa})$</p>

7. $K' - \text{Modulus } 330,000 \text{ psi}$
 $(1.9 \text{ GPa}) \text{ (LLL Fixture)}$
8. $E_s = \text{Trapped Modulus (Calculated from Case 6 or 7)}$

$$K = \frac{K' (1 + \mu)}{3(1 - \mu)} = 298,000 \text{ psi (1.72 GPa)}$$

$$E_s = K(3 - 6\mu) = 52,800 \text{ psi (364.2 MPa)}$$

$$\approx 10 E_c$$
9. $K_T = 31,700 \text{ psi (218.5 MPa)}$
at 45 psi, $\mu = 0.47$

A conformal coating is optimized for these reasons:

- The selection of ideal conformal coatings is approached through experience and is supported by a reasonably valid working model. Such materials as RTVs, thiokols, and foams are soft, can produce various coefficients of thermal expansion by using fillers, and are hydraulic enough to distribute point loads in low thicknesses. Because of their visco-elastic natures, they are suitable to being

trapped between the potting material and the component to be protected. When trapped, the modulus of elasticity must be measured by an experimental technique which adequately models the functional use of the material. If the functional coating is thick enough to do its job, then Cases 6 or 7 of Table 1.2-2 are probably mandatory test methods as they are primarily trapped bulk effects.

An analysis of Case 7 will serve as an example of how these concepts are applied. Consider a material constrained so that it will not expand in the x and y directions; these dimensions do not change so $\epsilon_x = \epsilon_y = 0$. The stresses on these surfaces will be zero when σ_z is zero and will increase with σ_z . The values can be found from the stress-strain relation of Equations 3 through 5.

$$\epsilon_x = 0 = \frac{1}{E} [\sigma_x - \mu(\sigma_y + \sigma_z)] + \sigma_x = \mu(\sigma_y + \sigma_z) \quad (11)$$

$$\epsilon_y = 0 = \frac{1}{E} [\sigma_y - \mu(\sigma_x + \sigma_z)] + \sigma_y = \mu(\sigma_x + \sigma_z) \quad (12)$$

Eliminate σ_y and solve the first for σ_x to get

$$\sigma_x = \frac{\mu\sigma_z}{1-\mu} \quad (13)$$

σ_y is then obtained from the second equation,

$$\sigma_y = \frac{\mu\sigma_z}{1-\mu} \quad (14)$$

and hence, $\sigma_x = \sigma_y$ as expected because $\epsilon_x = \epsilon_y$.

The volume strain now is:

$$\epsilon_v = \epsilon_x + \epsilon_y + \epsilon_z = \epsilon_z \quad (15)$$

and the mean stress is

$$\sigma = \frac{1}{3}[\sigma_x + \sigma_y + \sigma_z] = \frac{\mu + 1}{3(1 - \mu)} \sigma_z \quad (16)$$

so that the bulk modulus is

$$K = \frac{\sigma}{\epsilon_v} = \frac{(\mu + 1)}{3(1 - \mu)} \frac{\sigma_z}{\epsilon_z} \quad (17)$$

We also know the relation between σ_z and ϵ_z from the stress-strain relations is

$$\epsilon_z = \frac{(1 + \mu)(1 - 2\mu)}{E(1 - \mu)} \sigma_z \quad (18)$$

therefore, the bulk modulus simplifies to

$$K = \frac{E}{3(1 - 2\mu)} \quad (19)$$

Hence, this is consistent with the pure hydrostatic pressure case, as long as the mean stress is divided by the volume strain. However, the bulk modulus is not equal to the uniaxial stress divided by the colinear strain; that is

$$K \neq \frac{\sigma_z}{\epsilon_z}$$

The difference between K and $\frac{\sigma_z}{\epsilon_z}$ is the factor $\frac{\mu + 1}{3(1 - \mu)}$ which varies with μ as in Table 1.2-3.

Table 1.2-3. Conversion to Bulk Modulus

μ	$\frac{\mu + 1}{3(1 - \mu)}$
0	1/3
1/5	1/2
1/4	5/9
3/10	13/21
1/3	2/3
4/10	7/9
9/20	29/33
1/2	1

The result is that a trapped elastic modulus, E_s , for an elastomer used as a conformal coating can be calculated by rearrangement of Equation 19.

$$E_s = 3(1 - 2\mu) K$$

This is Case 8 in Table 1.2-2.

A graphical comparison of the use of a trapped modulus with use of the untrapped compressive modulus is shown in Figure 1.2-16.

Figure 1.2-16. Trapped Modulus/Untrapped Compressive Modulus

In Figure 1.2-16 at Stress A, there would be no difference. At Stress B, the modulus that best models the function should be chosen. If it is not trapped, E_c might be best. At Stress C the only existing relationship for usage is E_s , the trapped modulus.

Figure 1.2-17 shows the difference calculated for a Proseal 727 coated, GMB-Epoxy encapsulated, ceramic Sprytron tube when using E_s versus E_c .

Recommendations

- When a material is obviously functionally trapped in usage, the three-dimensional problems should be calculated using K and one-dimensional models using E_s .
- When calculating loads on a potted component with a linear elastic model, the pseudo bulk modulus of the component should be used.
- In the common case of a practically unconstrained epoxy or urethane foam, the ordinary compressive modulus, E_c is germane in most potting situations at Bendix Kansas City.

Figure 1.2-17. E_s Versus E_c for the Spraytron Tube

- In the case of a large deflection in the component or a large shape change during the loading, the argument should be reexamined to see if the line of reasoning is appropriate.
- In any case, a modulus is a man-made invention and should be construed for best effect.

1.2.6 Poisson's Ratio

Normally, fairly rough approximations of Poisson's ratio are validly used by designers. Some tri-axial stresses which include the terms $1 - 2\mu$ and $1 + 2\mu$ will have significant errors when designing with conformal coatings if correct values of Poisson's ratio are not used. Ratios as high as 0.40 to 0.48 occur in some elastomers.

A time-modified Poisson's ratio should normally be used² when visco-elastic stresses are calculated using other time-modified elastic moduli. It has the form

$$\mu = \frac{3K-E}{6K} \quad (20)$$

where

μ = time modified Poisson's ratio at time t .

K = bulk modulus.

E = time-modified Young's modulus at time t .

If the bulk modulus is not available, then the theoretical assumption for an isotropic material

$$K = \frac{E_o}{3(1 - 2\mu_o)} \quad (21)$$

can be substituted yielding

$$\mu = \frac{E_o - E(1 - 2\mu_o)}{2E_o} \quad (22)$$

where

E_o = initial Young's modulus, and

μ_o = initial Poisson's ratio.

It should be realized that the modulus of elasticity and Poisson's ratio usually are determined experimentally.⁸

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2. TECHNIQUES, MODELS, AND APPLICATIONS

2.1 Classical Design Techniques Applied to a Typical Encapsulated Resistor (G. D. Swanson, F. W. LaMar, and L. J. Lazarus)

This chapter attempts to aid in accumulation of general potting design philosophy. It also attempts to apply this philosophy to a particular cylindrical component. A design model is developed for one specific cylindrical resistor and exercised for one specific potting situation. Two of the generalized mathematical models to be included in Chapter 2.4 are used here for specific cases.

After working with encapsulated electronic assemblies for some time, it becomes apparent that electronic components can be grouped into a small number of classes of similar components, such as resistors and capacitors, and that alternate divisions into subclasses based on design details, such as geometric shape, are possible. This chapter considers the general class of cylindrical axial lead devices, and an axial lead resistor as a particular subclass. No electrical considerations are made here.

The formulations presented here can have two different uses. After electronic components have been screened by the math model and experimental models have been screened by several types of laboratory tests, the unsatisfactory components can be redesigned using the design methods developed here. An alternate use of these design principles is to provide sufficient formulations that can be added to the math model so it will more accurately and more completely predict the results from the experimental model. Stress concentrations, end loadings, bending moments, and other departures from concentric cylinders under a thermal shrink fit may be necessary for adequate description in some cases.

Electronic components are expected to withstand the following processing and service requirements:

- Thermal stresses from shrinkage caused by curing of the potting compound, by the cool down from curing temperature, by thermal cycling, by ambient storage temperature changes, or by the static or unstabilized thermal gradients;
- Stresses imposed by neighboring components (assembly interaction);
- Inertia body stresses from linear acceleration or rotational acceleration; and
- Sudden loads caused by impact or vibration.

The first two items impose design requirements somewhat opposite to the last two items. Components can best resist the first two items by cushioning them with low-modulus collapsible materials, while the last two items are best protected with semistiff energy absorbing materials that become progressively stiffer as deflection of the component increases. This section at present considers only the first item. The design model that follows includes only loads imposed by the first design requirement listed above.

2.1.1 Generalized Resistor Model

Resistors appear in a variety of styles. A cross-section of a popular one is shown in Figure 2.1-1. This resistor, an SA1229-18, is considered in detail below.

A ceramic core is the structural base for the resistor path, an extremely thin sputtered metallic film (to give a high resistance) with so many holes that it resembles a dense net rather than a continuous layer. A spiral groove less than 0.001 inch (25.4 μm) deep is machined into the ceramic core to lengthen the electrical path of the resistor. The length of the groove is varied to produce the required electrical resistance.

After machining, the end caps and leads are assembled, and a thin coating of silicone rubber is applied and vulcanized at 140°F (60 C). It is then covered with a thin coat of Gilsonite, a mined asphalt (essentially pure bitumen) suspended in a carrier. This should allow relative sliding between epoxy coat and core except at low temperatures. A filled epoxy covering is then brushed on. The filler is either 30 percent Al_2O_3 or 30 percent Cabosil (finely divided SiO_2). Leads are usually straight until the resistor is mounted in an assembly.

Resistors fail occasionally during assembly and in several different failure modes (Figure 2.1-2), indicating the design is marginal. The types of failures indicate that high hoop stresses, bending, and crushing might all be significant factors.

The resistor (Figure 2.1-1) can be simplified as shown in Figure 2.1-3 to represent all resistors of this style. The basic assumptions follow.

- The adequate component module model does not consider the effects of other components or PC boards.
- The Gilsonite and silicone coatings are so thin that they can be ignored.
- The conformal coating covers the epoxy cover but not the leads of the resistor.

Figure 2.1-1. Resistor Cross Section

Figure 2.1-2. Failure Modes of Resistor Core

Figure 2.1-3. Cylindrical, Axial-Lead Resistor Model

- When stresses encountered are above the proportional limit, the designer will use a modified elastic modulus or a similar approach to obtain more accurate stress levels.

- After the experimental component model resistor is received from the manufacturer, it is given a conformal coat which has practically no stress effect on the component.
- After conformal coating, the resistor is potted at the cure temperature of the potting compound and then allowed to cool slowly to the service temperature at which it is to be analyzed.

2.1.2 Design Analysis

Significant design stresses acting on the typical potted resistor are considered to be derived from one or more of the following phenomena:

- Differential contraction of materials from cure temperature to room temperature,
- Cure shrinkage of organic materials; and
- Differential expansion caused by thermal cycling between room temperature, -65°F (-54°C), and $+165^{\circ}\text{F}$ (74°C)

No thermal gradients are presently included in this analysis. However, when rapid temperature changes are undergone, thermal gradients and the resulting stresses must be considered.

2.1.3 Stress Analysis

A particular order of solution is necessary to check the design of a specific resistor in a specific package. The radial and tangential stresses come from interaction of all cylindrical layers under the anticipated maximum temperature change in use. The axial stresses also must be determined by a simultaneous solution. The appropriate stress concentration factors must be applied when adding these stresses or the resulting strains to determine the three-dimensional result. These stress concentrations are also necessary when checking for buckling, bending, and shearing and when comparing them with the appropriate material failure criteria.

The result can then be used for correcting the component design, deciding to change components, changing the packaging scheme, correcting the mathematical model, or deciding to use the current design and package. When a new component or package is chosen, the computations must be repeated; pilot production might require a further reevaluation before production continues. A flow chart of these operations is shown in Figure 2.1-4. The following resistor stress analysis uses this scheme.

Figure 2.1-4. Flow Chart of Stress Analysis

This analysis determines maximum stresses and their critical locations for the resistor style assumed in Figure 2.1-3. Proven formulations are selected and are used to determine which stresses are significant. These can be then combined into principal stresses to establish a design criterion (Table 2.1-i).

Table 2.1-1. Outline of Resistor Design Loads

Item	Applied Load	Load Source	Illustration
1	Ceramic Core Critical Buckling-Axial	GMB Epoxy
2	Critical Buckling-Circumferential	GMB Epoxy
3	Bending-Meridional Over Entire Surface	GMB Epoxy
4	Bending - Simple End	Leads
5	Stress Concentration	Grooves
6	End Crushing - Bearing	Caps
7	Thermal Stress-Tension (Axial and Radial)	Epoxy Cover

Table 2.1-1 Continued. Outline of Resistor Design Loads

Item	Applied Load	Lead Source	Illustration
8	Thermal Stress-Compression (Axial and Radial)	Caps	-----
9	Surface Shear - Compression	Epoxy Cover
10	Surface Shear - Tension	Caps Epoxy Cover
11	Bending-End Load Epoxy Cover	Leads
12	Bending - Centerline (Axial and Radial)	Caps, Leads
13	Thermal Stress-Tension-Radial	Caps, Leads
14	Thermal Stress-Tension-Axial	Caps

Table 2.1-1 Continued. Outline of Resistor Design Loads

Item	Applied Load	Load Source	Illustration
15	Thermal Stress-Compression (Axial and Radial)	GMB Epoxy
16	Stress Condontration-Voids	Air Bubbles
17	Stress Concentration-Section	Cap Corner
18	Stress Concentration-Grooves	Ceramic Core
19	Encapsulant (GMB Epoxy)	These loads are reactions and opposite of those on the epoxy cover. Calculations will be similar.	

Because only static loads are considered at the present time, differential thermal expansion will be the main source of applied loads. This effect will cause radial, tangential, and longitudinal stresses in the core, cover, conformal coating, and potting material.

These interacting stresses must be calculated simultaneously. They can be programmed for a computer when a study of the variations of parameters is desired. For a few data points, they can be calculated by hand as shown below. These are particular illustrations of the general mathematical models of chapter 2.4.

Case 1: Radial and Axial Stresses (Table 1, Items 7, 15). The differential thermal expansion assumes the cross section shown in Figure 2.1.5. The following terms need to be defined:

A, B, C, D, E = Radii to successive interfaces,
 a, b, c, d = Subscripts for material designation,
 E_a, E_b, E_c, E_d = Moduli of elasticity of successive layers,
 $\mu_a, \mu_b, \mu_c, \mu_d$ = Poisson's ratios,
 $\alpha_a, \alpha_b, \alpha_c, \alpha_d$ = Coefficients of thermal expansion,
 ΔT = Temperature change undergone, and
 $\underline{DBI}, \underline{DCI}, \underline{DDI}$ = Initial interfacial interferences.

Figure 2.1-5. Cross Section of Potted Resistor

Terms involving the material properties and geometry for each layer are used. The numerical values were calculated for an SA1229-18 resistor model with a 0.006 inch (0.15 mm) layer of EN-4 coating. All initial interferences were zero and the temperature changed from 165°F (74°C) to -65°F (-54°C). The materials properties were obtained from Chapter 3.4 and standard materials handbooks. Dimensions are from Figure 2.1-1. The following preliminary calculations are convenient and arbitrarily named. Equations 1 through 12 are expressed in inch³/pound.

$$AKO = \frac{2A^2B}{E_a(B^2 - A^2)} = 1.667 \times 10^{-10}. \quad (1)$$

$$ARO = \frac{B[(1-\mu_a)B^2 + (1+\mu_a)A^2]}{E_a(B^2 - A^2)} = 6.667 \times 10^{-10}. \quad (2)$$

$$BRI = \frac{B \left[(1 + \mu_b)C^2 + (1 - \mu_b)B^2 \right]}{E_b(C^2 - B^2)} = 1.788 \times 10^{-7} \quad (3)$$

$$BKI = \frac{2BC^2}{E_b(C^2 - B^2)} = 2.160 \times 10^{-7} \quad (4)$$

$$BKO = \frac{B}{C} (BKI) = 1.440 \times 10^{-7} \quad (5)$$

$$BRO = \frac{C \left[(1 + \mu_b)B^2 + (1 - \mu_b)C^2 \right]}{E_b(C^2 - B^2)} = 1.998 \times 10^{-7} \quad (6)$$

$$CRI = \frac{C \left[(1 + \mu_c)D^2 + (1 - \mu_c)C^2 \right]}{E_c(D^2 - C^2)} = 1.855 \times 10^{-6} \quad (7)$$

$$CKI = \frac{2CD^2}{E_c(D^2 - C^2)} = 1.973 \times 10^{-6} \quad (8)$$

$$CKO = \frac{C}{D} (CKI) = 1.741 \times 10^{-6} \quad (9)$$

$$CRO = \frac{D \left[(1 + \mu_c)C^2 + (1 - \mu_c)D^2 \right]}{E_c(D^2 - C^2)} = 1.874 \times 10^{-6} \quad (10)$$

$$DRI = \frac{D \left[(1 + \mu_d)E^2 + (1 - \mu_d)D^2 \right]}{E_d(E^2 - D^2)} = 1.101 \times 10^{-7} \quad (11)$$

$$DKI = \frac{2DE^2}{E_d(E^2 - D^2)} = 1.643 \times 10^{-7} \quad (12)$$

The initial interfacial interferences and the temperature-induced change enter through the following relations.

$$DB = DBI + B(\alpha_a - \alpha_b) \Delta T = 8.073 \times 10^{-5} \text{ inch} \quad (13)$$

$$DC = DCI + C(\alpha_b - \alpha_c) \Delta T = 8.59 \times 10^{-4} \text{ inch} \quad (14)$$

$$DD = DDI + D(\alpha_c - \alpha_d)\Delta T = 8.915 \times 10^{-4} \text{ inch} \quad (15)$$

The following intermediate terms are convenient.

$$W = \frac{[DB + (AKO)P_a]BKO + (ARO + BRI)DC}{(BRO + CRI)(ARO + BRI) - BKI(BKO)} = 490.9 \text{ psi (3.4 MPa)} \quad (16)$$

$$X = \frac{(ARO + BRI)CKI}{(BRO + CRI)(ARO + BRI) - BKI(BKO)} = 1.049. \quad (17)$$

$$Y = - \frac{DD + (DKI)P_e}{CKO} = 510.7 \text{ psi (3.5 MPa)} \quad (18)$$

$$Z = - \frac{CRO + DRI}{CKO} = -1.140. \quad (19)$$

The interfacial pressures are then found.

$$P_d = \frac{Y - W}{X + Z} = -217 \text{ psi (-1.5 MPa)}. \quad (20)$$

$$P_c = W + XP_d = 264 \text{ psi (1.8 MPa)}. \quad (21)$$

$$P_b = \frac{DB + (AKO)P_a + (BKI)P_c}{ARO + BRI} = 767 \text{ psi (5.3 MPa)}. \quad (22)$$

The hoop stress in each material adjacent to any interface can then be found from

$$\sigma_t = \frac{P_i R_i^2 - P_o R_o^2 + R_i^2 R_o^2 (P_i - P_o) \frac{1}{r^2}}{R_o^2 - R_i^2} \quad (23)$$

The pressures and radii are defined for any layer to be

P_i = internal pressure on this layer, psi, (MPa)

P_o = external pressure on this layer, psi, (MPa)

R_i = inside radius of this layer, in., (m)

R_o = outside radius of this layer in., (m) and

$r = R_i$ or R_o for the interface desired, in., (m).

This means that P_b is the exterior pressure on layer a, and also the interior pressure on layer b. Also, B is the outside radius of layer a, but the inside radius for layer b, as shown in Figure 2.1-5.

To illustrate the use of Equation 23, calculate the predicted hoop stress at the outer surface of the epoxy cover on the resistor (the strain gage location). Equation 23 can then be simplified to

$$\sigma_{tbo} = \frac{2 P_b B^2 - P_c (C^2 + B^2)}{C^2 - B^2} = 541 \text{ psi (3.7 MPa)} \quad (24)$$

All other hoop stresses can be determined in a similar manner. The results of this set of computations are shown in Table 2.1-2.

Table 2.1-2. Interfacial Stresses

Designation	Stress	
	Psi	(MPa)
P_a	15	0.103
σ_{tai}	-1710	-11.8
σ_{tao}	-958	-6.6
P_b	769	5.3
σ_{tbi}	1040	7.1
σ_{tbo}	541	3.7
P_c	265	1.8
σ_{tci}	4070	28.1
σ_{tco}	3590	24.7
P_d	-215	-1.5
σ_{tdi}	-245	-1.7
σ_{tdo}	-15.1	-0.103
P_e	15	0.103

None of these stresses is large enough alone to endanger any of the materials included. The closest are the hoop stresses in material c, the conformal coating, and they still have an engineering safety factor of about 1.5. It is not currently known if the tensile radial stress, P_d , is high enough to separate the EN-4 coating from the GMB-epoxy encapsulant at -65°F (-54°C). The problem would then reduce to a three-layer shrink fit and to a separate encapsulant cylinder under free thermal contraction; a gap would exist between EN-4 and GMB-epoxy at -65°F (-54°C).

Case 2: Bi-material Thermal Stress (Table 2.1-1, Items 6, 8, 13).
This is caused by the caps on the resistor core.

The bi-material case can be obtained directly from Equation 2-59 of Reference 1 which follows Figure 2.1-6. The difference in radial expansion is half the diameter expansion, or

$$\Delta r = \frac{B}{2} (\alpha_a - \alpha_b) \Delta T = 2.59 \times 10^{-5} \text{ inch (0.65 } \mu\text{m)}, \quad (25)$$

and the interfacial radial pressure is found to be

$$P_b = \frac{\Delta r}{\frac{B}{E_a} \left(\frac{C^2 + B^2}{C^2 - B^2} + \mu_b \right) + \frac{B}{E_b} \left(\frac{B^2 + A^2}{B^2 - A^2} - \mu_a \right)} = 4080 \text{ psi (28.1 MPa)} \quad (26)$$

The maximum hoop stress in the core is at the outer surface where

$$\sigma_{tao} = \frac{-P_b (B^2 + A^2)}{B^2 - A^2} = -5100 \text{ psi (35.3 MPa)} \quad (27)$$

and the maximum hoop stress at the inner surface of the cap is

$$\sigma_{tbi} = \frac{P_b (C^2 + B^2)}{C^2 - B^2} = 26,700 \text{ psi (184.1 MPa)}. \quad (28)$$

A safety factor of 2 remains before the copper caps yield.

For those caps affected by radial tension thermal stress (Item 13), results can be obtained by repeating the above to get $\Delta r = 1.691 \times 10^{-5} \text{ inch (0.42 } \mu\text{m)}$, $P_b = 36.4 \text{ psi (250 kPa)}$, $\sigma_{tao} = -226 \text{ psi (-1.5 MPa)}$, and $\sigma_{tbi} = 120 \text{ psi (827 kPa)}$.

Figure 2.1-6. Bi-Material Cross Section

Both stresses are safe. A more accurate solution for this and for compression thermal stress would use the formulation in Case 1 above for core, cap, and epoxy cover layers. In reality, a five-layer case should be developed with the En-4 coating and the GMB-epoxy encapsulant included in the full simultaneous solution. A stress concentration of 2 should be used to find the radial stress in the core at the end of the cap.¹

The leads subjected to radial tension thermal stress in Item 13 are solved again with the two layer case where radius $A = 0$, to get

$\Delta r = 6.038 \times 10^{-6}$ inch (0.15 μm), $P_b = 181$ psi (1.2 MPa), $\sigma_{tao} = -181$ psi, and $\sigma_{tbi} = 273$ psi (1.8 MPa). These are all safe.

Case 3: Axial Thermal Stress (Table 2.1-1, Items 7, 8, 15). The model is shown in Figure 2.1-7 where the core is Element 1, the two caps are Element 2, the two layers of cover are Element 3, the two layers of the coating are Element 4 and the encapsulant is Element c, the constraint. The four-element rod is considered to be constrained laterally and longitudinally. Free slippage at the interface is assumed. The force on each element in the bar is then given by Equation 29 (the leads are neglected).

$$F_B = \frac{(\alpha_1 L_1 + \alpha_2 L_2 + \alpha_3 L_3 + \alpha_4 L_4 - \alpha_c L_c) \Delta T}{\frac{L_1}{A_1 E_1} + \frac{L_2}{A_2 E_2} + \frac{L_3}{A_3 E_3} + \frac{L_4}{A_4 E_4} + \frac{L_c}{A_c E_c}} = 22.1 \text{ lb (98.3 N)}. \quad (29)$$

The stress in each bar element is then

$$\sigma_i = \frac{F_B}{A_i} \quad (30)$$

Figure 2.1-7 Model of Four
Element Bar/
Constraint

so that the axial stress in the core is $\sigma_1 = 8,800$ psi (60.6 MPa), the axial stress in the caps, epoxy cover, and EN-4 coating at the end of the resistor are $\sigma_2 = \sigma_3 = \sigma_4 = 7,820$ psi (53.9 MPa), and the axial stress in the GMB-epoxy encapsulant is in the opposite direction, or $\sigma_c = -0.782$ psi (5.3 kPa). None of these stresses is dangerous.

Case 4: Stress Concentration (Table 2.1-1, Item 5). This stress is determined from the grooves in the core and can be obtained from reference Figure 20-15 for tension and Figure 20-16 for bending.²

The notation is defined in Figure 2.1-8. For the existing ratios of $H/R = 0.143$ and $R/A = 0.350$; the stress concentrations are tension $K_t = 1.5$, and bending $K_t = 1.7$.

When these stress concentration factors are applied in previous cases, all are still safe.

Case 5: Stress Concentration (Table 2.1-1, Item 18). Stress in the epoxy cover caused by grooves in the core is determined. An approximation for this case can be obtained from reference Figure 20-58.² The case of bending will be nearly the same as the tension case which is illustrated in Figure 2.1-9.

$R/H = 0.0715$; therefore, the concentration is $K_t = 1.47$.

Figure 2.1-8. Hollow Resistor Core

Figure 2.1-9. Protrusion on Flat Plate

Applying this stress concentration still leaves ample safety for earlier cases.

Case 6: Stress Concentration (Table 2.1-1, Item 16). This is stress caused by spherical voids in the epoxy cover. A first approximation to this case is found from Equation M^3 (Figure 2.1-10). For a bar under tension with a spherical void, the maximum stress concentration is on the equatorial plane and only depends on Poisson's ratio which is about 0.38 for epoxy.

$$K_t = \frac{27-15\mu}{14-10\mu} = 2.09 \quad (31)$$

Figure 2.1-10. Spherical Void
in Bar

The stress concentration drops off quickly to 1.05 at twice the void radius. This does not provide the stress concentration for a void near the epoxy surface, but even if doubled, the epoxy cover localized stress would still be safe.

Case 7: Cap Corner Stress (Table 2.1-1, Item 17). This stress is caused by the cap corner. Because the stress concentration factor is only dependent on geometry, an inside corner is nearly equivalent to an outside corner. The case in Figures 20-1, 20-2, and 20-3² for the geometry shown in Figure 2.1-11 where

$\frac{R}{D} = 0.15$ and $\frac{H}{R} = 8.33$ gives the concentration below.

Tension $K_t = 1.85$.

Bending $K_t = 1.65$.

Torsion $K_t = 1.37$.

Case 8: Axial and Radial Buckling Loads (Table 2.1-1, Items 1, 2). The core can buckle as a bar under end loading while supported elastically by a lateral pressure⁴ (Figure 2.1-12). The cross-sectional moment of inertia of the core is

$$I = \frac{\pi}{4}[B^4 - A^4] = 6.28 \times 10^{-7} \text{ in}^4 (2.61 \times 10^{-13} \text{ m}^4). \quad (32)$$

The lateral force constant is a composite elastic modulus, E_{ser} , of the two materials in series between core and encapsulant, or

$$\beta = E_{\text{ser}} = \left[\frac{C-B}{(D-B)E_b} + \frac{D-C}{(D-B)E_c} \right]^{-1} = 157,000 \text{ psi (1.1 GPa)}. \quad (33)$$

Figure 2.1-11. Circular Fillet on Solid Shaft

Figure 2.1-12. Axial and Radial Buckling

The number of buckling half waves is then

$$M \geq \frac{1}{2} \left\{ \left[1 + \frac{4L^2}{\pi^2} \left(\frac{\beta}{E_a I} \right)^{\frac{1}{2}} \right]^{\frac{1}{2}} - 1 \right\} = 0.37. \quad (34)$$

$M = 1$ and the critical axial buckling load is then

$$p^1 = \frac{\pi^2 E_a I}{L^2} \left[M^2 + \frac{\beta L^4}{M^2 \pi^4 EI} \right] = 5200 \text{ pounds (23.1 kN)}. \quad (35)$$

Comparing this with the applied load to the core from Case 3 and applying the stress concentration for grooves from Case 4, the actual applied load of 37.6 pounds (167 N) is safe.

Radial buckling of the core can be approximated from Case 4, Table 16.⁴

$$p^1 = \frac{Et^3}{4(1 - \nu^2)R^3} = 1.20 \times 10^7 \text{ psi (8.3 GPa)} \quad (36)$$

and compared to the applied thermal stress of 958 psi (6.6 MPa) from Case 1 is safe.

Case 9: Meridional bending (Table 2.1-1, Item 3). The meridional bending caused by a circumferential line load⁴ could be generated from a circumferential nonuniformity in the EN-4 coating. It then transmits the GMB-Epoxy loading to the cover and core as shown in Figure 2.1-13. The line load then is nearly

$$P = 2\pi B\sigma_{tci} = 767 \text{ pounds/inch (134.3 kN/m)} \quad (37)$$

and

$$\lambda = \left[\frac{3(1-\nu^2)}{R^2 t^2} \right]^{\frac{1}{4}} = 64.8 \text{ in}^{-1} \text{ (2.55 mm}^{-1}\text{)} \quad (38)$$

so that the meridional bending stress is

$$S_1^1 = \frac{-3P}{2\lambda t} = 44,400 \text{ psi (306 MPa)} \quad (39)$$

Figure 2.1-13. Meridional Bending

This stress is dangerous, but such a concentrated circumferential line load would probably never occur in actual potted configurations. The coating and epoxy cover will spread them.

Case 10: Simple End Bending (Table 2.1-1, Items 4, 11). The force is caused by the differential thermal expansion change in length, which, for the dimensions shown in Figure 2.1-14, is

$$\Delta L = L_1(\alpha_a - \alpha_d)\Delta T + L_2(\alpha_{cu} - \alpha_d)\Delta T = 0.0016 \text{ inch (40.6 } \mu\text{m)}. \quad (40)$$

The resulting force is from the GMB-Epoxy constraint

$$F = \frac{E_d A \Delta L}{L} = 1.99 \text{ pounds (8.8 N)} \quad (41)$$

which yields a resultant bending stress in the cover and core

$$S^1 = \frac{F y C}{I} = 2890 \text{ psi (19.9 MPa)} \quad (42)$$

because the cross-sectional moment of inertia of the core and cover is

$$I = \frac{\pi}{4} [C^4 - A^4] = 2.58 \times 10^{-6} \text{ inch}^4. \quad (10.7 \times 10^{-12} \text{ m}^4).$$

Figure 2.1-14. Simple End Bending

This will be safe even after applying the stress concentration factor of Case 4. Also, the constraint of the lead wire by GMB-Epoxy will rapidly lower this estimate.

Case 11: Surface Shear-Compression (Table 2.1-1, Item 9). This compression is determined in the core from the epoxy cover. The shearing stress is

$$S_s = \frac{F_s}{A_s} \quad (43)$$

where the shearing force is from the differential expansion between the core and the cover. This is found from the constrained formulation of Equation 29. The copper caps provide the end connection. Because only one element each is in the bar and in the constraint, the lengths are equal; therefore, the force is

$$F_S = \frac{(\alpha_1 - \alpha_c)\Delta T}{\frac{1}{A_1 E_1} + \frac{1}{A_c E_c}} = 4.68 \text{ pounds (20.8 N)} \quad (44)$$

because the core cross-section is

$$A_1 = \pi(B^2 - A^2) = 0.00251 \text{ inch}^2 (0.16 \mu\text{m}^2),$$

the cover cross-section is

$$A_c = \pi(C^2 - B^2) = 0.00353 \text{ inch}^2 (2.2 \mu\text{m}^2), \text{ and}$$

the shear area is

$$A_s = \pi BL = 0.01696 \text{ inch}^2 (10.9 \mu\text{m}^2) \quad (45)$$

where L is the length between caps, so that $S_s = 276 \text{ psi}$ (1.90 MPa).

This will not damage either material.

Case 12: Surface Shear-Tension (Table 2.1-1, Item 10). Tension from copper caps and epoxy cover is determined. This uses Equations 43 and 44 and uses the copper cap length for L in Equation 45. The shear stress of 1240 psi (8.5 MPa) is a safe value.

Case 13: Thermal Stress-axial Tension (Table 2.1-1, Item 14). The force will be the same as Equation 44. This will cause a tensile stress in the core of $F/A_1 = 1860 \text{ psi}$ (12.8 MPa) and a compressive stress in the cover of $-F/A_c = -1330 \text{ psi}$ (-9.2 MPa).

Both values are safe, even when the cap corner stress concentration of Case 7 and the groove stress concentration of Cases 4 and 5 are applied.

Case 14.: Centerline Bending (Table 2.1-1, Item 12). This case can be approximated as Mohr's circle addition of the radial pressure on the leads from Case 2 and the bending moment from Case 9. The principal stresses are then found by inserting $\sigma_x = 0$, $\sigma_y = 2,890 \text{ psi}$ (19.9 MPa), and $\tau_{xy} = 181 \text{ psi}$ (1.2 MPa) into Equation 46 to 49.

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 2,900 \text{ psi (19.9 MPa)}. \quad (46)$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = -11 \text{ psi (-75 kPa)}. \quad (47)$$

$$\tau = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = 1,460 \text{ psi (10.0 MPa)}. \quad (48)$$

$$\theta = \frac{1}{2} \tan^{-1} \left(\frac{-2\tau_{xy}}{\sigma_x - \sigma_y} \right) = 3^\circ 35'. \quad (49)$$

Again, no danger is predicted.

2.1.4. Comparison With Failure Criteria

Throughout the stress analysis above, each individual stress was compared with the ultimate tensile, compressive, or bending strength of that material. Few individual loadings caused stresses approaching the danger level even when appropriate stress concentrations were included. However, when several loadings are superimposed, the resulting stress must be locally compared with the strength. The stresses must be converted to strains for comparison with a failure criteria based on strain.

The preceding set of loadings is not exhaustive. If another loading occurs in a particular case, it must be added. One example of such a loading is a bending perpendicular to the longitudinal axis, perhaps transmitted by a bending PC board on which the resistor is mounted.

REFERENCES (Section 2.1)

¹Joseph E. Shigley, *Mechanical Engineering Design, Second Edition*. New York: McGraw-Hill, 1972.

²Charles Lipson and Robert C. Juvinall, *Handbook of Stress and Strength*. New York: Macmillan, 1963.

³Stephan P. Timoshenko and J. N. Goodier, *Theory of Elasticity, Third Edition*. New York: McGraw-Hill, 1970.

⁴Raymond J. Roark, *Formulas for Stress and Strain, Fourth Edition*. New York: McGraw-Hill, 1965.

2.2 The Simplex Potting Technique (W. J. Stone)

Two philosophical paths are being pursued in this manual, one made up of more general design approaches, the other pursuing particular effects of those solutions. This chapter discusses one of the general techniques for solving potting problems.

The Simplex Potting Technique is one general design technique for systematizing Bendix Kansas City potting procedures. Several such approaches have been proposed previously (References and Chapter 2.1 of this Manual); however, they are either too sophisticated or too time consuming for everyday usage. The plan proposed here is tailored to the practicing Bendix Kansas City potting engineer.

The Simplex Potting Technique is a plan for achieving a best engineering compromise in the design of a potted assembly once a fixed envelope geometry, an allowed set of potting materials, and a required set of electronic components are given. This scheme approaches the Simplex Ideal potting model module by application of a simple but often adequate, systematic approach to evaluate and solve problems of potted components. It is purposely open-ended and flexible to grow and to be modified with advances in the field. The Simplex Ideal is a modularly repackaged component which is not significantly affected by its functional thermo-mechanical environment and which does not unfavorably affect the rest of the potted assembly.

2.2.1 Objectives

- Every potted component does not merit extensive reevaluation. First a sorting technique will be defined to find the best targets for the available effort.
- Every possible thermomechanical loading cannot be examined, so the problem is to be forced into a limited, well-defined case of loading.
- Off-the-shelf commercial electronic components will be used because of their ready availability, long functional histories, proven operational characteristics, and mass production economies in lieu of redesigning a component as in Chapter 2.1.
- A system will be described for potting assemblies that can be instrumented and monitored for experimental proof testing.
- A system will be described that can be modeled mathematically for easy optimization of the material, geometry, and size of those few factors under our control.
- This system of potting will be the simplest, barely adequate technique for potting as defined by a practical potting engineer.

2.2.2 Simplifying Assumptions

Each electronic component to be potted, no matter how complex in actual structure, can be considered as a simple entity (one not composed of parts) with a simple geometry and can be represented by a set of effective materials properties, a pseudo material.

A conformal coating can be designed in material, size, and geometry so that all thermomechanical loads transmitted through it are moderated to acceptable levels and are distributed well enough to be practically described as hydraulic loadings.

Sufficient volume and distribution freedom in the assembly is available to attain these design objectives:

- All parts will be under a nominal compressive load at all functional states to assure positional integrity.
- Individual parts will have adequate spatial isolation.
- Any penetrations of the conformal coating (such as electronic lead-wires) can be accomplished without significant loadings.

Component deflections can be kept small and elastic by proper design so that simple relationships and superpositions can adequately describe their behavior.

When these assumptions are coupled with the knowledge that most electronic components survive relatively high mechanical loadings if the loads are well distributed, the problem is greatly reduced in complexity. The potted assembly can be forced into the well-defined province of hydraulic loading. This resulting plan for thermomechanical potting optimization will be referred to as the Simplex Potting Technique.

2.2.3 The Simplex Modeling Procedure

The following steps are the basis of the Simplex Potting Technique. The flow chart of Figure 2.2-1 shows the sequence of application.

- All candidate components are sorted by hydraulic loading and concurrent monitoring for functional damage at normal Bendix Kansas City potted loadings.
- The failure criteria and effective composite material properties are established for any inadequate component.
- The maximum hydraulic loads caused by the thermal interaction between the component and required potting materials are calculated using their geometries, material properties, and functional temperature--a mathematical component module model.

Figure 2.2-1. Flow Chart of Simplex Potting Technique

- The failure criteria are applied, and when the results of this step are approached by those of the preceding step, the component is repackaged by optimizing the model used in the preceding step.
- The remaining particular problems, such as superimposed line and point loads from external sources, packing problems, and so forth, are solved by classical means (Chapter 2.1).

Packing factors can be modeled with the Mean Exclusion Radius technique. These highly particularized problems are not now amenable to a general solution but are discoverable before-the-fact by experimental models and after-the-fact by practical experience with the failed assemblies. They exceed the current scope of the plan.

The Simplex Technique is optimized by approaching pure hydraulic loading through conformal coating design. The Hydraulic analogy as delineated in Chapter 2.3 is thus the theoretical basis of the Simplex Technique. As the potted assembly approaches the repackaged Simplex Ideal, the analysis by this model becomes "more valid" and the number of problems encountered in the last step are minimized.

Some of the equipment and methods for the significant material parameter measurements are described in Chapter 3.3 and 3.4. The trapped modulus concept and the current math models used in the third and fourth steps are described and exercised in Chapters 1.2, 2.4 and 2.1.

With the above tools and concepts, the optimum repackaging called out in the next-to-last step can be accomplished. In general, it will result in the selection of the best confromal coating material in terms of differential thermal expansion, stiffness, and thickness to minimize external point and line loads. If the technique is exercised on enough of the weaker, more vulnerable components, then their undesirable thermomechanical interactions will be minimized and a viable potted assembly will be the result.

REFERENCES (Sections 2.2)

¹Deleted.

²Deleted.

³F. D. Swanson, "The Use of Conformal Coatings to Relieve Encapsulation Stresses," *Insulation*. February 1966.

⁴F. H. Gahimer, *Coatings to Reduce Encapsulation Stresses*. Indianapolis: Bureau of Naval Weapons, Report NAFI TR-814, March 1966.

2.3 The Hydraulic Analogy (W. J. Stone and G. D. Swanson)

This chapter presents one general approach to potting problem solutions. A detailed example using an experimental model is presented in discussing a ceramic Sprytron tube in its potted assembly.

Test laboratory studies of a ceramic Sprytron tube¹ developed a proficiency for evaluating the hydraulic component of external loadings and for optimizing conformal coatings. Conformal coatings are of significant value in adjusting load levels by plastic flow, spreading out concentrated loads, and in providing a potting element of adjustable thermal coefficient that reduces differential expansion loads. As potting schemes for Sprytrons evolved, a general movement was seen toward the optimization of these three requirements and toward a hydraulic loading of this component.

If a component is to be loaded in a hydraulic mode, the following repackaging scheme is valid.

- The component is test loaded hydraulically to its failure level, and then
- The component is repackaged in a hydraulic constraint in which that level is never exceeded and then it will never fail.

This is achieved by strain gaging the experimental component model in its sensitive areas and calibrating this transducer under controlled hydraulic conditions. An existing computer program allows selection of the best conformal coating with regard to its thermal coefficient of expansion, modulus, and thickness; the ability of the coating to support a shear over a period of time should also be considered. A prototype is assembled and cycled through its prescribed environment with allowances for thermal transients and spatial gradients. When the resultant loads are hydraulic, any strain gage reading can be interpreted from the hydraulic calibration curves of the experimental component model by comparing that gage with the hydraulic failure criteria. When the loads are far below failure, the unit is safely packaged; but as these criteria are approached, safety is less certain. An engineering safety factor can be defined to ensure a safety margin.

Nonhydraulic loads will be indicated by the relative change in the ratio of the readings of the other strain gages. Such loads can be the result of penetrants of the conformal coating (such as lead wires), or the interaction with nearby components, supports, or constraining containers. The success of design modifications in eliminating lead wire load problems can be assessed by similar experiments.

2.3.1 A Technique for Evaluating Potting Packages

When the strain gages in a component package reveal that the component is experiencing significant external loads, a modification of the package is in order. The strain readings are confounded by all the superimposed loads and interactions. One must establish which loading caused which strain effect and then rate the significance of each.

Three categories of external loads will be considered in assigning causalities: hydrostatic loads, static concentrated loads, and dynamic or transient loads.

For a simplified example assume that superposition holds; that is, no external loads interact with themselves or with internal loads, and that no sequence of application is significant.

Figure 2.3-1 is a graph of such a loading. The critical load level is different for different gage locations because the hydraulic failure criterion can vary from point to point on a real component.

By analyzing the load mixture indicated by the strain gage readings and by discovering their percentage of contributions, a basis is obtained for concentrating on the more significant loads; and some required characteristics of the individual loading mechanisms are defined.

Figure 2.3-1. Load at Strain Gage Positions

2.3.2 Hydraulic Evaluation

It is assumed for all potted components that the external load matrix can be divided into two categories: well-distributed and concentrated. The ideal well-distributed load is that load with the magnitude perpendicular to any surface element equal to that on any other element. This satisfies Pascal's Principal and permits modeling with a hydraulic analogy. The benefit to the analytical treatment of potting problems is the simplification which comes from reducing the infinite number of possible external force matrix and geometric matrix combinations. Only one well-defined external load pattern remains.

When one applies hydraulic pressures within the linear elastic range to a homogeneous, solid right cylinder, it will always remain a right cylinder. Repeating the experiment at the same pressures would repeat the loads at all points. When there was a strain gage at some point, its reading would repeat; and one could measure the modulus of the material. Any identical strain gage at any other similarly oriented point should give the identical reading.

$$\frac{\sigma_1}{\sigma_2} = \frac{\epsilon_1}{\epsilon_2} = \frac{E\epsilon_1}{E\epsilon_2} = R$$

where

σ = stress

ϵ = strain

E = modulus of elasticity

R = a constant

1,2 = subscript referring to point locations.

One may use this technique to evaluate the quality of the pressurizing medium versus a perfect hydraulic fluid.

When a thin-walled, hollow, right cylinder is substituted for the solid, it becomes a more sensitive strain gage transducer. A geometrical effect is now superimposed on the previous, purely dilational case. When no elastic limits are exceeded, the same pressures will repeat the same strains at a given location; however there can be a different apparent modulus at each point on the transducer.

$$\frac{\sigma_1}{\sigma_2} = 1 = \frac{(f_1 E) \epsilon_1}{(f_2 E) \epsilon_2}$$

Here f_n ($n = 1, 2$) is a function of geometry, so that

$$(f_n E) \neq (f_{(n+1)} E),$$

but each f is a constant for that point.

$$\text{Therefore, } \frac{\epsilon_1}{\epsilon_2} = R'$$

Where R' is a constant for identical gages at these locations for a given hydraulic load. Any deviation from this constant is a measure of external load concentration. It could therefore be used to measure these variables.

- How well a "solid" performs as an effective hydraulic fluid,
- The time dependence for supporting shears in a visco-elastic solid,
- The thickness of a conformal coating necessary to distribute a load; or
- Interactions between adjacent components potted together in a visco-elastic medium.

The Simplex Potting Technique recognizes these functions of a conformal coating:

- To protect components by converting concentrated loads to well-distributed loads;
- To reduce mechanical loadings on components by matching thermal expansion coefficient, bulk modulus, and thickness; and
- To relieve residual stresses by plastic flow during thermal excursions.

Hydraulic Evaluation of a Candidate.

Using classical elasticity theory, a stress tensor can be separated into two independent components. One component, the mean normal stress, represents only a volume change. The other component, the deviatoric stress, represents only geometric distortion. An attempt to obtain such a separation for one component is described below. The ceramic vacuum tube was proposed for potted assemblies in several possible packages (Figure 2.3-2).

Loading a component in a hydraulic fluid provides an ideal case in which the normal stress is uniform everywhere and the deviatoric stress is zero because a fluid will not support a shear. Strain

Figure 2.3-2. Ceramic Vacuum Tube

gages were mounted on the vacuum tube (Figure 2.3-2), a hydraulic load applied, and the strain-pressure calibration curves were obtained (Figure 2.3-3). This defines the zero deviatoric stress condition.

This vacuum tube was packaged and carried through thermal cycling with measured strain normalized to zero at 160°F (71°C). Gage 6 is expected to have little if any contribution from deviatoric stresses, so one can assume its reading came solely from the uniform, normal pressure. This strain and the hydraulic calibration (Figure 2.3-3) predict what strain any other gage should have if the package caused only a uniform, normal stress. Any deviation from this value was caused by deviatoric stresses.

Figure 2.3-3. Hydraulic Load Calibration of Tube

For each gage position, a deviatoric ratio, D , can be calculated.

$$D = \frac{\epsilon_h - \epsilon_n}{\epsilon_h}$$

Here, ϵ_h is the hydraulic strain and ϵ_n is the actual normalized strain. Small numbers (+ or -) are closer to a purely hydraulic condition.

The calculated values of D for the first thermal cycle and the average of four thermal cycles of several packages investigated are shown in Table 2.3-1 with the average absolute value for each component. Table 2.3-2 shows the hydraulic package rating based on the deviatoric ratios.

Linear Rating Scale

A linear scale for rating the deviation from a uniform, hydraulic loading can be derived in several ways. This method adds the magnitudes of the average gage deviation at -65°F (-54°C) and 77°F (25°C). This yields a positive number which can be compared on a linear basis (Table 2.3-3).

These ratings only compare packages on the basis of how closely they approach uniform hydraulic loadings. Some auxiliary rating

Table 2.3-1. First Cycle and Four Cycle Average Deviation - Gage 6 = Reference

First Cycle Deviation ¹ (D)									
Package	1	2	3	4	5	7	(°F)	(°C)	$\overline{D}$
E02-246-016	X	-0.46	-0.04	-0.07	0.89	-0.29	-65	-54	0.35
	X	0.27	1.40	1.46	1.87	-0.88	77	25	1.18
E01-242-014	0.72	-0.95	-2.88	-2.82	-0.16	-0.07	-65	-54	1.27
	0.04	-0.75	-0.78	-0.85	0.03	-0.13	77	25	0.43
E01-242-008	-0.31	-1.19	-1.73	2.82	-1.44	1.11	-65	-54	1.43
	-0.61	-2.49	-2.26	-0.72	-1.31	0.53	77	25	1.32
E01-242-001	19.89	0.26	52.68	47.07	4.10	15.34	-65	-54	23.22
	0.00	-1.02	-1.33	-0.89	0.65	-0.96	77	25	0.81
E02-246-014	0.76	-0.35	-0.50	-0.37	0.63	-0.91	-65	-54	0.59
	0.54	-0.55	-0.31	-0.27	0.71	-0.49	77	25	0.48
Plus: Bowed out. Minus: Bowed in. X: Data for this gage not available.									

Table 2.3-1 continued. First Cycle and Four Cycle Average Deviation -
Gage 6 = Reference

Four Cycle Average Deviation (D)									
Package	1	2	3	4	5	7	(°F)	(°C)	$\overline{ D }$
E02-246-016	X	-0.46	-0.10	-0.08	0.88	-0.42	-65	-54	0.39
	X	0.04	1.64	1.82	1.14	-1.20	77	25	1.17
E01-242-014	0.59	-0.96	-2.95	-2.92	-0.92	0.28	-65	-54	1.44
	-0.52	-0.90	-1.01	-0.99	-0.52	-0.53	77	25	0.75
E01-242-008	0.08	-1.09	-1.55	12.18	-0.99	1.12	-65	-54	2.84
	0.42	-1.57	-1.24	4.98	-0.03	0.42	77	25	1.44
E01-242-001	-5.58	-2.40	-17.58	-15.23	-0.96	-5.20	-65	-54	7.82
	-0.07	-1.39	-1.60	-1.02	0.68	-1.10	77	25	0.98
E02-246-014	0.72	-0.28	-0.42	-0.34	0.71	-0.89	-65	-54	0.56
	0.54	-0.56	-0.27	-0.28	0.73	-0.49	77	25	0.48
Plus: Bowed out. Minus: Bowed in. X: Data for this gage not available.									

Table 2.3-2. Hydraulic Package Ratings Based on the Deviatoric Ratios. Higher ratings have lower magnitude numbers for D and $\overline{D}$

Test I.D.	First Cycle			Four Cycle Average		
	Total Rating	D Rating	$\overline{D}$ Rating	Total Rating	D Rating	$\overline{D}$ Rating
E02-246-016	2	2	2	2	2	2
E01-242-014	3	3	3	3	3	3
E01-242-008	4	4	4	4	4	4
E01-242-001	5	5	5	5	5	5
E02-246-014	1	1	1	1	1	1

Table 2.3-3. Data for Linear Rating Scale

Rating	Package I.D.	First Cycle	4 Cycle Average
1	E02-246-014	1.07	1.04
2	E02-246-016	1.53	1.56
3	E01-242-014	1.70	2.19
4	E01-242-008	2.75	4.28
5	E01-242-001	24.03	8.81

is necessary to decide whether this loading is excessively high at any point for the safety of the packaged component. One such auxiliary rating system is based on an equivalent hydraulic load, a scale for weighting external forces on a component.

Equivalent Hydraulic Load

When a component is gaged as a transducer, a hydraulic calibration can be made. When no elastic limits have been exceeded, the strain at any gage can thereafter be used as a measure of hydraulic loading.

Components are sophisticated structures; therefore, combinations of point loads would not duplicate the hydraulic load pattern

because there would be interactions between such loads. Here the sequence of loading could be significant. Despite these restrictions, superposition is assumed and the equivalent hydraulic load is defined as that hydraulic load which could have caused an equal strain at any position monitored on the component.

This concept is a normalization to a common scale which allows a comparison of the percentage contribution of each external force whether well distributed or concentrated, static or dynamic. This concept is most valid as the package approaches the Simplex Ideal potting package discussed in Chapter 2.2.

An example using this technique is the "Vienna Sausage" Package E01-242-001 (Figure 2.3-4). It produced the data shown in Table 2.3-4 for the gage readings (ϵ_{ex}) and the Equivalent Hydraulic Loading (SE_H) on the first cycle of thermal loading.

Figure 2.3-4. Vienna Sausage Potting Configuration

Table 2.3-4. Data for Package E01-242-001

Value	Units	Temp.	Gage Location						
			1	2	3	4	5	6	7
ϵ_{ex}	$\mu\epsilon$	-65°F (-54°C)	-107	56	-440	-403	-73	-37	-142
S_{EH}	PSI	-65°F (-54°C)	1850	-400	4800	5250	300	300	1200

A significant lack of hydraulic uniformity or concentration of forces along the longitudinal axis is indicated. The compressive loads at Positions 3 and 4 are rated significant to dangerous, and Position 2 has even changed sign, indicating a high degree of bending in the base. This information enhances the knowledge of the part behavior gained from the deviatoric ratio data presented in Table 2.3-1.

This package was redesigned according to the principles discussed in this manual and appears in redesigned form as package E02-246-016, which is acceptable as can be seen in Table 2.3-3. This illustrates the usefulness of the hydraulic analogy concept.

REFERENCE (Section 2.3)

¹Deleted.

3. MATERIAL PROPERTIES AND SELECTION

3.3 Special Test Procedures

The various general and specific potting conditions and configurations for electronic components sometimes require new, non-standardized test procedures for evaluation. This Chapter includes descriptions of such test procedures and of the special equipment used. Most of this equipment was developed and fabricated at Bendix Kansas City. The goal is to evaluate the appropriate properties of unfilled and filled plastics and elastomers from various cures for use in electronic potting. Some serve as quick functional screening tests, and some produce accurate, quantitative material properties for use in engineering design calculations. As design models and mathematical models advance in applicability, new test procedures will be added.

3.3.1 Bulk Modulus by Cylinder with Double Pistons (F. E. Lormis)

The following procedure describes the bulk modulus testing performed in the Bendix Laboratory (Figure 3.3-1). Bendix Kansas City fixture after Green and Brady, LRL engineering note, ENS 67-287 (letter 9/11/70, Reference R. L. Brady to F. E. Lormis).

- **Specimen.** The test specimen is 0.440 inch (11.1 mm) in diameter and 0.5 to 1.0 inch (12.7 to 25.4 mm) long.
- **Testing Speed.** The testing speed should not exceed 100,000 psi/min. (689.4 MPa/min.), so it depends on the modulus of the material tested and the length of the test specimen. A cross-head speed of 0.050 inch/min. (1.2 mm/min.) is convenient for some plastics.
- **Test Procedure.** The upper piston is removed and the specimen placed in the cylinder. The upper piston is then replaced and the fixture is then placed into a compression tester. The load is applied at the proper rate and the relative piston movement versus load is recorded. These results are then used to calculate the ratio of stress to axial strain which is K' . This value then can be converted into the hydraulic bulk modulus K by the equation (Section 1.2.5)

$$K = \frac{K'(1 + \mu)}{3(1 - \mu)}$$

where μ is Poisson's ratio.

- **Apparatus.** The testing fixture consists of a cylinder with two pistons. The test specimen is inside the cylinder between the two pistons.

The cylinder is 0.440 inch (11.1 mm) in diameter, and the pistons are polished for a close fit of less than 0.0005 inch (12.7 μ m) clearance. The cylinder and pistons are made from stainless steel.

Figure 3.3-1. Bulk Modulus Test
Fixture

3.4 Material Properties (G. D. Swanson and F. E. Lormis)

3.4.1 Mechanical and Thermal Properties

This manual was written to aid in accumulation of potting design philosophy, general formulae, and material data so that the processing engineer could better choose compatible materials initially, and so that later design changes to solve one problem would avoid creating others. To these ends, this chapter presents material property data that may be used in design models and mathematical models. These models then may aid in selecting the initial set of compatible materials which will provide safe thermo-mechanical loading levels. They may also aid in later redesign when problems arise during pilot production of new electronic modules.

This chapter presents mechanical thermal, and electrical properties of materials¹ suitable for use as potting compounds, conformal coatings, and resistors. These materials are expected to provide potting protection from failure caused by environmental conditions experienced during manufacture, testing, storage, and usage.

Electrical characteristics are used to select materials before the mechanical design begins. The mechanical designer must maintain sufficient component spacing, and protective coatings thick enough to insulate electrical operation. Hence, such electrical properties as breakdown voltage should be available for ready reference during the mechanical design phase. Processing data not included will be written into a later edition.

In the tables which follow, a comparison of physical properties of several materials is made. Some values have been measured at -65°F (-54°C) and at 165°F (74°C). Included in the comparison are Adiprene L-100; Conathane EN-4 and EN-7; CS3100, Pro-Seal 727 and PRC 1201Q polysulfides; RTV 3140 silicone and Laminar X500, 7-C-23 epoxylated-urethane; two glass microballoon-filled epoxies; and Wesgo A1500 ceramic.

Whenever possible, standardized ASTM specifications were used. While ultimate tensile strength is reported for polymers and elastomers, the immense elongations encountered prohibit its use as a design criterion as is the practice with metals. Compressive ultimate strength is not reported for polymers and elastomers (except GMB-Epoxy) as this is too nebulous a concept for design usage.

The largest changes are in room temperature tensile strength, modulus at 200 percent elongation, tear strength, room temperature compressive modulus, and some torsional modulus data (Tables 3.4-1 through 3.4-4).

Table 3.4-1. Mechanical Properties of Polyurethanes

Material Cure	EN-4 24 hr., 160°F	EN-4 16 hr., 180°F	EN-4 10 hr., 200°F	EN-7 16 hr., 175°F
Specific Gravity	1.00	1.01	1.00	1.01
Hardness, Shore A	89	89	89	91
Thermal Expansion, $\mu\text{in./in./}^\circ\text{F}^*$				
-65°F to 75°F *	97	98	97	91.9
75°F to 165°F	75	75	75	65.4
Tensile Properties:				
Ultimate Strength, ksi*	2.00	1.83	1.58	2.07
Tangent Modulus at 200%, ksi	1.32	1.20	1.20	1.25
Elongation, percent	315	330	280	380
Poisson's Ratio	0.46	-	-	-
Tear Strength, lb./lineal in.*	275	214	251	250
Compressive Properties:				
Tangent Modulus at 300 psi,* ksi	5.55	-	4.90	3.63
Poisson's Ratio	0.45	0.48	0.47	0.49
Compression Set, %, 22 hr. at 75°F	-	-	-	51 at 25%
72 hr. at 158°F	67 at 10%	-	-	-
Shear Properties:				
Torsional Shear Modulus, ksi,				
-65°F	12.9-17.3	29.7	16.3	3.50
75°F	3.40-3.84	3.30	3.20	-
165°F	0.236-0.253	0.271	0.251	-
Lap Shear, psi	854	-	-	901
Loss Modulus, psi, -65°F	1100-1230	1410	1190	-
75°F	598-615	590	586	740
165°F	48-58	73	54	-
Loss Tangent				
-65°F	0.069-0.85	0.047	0.073	-
77°F	0.161-0.175	0.185	0.185	0.21
165°F	0.200-0.228	0.270	0.210	-
<p>*Metric Conversion: ksi = 1000 psi = 6.89 MPa, lb./lineal in. = 0.00567 N/m,</p> <p>$\mu\text{in./in./}^\circ\text{F} = 1.8 \mu\text{m/m/}^\circ\text{C}$, $^\circ\text{C} = \frac{5}{9} (^\circ\text{F} - 32)$</p>				

Table 3.4-2. Mechanical Properties of Adiprene and Laminar X500

Material* Cure	Adiprene L-100 MOCA			Laminar X500, 7-C-23
	75°F	4 hr., 160°F	3 hr., 200°F	
Specific Gravity	-	1.10	1.10	-
Hardness, Shore A	92	92	90	-
Thermal Expansion, μin./in./°F*				
-65°F to 75°F*	105	109 - 110	109	-
75°F to 165°F	107	106 - 108	109	-
Tensile Properties				
Ultimate Strength, ksi*	3.25	4.68	4.62	-
Tangent Modulus at 200 percent, ksi	1.19	1.47	1.52	-
Elongation, percent	600	430	420	-
Tear Strength, lb./lineal in.*	370	344	311	-
Compressive Properties				
Secant Modulus, ksi	9.50	9.10	7.53	-
Poisson's Ratio	0.45	0.47	0.41 - 0.47	-
Compression Set, percent				
22 hr. at 75°F	-	14 at 25%	14 at 10%	-
72 hr. at 75°F	-	12 at 10%	14 at 10%	-
Shear Properties				
Torsional Shear Modulus, ksi,				
-65°F	65.0	61.4	54.0	-
75°F	3.10	3.20 - 3.31	3.10	-
165°F	2.71	3.44	3.68	-
Lap Shear, psi*				214
Loss Modulus, psi				
-65°F	2570	2890 - 3400	2200	-
75°F	170	120 - 138	110	-
165°F	90	50	34	-
Loss Tangent,				
-65°F	0.039	0.049 - 0.073	0.033	-
75°F	0.055	0.035 - 0.039	0.035	-
165°F	0.033	0.016	0.009	-
*Metric Conversions: ksi = 1000 psi = 6.89 MPa, lb./lineal in. = 0.00567 N/m, μin./in./°F = 1.8 μm/m/°C, °C = $\frac{5}{9}(\text{°F} - 32)$				

Table 3.4-3. Mechanical Properties of Polysulfides and Silicone

Material	Proseal 727	Chemseal CS3100	Prod. Res. Corp. 1201 Q	Silicone RTV 3140
Specific Gravity	2.04	1.73	1.86	-
Hardness, Shore A, -65°F*	88	77	83	-
75°F	69	50	53	20
165°F	65	45	50	-
Thermal Expansion, $\mu\text{in./in./}^\circ\text{F}^*$				
-65°F to 75°F	-	-	164	164
75°F to 165°F	-	-	164	164
Tensile Properties:				
Ultimate Strength, ksi, -65°F	5.00	6.20	6.00	-
75°F	0.420	0.250	0.267	0.146
165°F	0.260	0.165	0.182	-
Tangent Modulus at 200%, psi *	-	-	-	61-70
Elongation, percent	-	-	-	240
Poisson's Ratio	-	-	-	0.44-0.45
Tear Strength, lb./lineal in., -65°F	80	175	135	-
75°F	60	32	43	18
165°F	60	20	23	-
Compressive Properties:				
Stress at 30% Comp., ksi, -65°F	7.50	6.10	5.50	-
75°F	1.80-	1.20	1.10	0.136 at 50%
165°F	1.90			-
Tangent Modulus at 300 psi, ksi, -65°F	0.600	0.630	0.700	-
75°F	5.63	6.30	5.20	-
165°F	1.91	2.20-	1.90	0.061 at 200 psi
Poisson's Ratio	1.96	3.20		0.375 at 50%
Compression set, %, 22 hr. at 158°F	0.48	3.15	2.26	-
Shear Properties:				
Torsional Shear Modulus, psi, -65°F	-	0.45	-	23 at 25%
75°F	261	110 at 25%	-	-
165°F	154-		-	39
Loss Modulus, psi, 75°F	163	154	-	-
165°F	19.5-	35	-	11.5
75°F	35		-	-
Loss Tangent	34	-	-	-
75°F	0.130-	0.226	-	0.295
165°F	0.133			
*Metric Conversions: ksi = 1000 psi = 6.89 MPa, lb./lineal in. = 0.00567 N/m, $\mu\text{in./in./}^\circ\text{F} = 1.8 \mu\text{m/m/}^\circ\text{C}$, $C = \frac{5}{9} (^\circ\text{F} - 32)$.				

Table 3.4-4. Mechanical Properties of GMB-Epoxy and Alumina

Type of Material	*GMB-Epoxy	GMB-Epoxy	Alumina
Cure Time, hrs.	24	24	
Cure Temperature, °F††	150	150	--
Specification No.	9927066**	9927085***	Wesgo A1500
Physical Properties:			
Tensile:			
Ultimate Strength, psi	9,280	5,540	35,000
Young's Modulus, psi	713,000	480,000	33,800,000
Poisson's Ratio	--	--	--
Tear Strength, lb/lineal in	--	--	--
Compression:			
Ultimate Strength, psi	--	--	--
-65°F	15,400	20,280	--
75°F	10,300	11,960	328,000
+165°F	4,100	5,320	--
Modulus, psi, -65°F	621,000	520,000	--
75°F	414,000	480,000	45,000,000
+165°F	210,000	260,000	--
Bulk Modulus, psi	--	--	25,000,000†
Poisson's Ratio	0.34	0.31	0.22
Shear:			
Shear Modulus, psi	121,000	--	18,000,000†
Loss Tangent	0.008	--	--
Loss Modulus, psi	1,140	--	--
Thermal Expansion, μ in/in/°F			
-65°F to 75°F	21	18	2.3
75°F to 165°F	22	21	2.5

*Glass micro-balloon filled epoxy

**Bendix

***3M-B40-A

†Ref. No. 2

††Metric conversions: $^{\circ}\text{C} = \frac{5}{9} (^{\circ}\text{F} - 32)$

1 psi = 6.89 kPa

lb/lineal in. = 0.00567 N/m

μ in/in/°F = 1.8 μ m/m/°C

The three polysulfides used as interchangeable materials are compared. Significant differences are found in the ultimate tensile strength as the testing temperature is changed from -65°F (-54°C) to 165°F (74°C). There are significant differences in the tear strength at all three test temperatures.

Thin film material is not satisfactory for physical testing because certain skin effects can alter properties, especially those showing dynamic character. Certain resinous coating materials (RTV 3140 and Laminar X500, 7-C-23 in this study) are not normally used in thick sections because of the excessive cure time required or because of the shrinkage occurring during the cure. Polysulfide compounds contain solvents to make them processable and also require excessive cure times to stabilize the cured resins. Gas pockets and bubbles will affect the dynamic properties as shown for Pro-Seal 727 in Table 3.4-3. To obtain material for physical properties testing, unusually long cure times were required for either solvents to be released or for moisture to penetrate as in the Dow Corning silicone RTV 3140.

Unless otherwise noted, properties were determined at room temperature. Traditionally, modulus values on elastomers are recorded at 300 percent strain offset. However, many of the materials reported here failed before reaching that point, so an offset value of 200 percent is used. Some stress-strain moduli were recorded at 300 psi (2.1 MPa) for some materials (instead of at 200 percent strain) because the slope of the stress-strain curve was very nearly flat at 200 percent (Figure 3.4-1).

Figure 3.4-1. Tangent Modulus Slopes

Note also that:

$$\text{Loss tangent} = \frac{\text{Shear modulus}}{\text{Loss modulus}}$$

Tear strength is reported in pounds per lineal inch width (N/m). Specimens of various thicknesses are tested, but the load at failure is extrapolated to a standard one-inch (25.4 mm) thickness.

Mechanical data are presented for glass micro-balloon filled epoxy and alumina (95 percent Al_2O_3) in Table 3.4-4. The alumina shown is similar to that of many resistor cores. The density, porosity, and purity of alumina can vary over a wide range causing the mechanical properties to vary correspondingly.³

When an elastomeric conformal coating material is trapped in the region between a stiff component and a stiff encapsulant, the classical Young's elastic modulus does not provide mathematical model results that agree with experimental results. However, when a trapped modulus value derived from the bulk modulus is used, the agreement is much better. Table 3.4-5 presents the measured ratio of applied axial stress to resulting axial strain, K' . The bulk modulus, K , is then calculated from K' and Poisson's ratio, μ , by

$$K = \frac{(1 + \mu)K'}{3(1 - \mu)}$$

and the trapped modulus, E_s , is then calculated theoretically as

$$E_s = 3(1 - 2\mu)K.$$

These derived values are also recorded in Table 3.4-5. For a Poisson's ratio near 0.5, as for these materials, an error of 5 percent can give an error of 5 to 7 percent in the bulk modulus which results in an error 40 to 100 percent in the trapped modulus value. These trapped modulus values can thus be in error, but are still much better than using the classical compressive modulus of Tables 3.4-1 through 3.4-3 in mathematical models.

The mechanical properties of the materials included in this section are dependent on the test temperature. Tables 3.4-1 through 3.4-4 record data explicitly at -65°F (-54°C), room temperature, and 165°F (74°C). Further information is now available for additional temperatures for GMB-Epoxy, 3M type B40 A per 9927085. Figure 3.4-2 shows graphs of temperature dependence on the compressive, tensile, and flexural properties. The vertical scale takes the units included on the graph for the modulus, ultimate stress, or maximum strain. The heat distortion temperature and glass

Table 3.4-5. Trapped Elastic Moduli

Material	Measured K' , ksi *				Bulk Modulus, ksi $K = \frac{1 + \mu}{3(1 - \mu)} K'$			Trapped Modulus, ksi $E_s = 3(1 - 2\mu)K$		
	-65°F*	75°F	165°F	μ^{**}	-65°F	75°F	165°F	-65°F	75°F	165°F
EN-4 (24 hr. at 160°F)	666	375	-	0.44	572	322	-	206	116	-
EN-4 (16 hr. at 180°F)	-	375	-	0.48	-	353	-	-	42.3	-
EN-4 (10 hr at 200°F)	-	370	-	0.47	-	342	-	-	61.5	-
RTV 3140	360	244	243	0.48	341	232	231	40.9	27.8	27.7
EN-7 (16 hr. at 175°F)	586	355	284	0.49	570	346	277	34.2	20.7	16.6
L-100 (75°F Cure)	-	-	-	0.45	-	-	-	-	-	-
L-100 (4 hr at 160°F)	706	409	383	0.47	653	378	354	118	68.0	63.7
L-100 (3 hr. at 200°F)	-	406	-	0.47	-	375	-	-	67.5	-
Proseal 727	909	550	500	0.477	856	517	471	118	71.3	65.0
Chemseal 3100	700	500	500	0.45	615	439	439	184	132	132

*Metric conversions: 1 ksi = 1000 psi = 6.89 MPa,
 $^{\circ}\text{C} = \frac{5}{9} (^{\circ}\text{F} - 32)$

**Poisson's ratio in compression

Figure 3.4-2. Temperature Dependence of GMB-Epoxy Mechanical Properties

transition temperature of this material are between 170 and 180°F (76.6 and 82.2°C). The coefficient of thermal expansion also doubles above 165°F (74°C).

3.4.2 Stress-Strain Curves

The need for using an apparent modulus of elasticity to determine actual stresses which fall on the stress-strain curve of the material was discussed in Section 1.2.2. Stress-strain curves suitable for this type of analysis appear in Figures 3.4-3 through 3.4-10. These curves should be used to obtain the apparent modulus wherever applicable.

In Figures 3.4-3 and 3.4-4, two curves are displayed, and they represent different material specifications. A study of these figures should emphasize the caution with which the moduli of the various materials should be used. The tensile modulus cannot be used in lieu of the compressive modulus (as practiced in metals design) when designing with polymers. In Figure 3.4-6, the abscissa is labeled Elongation (percent); this was to point out that the tensile strain was estimated from crosshead displacement and not measured directly. The test for Figure 3.4-6 used an ASTM 0.080 by 0.25 inch (2.03 by 6.35 mm) dumbbell specimen gripped as shown. Figure 3.4-9 illustrates the effect of strain rate on a material.

Several cautions must be observed in using stress-strain curves and other mechanical property data. Much of the data presented here was obtained during the development of a material. Minor modifications in composition or cure cycle can change properties significantly. In some cases, the material continues to cure at room conditions. Properties, after 75 days, can be quite different from those after 5 days.

Figure 3.4-3. GMB-Epoxy in Compression

Figure 3.4-4. GMB-Epoxy in Tension

Figure 3.4-5. Adiprene L-100 in Compression

Figure 3.4-6. Adiprene L-100 in Tension

Figure 3.4-7. RTV-3140 in Compression

Figure 3.4-8. RTV-3140 in Tension

Figure 3.4-9. EN-4 in Compression

Figure 3.4-10. EN-4 in Typical Compression

REFERENCES (Section 3.4)

¹Deleted.

²"Thermal, Mechanical, and Electrical Properties of Typical High-Alumina Ceramics," *Materials in Design Engineering*, November 1970.

³"Engineer's Guide to Structural Ceramics," *Materials Engineering*. Volume 72, Number 5, November 1970.

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