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UC-38

PRESSURE AND
TEMPERATURE
GRADIENTS IN
TRANSFER MOLDING
OF ELECTRONIC
ASSEMBLIES

BDX-613-333

April
1970

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ABSTRACT

Pressure-Temperature Gradients in Transfer Molding of Electronic Components

A specially designed strain gage transducer was used to determine pressure exerted upon parts and electronic assemblies during encapsulation by transfer molding. Encapsulation of the transducer and thermocouples in the molding compound were used to determine pressure buildup and decay, simultaneously with the temperature variation during the molding cycle. Pressure buildup during molding was shown to reach a peak value lower than the transfer pressure, while peak cavity temperature approached 100°F above the mold temperature. Results of these experiments and a simplified mathematical analysis provided new capabilities for use in future transfer molding applications.

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SUMMARY

Relatively high molding temperature and transfer pressure required to encapsulate electronic assemblies raise conjecture regarding the ability of fragile, temperature-sensitive components to withstand this process. Although transfer molding has been successfully used for many products, process parameters have been selected largely by experimental methods, with very little being known of the pressures and temperatures to which encapsulated components are subjected during the molding process.

It was known that the temperature of the molding compound within the cavity rises during the molding cycle because of the exothermic reaction of the material. Therefore it appeared obvious that temperature within the mold cavity would increase during the molding process. Also, because of frictional losses in the transfer ram, the compound, and the gate and runner system, it was apparent that the full pressure of the transfer ram would not be applied to the compound within the cavity. Additional pressure buildup from other causes, however, might offset the frictional losses, thus adversely affecting the pressure exerted upon components within the mold.

Temperature rise within the mold cavity can be readily determined by embedding thermocouples in experimental moldings; however, no method of measuring the internal mold pressures that develop during the molding cycle was available prior to this program. For that reason, a series of experiments were conducted to determine the inner cavity pressure gradients that occur during the cycle, and to relate the resulting pressure data to the temperature rise and decay in the molding compound.

A small transducer of the approximate size and configuration of a TO-5 transistor case was developed for encapsulation in a transfer molding, and thermocouples were embedded at various locations within the cavity. These were used with appropriate instrumentation to determine the temperature and pressure gradients at several areas within the compound throughout the molding cycle.

Photoelastic techniques were used to study the stress concentrations upon the surfaces of the transducer when encapsulated in a clear compound, indicating that stresses were essentially equal across the top diaphragm area of the transducer.

Two Fiberite molding compounds, E-9405 and E-3824C, were compared for molding characteristics and molding temperature requirements. It was shown that E-3824C can be satisfactorily molded at temperatures 15 to 20 percent lower than those required to mold E-9405 under similar conditions; however, the E-9405 specimens exhibited slightly superior tensile and compressive strengths.

Mathematical analyses of a simplified component encapsulated in a molding compound were conducted to investigate the feasibility of an analytical method of predetermining the stresses exerted upon a specific component surface, with a hope of eliminating much of the experimentation which is frequently required to establish a satisfactory molding process. These analyses, however, accomplished little more than to illustrate the overwhelming complexities posed by the many variables in even the most simplified case.

Although this investigation did not provide answers to all of the questions which may often arise in transfer molding encapsulation of electronic parts or assemblies, new capabilities were attained or additional knowledge was acquired in the following areas:

1. Temperature rise due to the exothermic reaction of the compound can exceed 100°F above the mold temperature; therefore, the exotherm characteristic of the compound is an important factor in selecting the compound and molding temperature for a specific application.
2. The exothermic reaction of Fiberite E-3824C produces fairly steep slopes of both rise and decay of temperature, with a relatively brief period at peak temperature.
3. The mass of the components and that of the mold itself act as heat sinks which often cause uneven heat distribution in the mold cavity, thereby creating alterations in the flow of the compound and its setting characteristics.
4. Improved heat insulating properties in the component housings result in lower component internal temperatures during molding. Such components, therefore, are less likely to be damaged in molding.
5. The mathematical analyses indicate that a computerized program which would be useful in predetermining the stresses upon a specific component might be possible, despite a complex array of interacting variables.
6. Although selection of Fiberite E-3824C for use in this project does not preclude the possibility that other compounds may prove equally suitable, the work provided additional guidelines for future use of this compound, and much general information which may be adaptable to the use of other compounds.

7. The pattern of buildup and decay of cavity pressure during the Fiberite E-3824C molding cycle was determined with satisfactory accuracy, proving that pressure exerted upon encapsulated component interfaces is appreciably lower than the transfer pressure, and that shrinkage of the compound during the cure portion of the cycle reduces component interface pressure to a value below that exerted during the flow portion of the cycle.
8. The development portion of this project has made a suitable transducer design and techniques for measurement of internal cavity pressure available for possible use in similar future programs.

Section 1

INTRODUCTION

1.1 BACKGROUND

Encapsulation of components, or entire assemblies, is a common practice in the electronic industry, often providing required structural support, protection from physical damage, exclusion of moisture or contaminants, etc. Many Bendix products are encapsulated, and many different compounds and processes are used. Although each encapsulant is selected for compatibility with specific product requirements and all processes are rigidly controlled, comparatively little is actually known regarding the finite values of stresses and strains which the compound may exert upon encapsulated components. Most applications are not highly critical, and many of the less rigid compounds present no major problems; however, some applications require highly rigid compounds and moldings which are essentially free of voids and bubbles. It is these that are of principal concern.

Relatively high molding temperatures and material viscosities are characteristic of most transfer molding compounds. Since these semifluid materials are forced into mold cavities by pressure, appreciable amounts of force may be applied to the components within the mold. Temperature required to produce sufficient flow and cure of the material becomes an important factor when temperature-sensitive components are to be encapsulated. Pressure becomes highly important when the components or assemblies to be encapsulated are of a fragile nature, or when electrical parameters are pressure-sensitive. Moreover, transfer molding compounds are normally subject to shrinkage as they 'set' and become rigid, thereby altering the pressures exerted upon embedded components. In addition, changes in dimensions due to the coefficient of thermal expansion of the compound are reflected as stress changes upon components within a molded part.

Patterns of stresses and strains that are exerted upon an encapsulated assembly are often further complicated by the frequently complex contour of the assembly itself, which sometimes tends to restrict the flow of the compound into given areas of the mold cavity.

Excessively rapid setting of the compound will also restrict the free-flow filling of the cavity. Therefore, the curing characteristic of the molding material is another important factor in the overall transfer molding process.

Because of complex combinations of the above mentioned variables, the design of a mold for a specific molded part may be critical, and size and placement of the gate and the vents may be highly important considerations.

The brief outline of molding parameters above illustrates the general complexities of the problems which are frequently encountered in defining and evaluating the many inter-acting variables and unknown quantities involved in encapsulation of a specific electronic part or assembly. These complexities indicate a need for a practical method of determining the collective effects of all interacting parameters in a quantitative form which is directly applicable to specific design problems and product requirements. Such a method is not presently available; therefore, solutions to many critical product requirements frequently depend largely upon testing of experimental designs.

Electronic assemblies have been successfully encapsulated by transfer molding methods at the Bendix, Kansas City facility for some time, with no defects which were directly traceable to the encapsulation process. However, failures have in some instances created conjecture in regard to transfer pressure, curing temperature, shrinkage pressure, etc. as possible underlying causes. Although these contentions have not been sustained by tests, neither have they been disproven. Therefore, an investigation was initiated to acquire additional information, particularly that related to internal mold pressures and temperatures during the transfer molding cycle.

1.2 OBJECTIVES

Objectives of this program were:

1. Determine the pressures developed inside of the mold cavity during the transfer molding cycle and assess the effective pressure and stress exerted upon an electronic component when encapsulated by this method.
2. Determine the possible contribution of the transfer ram pressure to the total effective pressure and stress upon an encapsulated component during transfer of the compound to the mold and to the final pressure after the compound is cured.
3. Establish internal mold temperature curves for a transfer molding cycle.
4. Determine the effects of the thermal coefficient of the compound upon component stresses.

5. Compare various prospective transfer molding compounds to determine those with lowest temperature and pressure molding requirements and best physical properties.
6. Study the feasibility of mathematical analyses in selection of materials and determination of optimum transfer molding process parameters through analytical evaluation of stresses and strains upon a component embedded within a compound.

1.3 ORGANIZATION AND REPORTING

Due to dissimilarities of the objectives of the work and differences in the techniques and facilities required to achieve them, the work of the investigation was divided into several categories, following the order of the objectives listed on the preceding page.

Categories 1 through 4

Responsibility for the work of categories 1 through 4 was assigned to F. H. Lewis, who was also to assume responsibility for coordination of the various phases of the work and preparation of this final development report. Assistance was to be provided by the Bendix Test Laboratory and other Bendix facilities, as required.

Category 5

Comparison and evaluation of transfer molding compounds was assigned to F. W. Dennen, who prepared a final report on this phase of the study. This work is not related in full to this report, but is briefly summarized in Section 4.

Category 6

The analytical work of the program was divided into two parts.

Part 1

The first analytical phase of the study, pertaining to mathematical determination of stresses upon a simplified case encapsulated component, was performed by M. C. Collins, who prepared a report titled, "Generalized Theoretical Stress Analysis," a digest of which is designated Appendix B, Part 1 to this report.

Part 2

Messrs. F. W. LaMar, J. S. Beshoner, J. I. Amershek, and J. R. Milkey performed the second analytical portion of the work and prepared a report, "Thermal Stress Analysis of a Typical Electronic Component Buried in a Potting Compound." This report, a digest of which is identified as Appendix B, Part 2, illustrates the extreme complexities of an analytical approach to determination of molding parameters, even in the most oversimplified procedure is also included.

Section 2

EXPERIMENTAL STUDY OF INTERNAL MOLD PRESSURE

2.1 GENERAL

In normal transfer molding processes, a molding compound is preheated and is forced by a transfer ram, through a runner and gate system, into the cavity of a preheated mold. Because of losses due to friction of the transfer ram, molecular friction within the compound, and resistance of the gate and runner system to the flow of the semiliquid compound, transfer ram pressure is not a comprehensive indication of the actual pressure developed within any part of the mold. The pressure developed within the mold cavity is always appreciably lower than that applied by the transfer ram; however, pressures at various locations within the mold may be altered by the curing characteristics of the molding compound.

As curing progresses, a chemical cross-linkage between the components of the molding compound causes a slight reduction in the volume of the compound within the cavity, thus reducing the effective pressure upon the encapsulated part or assembly. The chemical reaction which occurs during the curing interval also produces an increase in temperature, thereby causing the temperature on the compound in the mold cavity to rise above the preheat temperatures of both the compound and the mold. Although the usual positive coefficient of thermal expansion of the compound momentarily produces an increase in volume of the compound, which might tend to counteract the reduction in pressure due to shrinkage, the temperature rise is of short duration and, therefore, does not affect the final dimensions of the molding. The behavior of temperatures and pressures that develop within a mold cavity during a typical transfer molding cycle were studied in the course of this investigation, and are discussed and illustrated in a following section of this report.

2.2 MEASUREMENT OF INTERNAL MOLD PRESSURE

Since transfer ram pressure is not directly indicative of the pressure to which a component is subjected during the molding cycle, the investigation required a direct method of measuring the pressure developed upon the interface of a component embedded within the compound. This could apparently be accomplished only by use of a pressure transducer which could be embedded within the compound, replacing the component to be evaluated. Moreover, if the result of such a procedure were to be meaningful, the transducer would have to be of the same size and configuration as the component which it represents.

For the sake of simplicity, a transistor can was chosen as the component for analysis. Two sizes of special transducers were developed. The smaller of these, corresponding to the size of a TO-5 transistor can, is shown in Figure 1. Since the top of the can was considered the most pressure-sensitive area of the transistor, the diaphragm of the transducer was placed in that position. Although the flange of the transistor unquestionably alters the distribution of stresses along the bottom and sides of the can, it was reasoned that the stress pattern along the top would not be appreciably affected. Therefore, the flange was omitted from the transducer design.

2.2.1 The Transducer Design

Since a preliminary search revealed that no suitable transducers were commercially available, it was necessary to design and fabricate special transducers to the requirements of this study, as follows:

1. A pressure range of 0 to 3000 psi.
2. Capability of withstanding temperatures of at least 350 degrees Fahrenheit.
3. Dimensions to simulate those of a standard TO-5 transistor can.
4. Recoverable from the encapsulant, for repeated use.

The first, and largest, of a number of experimental transducers produced had a one-inch outside diameter, 0.187-inch height, an active diaphragm diameter of 0.50-inch, and a diaphragm thickness of 0.018. The diaphragm of this, as well as of all subsequently fabricated transducers, was nickel-brazed to a machined housing. A strain gage was cemented to the inside surface of the diaphragm, the strain gage leads were threaded through small holes in the housing, and a back cover-plate was attached.

Use of a special-design diaphragm-type strain gage enabled ultimate reduction of transducer dimensions to 0.50-inch outside diameter and 0.10-inch height, with an active diaphragm diameter of 0.220-inch. These dimensions are approximately the same as those of a TO-5 transistor case. The smaller transducers were fabricated in the same manner as the larger ones previously described. The transducer design is shown in Figure 1, with the dimensions given being those of the final small-size type. Although these transducers were fabricated and tested with two different diaphragm thicknesses, 0.006 and 0.012-inch, the thinner diaphragm proved to be the most sensitive.

Figure 1. Cross-Sectional Drawing of the Special Pressure Transducer

2.2.2 Photoelastic Analysis of Stresses on the Transducer

Representative large and small transducers were encapsulated in transparent epoxy for photoelastic stress analysis to determine distribution of stresses and uniformity of loading on the surfaces of the diaphragms. The small transducer was cast in Epoxylite 555, whereas the large one was cast in a similar unidentified clear plastic material.

The photoelastic stress analysis technique utilizes principles of prismatic defraction and light polarization to produce visible stress lines in the compound. The color photographs in Figures 2 and 3 show the stress lines surrounding the two transducers. The small irregularity in the fringe pattern along the diaphragm side in Figure 2 (the large transducer) is due to a bubble in the epoxy. Since the small transducer was used exclusively during the remainder of the investigation, data for the large transducer is not given.

P-48808

Figure 2. Photoelastic Analysis of the First Experimental Transducer

P-48809

Figure 3. Photoelastic Analysis of the Final Transducer

Stresses in the epoxy surrounding the small transducer, as determined by photoelastic analysis, are listed below.

<u>LOCATION</u>	<u>STRESSES IN EPOXY</u>	
	<u>Parallel to Surface</u> (psi tension)	<u>Perpendicular to Surface</u> (psi compression)
Diaphragm	60	1430
Bottom	1070	2900
Left Side	60	1500
Right Side	430	1140

These values were reasonably constant over each surface, with the corners having a stress concentration factor of approximately 1.5.

The equations used to separate the two principal stresses in the plane of the slice (as shown in Figures 2 and 3) are:

$$\sigma_1 = \frac{f}{h} \frac{\cos \theta}{\cos^2 \theta} (N_\theta - N_o \cos \theta)$$

$$\sigma_2 = \frac{f}{h} \frac{1}{\sin^2 \theta} (N_\theta \cos \theta - N_o)$$

Where:

N_θ = Number of fringes at oblique incidence

N_o = Number of fringes at normal incidence

θ = Angle of oblique incidence = 30 degrees

h = Thickness of slice = 0.228-inch

f = Fringe constant = 78.5 psi/fringe/inch

Data for the small transducer is listed below.

Location	N_o	N_θ
Diaphragm	4.31	3.78
Bottom	5.33	3.72
Left Side	4.56	4.00
Right Side	4.56	4.31

2.2.3 Transducer Calibration

The transducers were placed in a pressure chamber and exposed to carefully measured nitrogen gas pressures for calibration. Strain gage output was monitored through several temperature cycles to stabilize the materials and determine the effects of temperature upon the transducer response. Outputs were then noted at several pressure levels and a response curve was plotted for each transducer. A typical response curve for one of the small (0.22-inch diaphragm) transducers is shown in Figure 4.

Figure 4. A Typical Transducer Calibration Chart

2.3 DETERMINATION OF TRANSISTOR CAN DEFLECTION AND PERMANENT SET

Since effects of pressure might be expected to first become evident as a deformation of thin-wall components, a transistor can was selected to provide a typical example of this type of component. The top of the transistor can was considered most likely to deform first; therefore, measurement of mold cavity pressure and comparison to static pressures required to damage the component appeared to be the most logical approach.

Nitrogen gas was used in an experimental setup to determine the pressures necessary to deform the top of the transistor can and to establish criteria for measurement of the effects of pressure in later tests. The results of the initial tests on the top of a TO-5 transistor case are tabulated below and shown graphically in Figure 5.

Transistor Can Deflection and Permanent Set Versus Pressure Applied		
<u>PSI</u>	<u>Pressure Applied Deflection in Inches</u>	<u>Pressure Removed Permanent Set in Inches</u>
100	0.00025	0.0
200	0.00075	0.0
300	0.00125	0.0
400	0.00175	0.00025
500	0.00275	0.00075
600	0.00950	0.00950

The above data indicates that this particular component (a TO-5 transistor) will not deform permanently under normally used transfer ram pressures alone. Therefore, the simulated transistor case transducer was considered suitable for the molding pressure study.

2.4 GATE SIZE AND MOLDING TEMPERATURE

Since gate size and the molding temperature are significant factors in the transfer molding process, a special test mold was designed to determine the effects of gate size and molding temperature upon material flow. The concept was to extrude compound from a direct-filled cavity, using various

Figure 5. Transistor Can Deflection and Permanent Set Versus Applied Pressure

extrusion gate, or escape port, sizes; then to weigh the mass of compound which was extruded from each gate at various molding temperatures. This test mold is shown in Figure 6.

A charge weight of 100 grams of Fiberite E-3824C was used, with ram pressure being held constant. A graph of the results obtained with two of the gate sizes is shown in Figure 7. These curves show that a difference in weight of extruded material over the temperature range of 220° to 280°F was approximately 20 grams for the 0.020-inch gate, whereas the 0.010 gate produced a difference of 50 grams; however, a much larger volume of material was extruded through the larger gate. The difference in volume of material through the two gate sizes was greatest at low molding temperatures and decreased at higher temperatures. The curves also indicate that 250° to 260°F is the ideal molding temperature of this material, since the greatest volume of material can be transferred within this temperature range. It is also shown that the large gate is less critical in regard to molding temperature, thus allowing use of lower temperatures when encapsulating temperature-sensitive components.

2.5 DETERMINATION ON INTERNAL MOLD TEMPERATURES

A temperature rise, attributable to the effects of surface and molecular friction due to flow of the compound under pressure and to the exothermic reaction of the compound during the interval of solidification, is expected in all thermosetting compounds. These temperatures may not be equal in all parts of the mold cavity, but are often substantially higher than the preheat temperature of either the compound or the mold. Determination of actual temperature levels attained requires a means of continuously monitoring temperatures inside of the mold cavity at various locations within the compound during the entire time interval of the molding cycle.

2.5.1 Initial Internal Temperature Measurements

A special mold, similar to those normally used for encapsulation of assembly modules, was fabricated for exclusive use throughout this study; thus eliminating a possible variable which might be introduced through use of different molds.

A thermocouple was mounted in the mold cavity in such position that only the air temperature inside of the mold would be indicated prior to transfer of the compound. The epoxy was then transferred into the cavity, and temperature was monitored continuously throughout a programmed 10-minute molding cycle.

A. PLAN VIEW OF MOLD GATES

B. CROSS-SECTION OF TEST MOLD

Figure 6. The Test Mold

(Designed for use in determination of the effects of different gate sizes)

Figure 7. Effects of Gate Size and Molding Temperature Upon the Compound

2.5.2 Additional Experiments

This series of experiments was designed to determine the maximum temperature to which a component is subjected during the mold cycle. The following molding parameters were maintained constant throughout the series:

Material	Fiberite E-3824C
Clamp Pressure	600 psi
Ram Pressure (Force on material)	300 psi
Charge Weight	100 grams
Platen Temperature	250°F
Mold Temperature*	240°F
Cycle Time (Actual molding)	10 minutes

*When the mold reached the specified temperature, the transfer pocket was charged with material and the molding cycle was started.

The mold used was a three-plate mold with a thermocouple attached to the top plate for monitoring the mold temperature. The cavity size was approximately 3-1/2 inches long by 1-1/4 inches wide by 3/4-inch deep. Provision was made for locating a circuit board 1/16 inch above the bottom surface of the cavity. Grooves were machined in the parting line surface of the mold to accommodate the thermocouple lead wires.

The first of this series of molding experiments was conducted with only the thermocouple (no components) mounted in the mold cavity. Temperature variation at the center of the cavity during the 10-minute molding cycle is shown in Figure 8. A temperature rise of approximately 90°F was recorded at the point of peak exothermic reaction of the compound, at seven minutes after the start of the cycle.

The second of these experiments was performed with the mold cavity filled with scrap parts to roughly simulate the conditions which exist when encapsulating a module. The resulting graph, Figure 9, shows the peak exotherm to be approximately 40°F lower than that in the preceding experiment. This is believed to have been due to the heatsinking action of the parts, the reduced volume of the compound in the cavity, and greater heat conduction of the parts as compared to that of the compound.

Additional experiments, the results of which are shown in Figures 10 through 14, were conducted to determine the temperature rise at different points in the mold and differences in temperatures developed between two specific points. Placement of thermocouples and parts in each instance is shown by the sketch in the corresponding figure.

Figure 8. Temperature Rise at Center of Cavity - No Components

Figure 9. Temperature Rise in Cavity Filled With Random Parts

Figure 10. Temperature Rise at Two Points in Trainer Module (With Circuit Board, Two Resistors, and Transformer)

Figure 11. Temperature Rise Inside and Outside of Cavity

Figure 12. Comparison of Temperatures at Top and Bottom of Encapsulated Transformer

Figure 13. Comparison of Temperatures - Inside of Transformer to that of Mold Surface

Figure 14. Comparison of Transformer Temperatures - Inside to Outside Surface

In Figure 10, two thermocouples were mounted in the cavity. One was placed between two resistors mounted toward one end of the circuit board; the other was placed between the circuit board and a transformer which was mounted upon the other end of the board. At the start of the molding cycle, both thermocouples indicated temperatures of 15 to 20 degrees F below that of the mold. The peak temperature rise at the point between the resistors was approximately 50 degrees above the mold temperature, while that below the transformer was only about 30 degrees.

Figure 11 shows a comparison between the temperature rise at a point adjacent to the transformer and that of the top surface of the mold. The thermocouple at the transformer location indicated a peak temperature rise of 50 degrees, while the rise in mold temperatures was only about 15 degrees above the initial mold temperature. A part of the rise in mold temperature can be attributed to the platen temperature, which was higher than that of the mold at the start of the cycle.

Figure 12 compares the temperature rise at the top of the transformer to that between the transformer and the circuit board. Both thermocouples were placed approximately 1/16 inch from the transformer surfaces. Both temperature curves are essentially equal in form, with only a slightly higher rise at the bottom location (approximately 40 degrees above the initial mold temperature).

Figure 13 illustrates the difference in temperature rise at the center of the transformer was approximately 10 degrees lower than the temperature of the mold at the start of the cycle, rising to a maximum value approximately 10 degrees above the mold temperature rise at intervals during the cycle, and approaching the temperature gradient of the mold at completion of the cycle. This behavior is apparently due to the thermal conductivity characteristics of the transformer; however, the dip below the mold temperature at one point of the curve cannot be readily explained.

The last of this series of experiments, shown in Figure 14, compares the temperature rise inside the transformer to that of the compound adjacent to the outer surface of the transformer at the side toward the center of the mold. All components except the transformer and the circuit board were removed from the cavity to prevent possible restriction to flow of the compound. The temperature rise inside the transformer was approximately 50 degrees; whereas, that in the previous experiment (when other parts were removed from the board) was only about 40 degrees. The peak temperature rise on the outside of the transformer was approximately 65 degrees, compared to a more gradual rise of 20 degrees in the preceding experiment. The differences in the results of these last two experiments are apparently related to heatsinking and compound flow pattern changes created by additional components on the circuit board.

2.5.3 Conclusions Derived from the Temperature Experiments

1. The exothermic reaction of the compound is an important factor which must be considered in matching the compound to the characteristics of components which are to be encapsulated. The exothermic temperature may be more than 100 degrees F above the temperature required to mold a specific thermosetting compound.
2. The components, and the mass of the mold itself, act as heat-sinks to dissipate a portion of the heat generated by the compound, thus resulting in uneven distribution of temperature within the cavity during the molding process.
3. Better heat insulating properties in component housings result in lower component internal temperatures during molding.
4. The exothermic reaction of the material used (Fiberite E-3824C) displays fairly steep slopes on both rise and decay of temperature, with a relatively short time period at peak temperature.

2.6 INVESTIGATION OF MOLD CAVITY PRESSURE GRADIENTS

Although the behavior of temperature during the transfer molding cycle was established by the preceding experiments, pressure gradients and the final pressures exerted upon encapsulated components had not been determined at this time. Therefore, the following experiment was designed to determine the pressure buildup and decay in the mold cavity during the molding cycle and to compare the resulting pressure curve to that of the temperature-rise which occurs simultaneously during the cycle, as shown in Figure 15.

Simultaneous monitoring of both pressure and temperature throughout the molding cycle was accomplished by enclosing the previously described pressure transducer and a thermocouple in the center area of the mold and performing a 10-minute molding cycle. Fiberite E-3824C molding compound was used, with a platen temperature of 250°F, a mold temperature of 240°F, and a transfer ram pressure of 300 psi.

The pressures and temperatures recorded throughout the molding cycle are shown in Figure 15. The time intervals designated 1 through 5, which comprise slightly more than the first six minutes of the molding cycle, may be explained as follows:

Figure 15. Plots of Cavity Pressure and Material Temperature During the Molding Cycle

Interval 1

The first interval marks the initial one minute duration of the cycle. During this time the temperature of the compound is being heated to 240°F in the transfer pocket of the press. Temperature within the cavity is 240°F (the mold temperature) and the cavity is at atmospheric pressure (zero reference). Some material flow occurs during this interval, but the cavity has not filled sufficiently to apply pressure upon the transducer.

Interval 2

The second interval, from one to approximately 1-3/4 minutes, begins as material starts to flow into the cavity. As flow starts, pressure begins to increase very slowly -- with no increase in temperature. The curve exhibits a sharp toe at approximately 1-1/2 minutes into the cycle, followed by a rapid rise in pressure. Pressure begins to level off near the end of this interval, at which time the cavity has approached a completely filled condition.

Interval 3

During the early portion of the third interval, the pressure reaches a maximum value of approximately 250 psi, then levels off and begins a slow decline. Because of the exotherm of the compound as it begins to solidify, temperature begins to rise during this interval.

Interval 4

During this interval (from three to four minutes into the cycle) curing continues at a more rapid rate, accompanied by a rise in temperature. Pressure in the cavity during this interval continues to decline because of chemical shrinkage which continues after the gate and the vents are frozen off.

Interval 5

During the interval of about 3-1/2 minutes ending at approximately 6-1/4 minutes into the cycle, exothermic reaction accelerates, resulting in a further rise of temperature and a faster rate of shrinkage, with a corresponding reduction of internal pressure.

At the end of Interval 5, the exothermic reaction is essentially completed; therefore, temperature begins to decline slowly. However, since shrinkage continues for a time, internal cavity pressure continues to decrease to a time of about 7-1/2 minutes into the cycle. At this time, a small, sudden rise in pressure was observed. This pressure discontinuity appears to have occurred because of the sudden conversion of the plastic to a rigid state, although the cause is not fully known.

Section 3

SUMMARY OF POTTING COMPOUND STUDY

(Category Number 5)

Although several molding compounds were considered for evaluation in this program, many were obviously too 'stiff' for the present process and sufficient information was not available on several of the others to provide a tentative indication of suitability. Therefore, this phase of the study was essentially narrowed to a comparison of two Fiberite Corporation materials, E-9405 and E3824C.

The E-9405 material was designed for molding at approximately 300°F and a pressure of 300 psi, whereas, E-3824C was designed for processing at 250°F and 250 psi. Because of these lower temperature and pressure requirements, E-3824C appeared to be a desirable substitute for the previously used E-9405 compound. Also, the otherwise similar characteristics of the two materials made them readily adaptable to a direct comparison of physical properties and processing parameters.

Since the work required for a comprehensive treatment of this phase of the study was rather lengthy and complex, only the general findings and conclusions are summarized in this report.

3.1 PRINCIPAL PHYSICAL PROPERTIES

<u>Property</u>	<u>E-9405</u>	<u>E-3824C</u>
Tensile Strength (psi)	8 to 10,000	7 to 9,000
Tensile Modulus (psi x 10 ⁶)	0.8	0.75
Flexural Strength (psi)	13,500	13,000
Flexural Modulus (psi x 10 ⁶)	1.39	1.3 to 1.6
Impact Strength (foot-pounds)	0.31	0.35
Deflection Temperature		
Under Load (degrees F)	270	310
Coefficient of Linear Thermal		
Expansion (in/in/°F x 10 ⁻⁶)	12 to 16	12 to 16
Specific Gravity	1.599	1.606

The above values were derived from non-postcured specimens. The E-9405 material was molded at 300°F and 300 psi (ram pressure); the E-3824C at 250°F and 250 psi. The E-9405 is characterized by slower, less complete thermosetting during the molding process; therefore, postcuring has a somewhat greater effect upon E-9405 than upon the faster curing E-3824C.

3.2 MOLDABILITY

Since E-9405 cures at a slower rate than E-3824C, making it appreciably less difficult to fill the cavity, transfer speed and gate size can be varied through greater limits without encountering premature gel. Therefore, molding parameters are somewhat less critical for the E-9405 material. Because of more rapid thermosetting and higher exotherm in the E-3824C, slower transfer feed tends to cause premature gate 'freeze-off' and incomplete filling of the cavity. These problems, however, can be overcome by more critical control of the molding process, to take advantage of the lower temperature and pressure molding requirements of the E-3824C. This is particularly desirable when encapsulating fragile, temperature-sensitive components.

3.3 EFFECTS OF PROCESS VARIABLES

Both of the materials tend to produce higher, more consistent strength properties at lower molding pressures. This tendency can be reduced at a given pressure by reducing the transfer speed. Gel time then becomes the principal limiting factor in the process.

Either material can be successfully processed at temperature levels somewhat above or below the values used in these molding experiments. The physical properties of the molded compounds, within limits, tend to follow curves which are apparently related to the product of molding temperature and pressure; therefore, temperature may be reduced and pressure increased within suitable limits when encapsulating temperature-sensitive components. Conversely, higher temperature and lower pressure may be used when pressure is the principal consideration.

Section 4

SUMMARY OF THE ANALYTICAL PROCEDURE

(Category Number 6)

(See Appendix B, Parts 1 and 2 for complete analyses)

Because of the many limitations involved in the direct measurement of stresses and strains upon electronic components by encapsulation in a potting compound, it becomes very difficult, perhaps even impossible, to determine all of the conditions to which such components are subjected, both during encapsulation and after the compound is cured. This is especially true in transfer molding of rigid compounds. Experimental methods, therefore, frequently result in establishing only a 'go' or 'no-go' situation, with very little actually being known about the values and distributions of stresses imposed upon the encapsulated components and circuitry. This problem is made even more complex when an entire assembly, composed of many components in a densely arranged configuration, is transfer molded in a rigid compound.

A mathematical approach which would make it possible to predict the conditions imposed upon a specific design by a stipulated molding process would therefore be very helpful, if such a method could be devised. Unfortunately, complex component and assembly configurations, together with many variables and variations in material properties, complicate the task of mathematical analysis to such a degree that the many thousands of terms and cross-products may extend beyond the capacity of the largest computer to say nothing of the vast number of man-hours required to program them. Although it may seem evident that such an approach is not practical, it seems logical to pursue the matter to a sufficient extent to develop concrete proof of the validity of this assumption. Despite these seemingly overwhelming complexities, however, it may prove practical to utilize a computerized analytical approach, if suitable simplifying assumptions are made.

Stating the problem concisely, exact values of all process variables, molding parameters, and material properties must be mathematically stated in equations which will relate the pressures, temperatures, stresses, and the cross-products thereof which are exerted upon any component by a given molding process. Known data must include the following:

- a. The geometries of both the components and the epoxies.
- b. The physical and mechanical properties of all materials and components involved.

- c. The permanent effects of temperature-rise upon temperature-sensitive components.
- d. The permanent effects of molding pressure and stresses upon pressure-sensitive or mechanically fragile components.
- e. The nature and values of external loads (whether steady state, continuously changing, or pulsed).
- f. Specific locations selected for study. Since it is usually possible to predict the most critical areas, this constitutes a simplifying step.

Solutions to equations which meet all of the requirements stipulated above can be expected to provide the following information:

- a. Stress and strain values at any point on the interface of the component selected for analysis.
- b. The maximum temperature rise at selected points or areas during the molding process.
- c. Determination of process parameters and material properties required to meet product specifications.

The mathematical procedures that were used, together with a proposal for a computerized stress analysis procedure, are shown in Appendix B, Parts 1 and 2. Explanatory illustrations and examples of the calculations required for the various analyses are included in Addenda A through H to this appendix. Addendum I lists references from which much of the information and data which was used in the analyses were obtained. The analyses of both Part 1 and Part 2 of Appendix B were based upon a simple closed cylinder (in this instance, a simplified transistor can), and certain simplifying assumptions were made.

Although these analyses illustrate the many complexities involved (even in this over-simplified case), they also indicate that a computerized program could be developed, which would yield a reasonably accurate analysis of the stresses to which a selected component would be subjected during transfer-mold encapsulation, curing, and subsequent thermal cycling.

Section 5

COMMENTS AND CONCLUSIONS

Temperature and pressure behaviors within the mold during the molding cycle, although not fully conclusive in application to a specific encapsulation problem, was clearly shown by the experiments conducted during this investigation. Although there are many variables and cross-products thereof involved in the transfer molding process, and despite the use of only the two Fiberite materials in this study, the results obtained generally verified theoretically derived expectations in regard to patterns of pressure and temperature-rise within the mold.

Selection of the Fiberite materials for this study does not preclude the possibility that other compounds are equally suitable for transfer molding encapsulation of pressure and/or temperature-sensitive electrical assemblies, or that processing requirements of other compounds may vary somewhat from the molding parameters used in this study. However, the processes developed for the Fiberite compounds provide guidelines which should prove useful in establishing process levels for other compounds in this class of materials.

Pressure remains applied by the press during the entire molding cycle; therefore, the transfer ram continues to move as long as the compound remains sufficiently fluid to flow through the gate and mold vents, even though the cavity is completely filled. The rate of this flow decreases as curing progresses, finally ceasing when the compound solidifies sufficiently to freeze off the vents and the gate. At that time the force of the transfer ram becomes completely isolated from the cavity, and release of cavity pressure through the vents is blocked. After the flow is thus stopped, the pressure inside of the cavity is wholly dependent upon the shrinkage pressure developed by the compound as curing progresses. Therefore, it is evident that transfer pressure is not additive to the shrinkage pressure of the compound and does not influence the final pressures exerted upon a component encapsulated within the compound.

APPENDIX A

TEST LABORATORY REPORTS

SUMMARIES OF LABORATORY TEST REPORTS

128994, 161144, 161146 and 161147

These tests were designed to monitor the performance of "A" and "E" modules under pressure and temperature conditions well beyond the ranges normally encountered in transfer molding encapsulation processes.

The modules were electrically connected to the power source and monitoring equipment as shown in Figure 1. Both "A" and "E" modules were tested in the same manner except for changes in load terminations, as noted.

To simulate pressures encountered in transfer molding, each module was submerged in oil which was contained in a glass beaker and placed in a pressure vessel. The module was connected to the test setup by means of sealed feed-through terminals in the wall of the pressure vessel.

Figure 1

Test Setup for E Module

(A Module setup similar, except as noted)

LTR 128994

Unpotted E Modules

Serial Numbers BBN-2605-C7 and BBN-2648-C7

Effects of Pressure Upon E Module Output

TEST PROCEDURE

Each module was connected in the test setup as shown in Figure 1. A regulated continuously adjustable zero to 30-volt d-c power supply was used to power the circuitry. The output functions were monitored by means of a dual-beam oscilloscope. An accurate 1000 ohms per volt voltmeter was also connected to Terminal No. 1 of the module and to the common negative return. Dry nitrogen was used to pressurize the vessel.

Without applying gas pressure to the vessel, the d-c supply voltage was slowly increased from zero until an internally triggered sweep appeared on the oscilloscope, thus indicating a step function output from the module. The supply voltage was then interrupted and reapplied several times to ensure stable operation. This procedure was repeated at vessel pressures of 100, 300, and 1000 psig. Vessel pressure was increased at a rate of 100 psig per minute and was decreased upon completion of each test at three to five times that rate.

TEST RESULTS

BBN-2605-C7

No indication of malfunction was observed at any time during testing of this module.

BBN-2648-C7

Indications were normal in every respect, except that an unexplained triggering occurred once at 150 psig. Since both inputs of the scope were connected and the sweep was set to trigger internally, this spurious trace could have been caused by a transient at either input due to radiation from an external source or a transient in the a-c power line. This result was not observed again during the repeated tests. The module suffered no physical damage.

LTR 161144

A Module

Serial Numbers A2904 and A2977 (Unpotted Modules)

Effect of Pressure Upon a Module Performance

TEST PROCEDURE

Each module was connected in a test setup similar to that used for the E Module (Figure 1), except that Terminal 6 of the module was loaded by a diode (anode to Terminal 6 and cathode to Terminal 2) shunted by a 620-ohm 5% resistor. A fixed d-c potential of 19.7 to 20.3 volts was applied, positive to Terminal 1 and negative to Terminal 2 and common return.

The frequency of the A Module was monitored throughout the tests by use of an Erie, Model 100T counter which was connected to Terminal 6 of the module. A Hewlett Packard, Model 201C oscillator was used as a reference signal to set up a Lissajous pattern on the oscilloscope screen. The output frequency was checked carefully at specific pressures of zero, 100, 300, and 1000 psig.

TEST RESULTS

No excessive frequency drift was encountered when testing either of the two A Module sample units. The Lissajous pattern displayed no sudden changes, and the ten-second frequency count was at all times within plus or minus six counts on each unit. There was no evidence of physical damage to the modules at the end of the tests.

LTR 161146

A Module, Part Number 191779 (Unpotted Modules)

Serial Numbers 2904C7, 3100C7, 3051C7, 3000C7, 3102C7, and 3001C7

TEST PROCEDURES

The test setup and procedures used were essentially the same as those described in LTR 161144, with the addition of a Lindberg oven to apply and control temperature of the module and pressurizing canister.

The oven and the pressurizing canister were preheated to 240°F before starting the tests (time, t_0). After one minute at 240°F, pressure was increased to 250 psig during a period of 20 seconds. The temperature was then further increased to a maximum temperature (t_m) of 285°F at six to seven minutes after t_0 , during which time interval the pressure was decreased to conform to the temperature/time profile specified in the test instructions. When the maximum temperature was reached, the canister temperature was reduced to 270°F in an interval of three minutes by removing the canister assembly from the oven.

TEST RESULTS

The measured conditions under which each specimen was tested and the frequency of the modules before test (at room temperature), at the start of the test (240°F), and the highest frequency reached during the test period are tabulated below:

TEST CONDITIONS

Specimen Serial Number	Maximum Temperature (t_m °F)	Time t_0 to T_m (minutes)	Temperature at End of Test t_0 - 10 minutes (°F)
2904C7	285	6.4	273
3100C7	284	7.6	272
3051C7	284	8.0	273
3000C7	285	7.6	270
3102C7	282	7.0	270
3001C7	286	6.7	274

OUTPUT FREQUENCY VARIATIONS DURING TESTING

Specimen Serial Number	Frequency		
	Before Test (Ambient, Hz)	Start of Test (Time t_0 , Hz)	During Test (F_{max} , Hz)
2904C7	No Test	No Test	1004.8
3100C7	999.4	1005.6	1006.8
3051C7	1000.2	1004.2	1004.2*
3000C7	999.5	1002.2	1006.5
3102C7	999.9	1002.4	1005.6
3001C7	1000.3	1004.1	1007.6

*Shortly after the start of the test the count on this module decreased to 1003.8, but gradually increased to 1004.1 at maximum temperature.

A large sudden change in frequency of 2904C7 occurred at time t_0 plus 8.25 minutes due to a loosened wire at Terminal 2 of the module. The count returned to normal when the wire was resoldered. No other significant changes occurred, and no physical damage to any of the modules resulted from testing.

The results of these tests indicate that neither temperature nor pressure, through the ranges applied in these tests, have any significant effect upon the frequency or functions of the A Module.

LTR 161147

E Module, Part Number 207018 (Unpotted Modules)

Serial Numbers 2648C7, 2531B7, 2434B7, and 2480B7

Effects of Temperature and Pressure Upon E Module Functions

TEST PROCEDURE

Each module was tested separately in the test setup described in LTR 128994 and shown in Figure 1, with the addition of a Lindberg oven which was used to heat the module and the pressure canister.

The pressure canister in which the module was mounted was preheated to a temperature of 240°F before starting the tests (time t_0). After 1.5 minutes at 240°F, the pressure was increased to 250 psig during an interval of 20 seconds. The temperature was then further increased, reaching a maximum value (T_m) at t_0 plus six to seven minutes. During this time the pressure was decreased as specified in the test instructions. When the maximum temperature (285°F) was reached, the canister was removed from the oven and allowed to cool to 270°F in a period of three minutes.

Performance of the E Module was monitored as described in LTR 128994 during the interval of each test.

TEST RESULTS

The measured temperature conditions under which each test was conducted are tabulated below:

TEMPERATURE TEST CONDITIONS

Specimen Serial Number	Maximum Temperature (°F)	Time to T_m (minutes)	At End of Test (t_0 - 10 minutes) (°F)
2648C7	286	6.0	270
2531B7	286	5.3	270
2534B7	286	6.2	272
2480B7	281	5.8	271

A spurious response of about 10 seconds duration, believed to be due to oven control relay chatter, occurred in the first test of Serial 2648C7.

The oven control unit was therefore disconnected during the remaining tests. No other departures from normal operation were encountered in testing the remainder of the modules.

These tests indicated that the ranges of temperature and pressure employed have no degrading effect upon the performance of E Modules.

APPENDIX B

An Analytical Approach (In Two Parts)

Part 1

A Generalized Theoretical Stress Analysis

By

M. C. Collins

Part 2

Thermal Stress Analysis of a Typical Electronic Component Buried in a Potting Compound

By

F. W. LaMar

J. S. Beshoner

J. I. Amershek

J. R. Mulkey

APPENDIX B - PART 1

A GENERALIZED THEORETICAL STRESS ANALYSIS

THE GENERAL PROBLEM

A need for an analytical method of predicting localized pressures and stresses and their effects upon encapsulated components arises from the fact that sufficient knowledge of pressures and stresses at work in different potted parts, either during molding or during later thermal cycling, is not available. However, some capability does exist for setting up instrumentation to measure a few of the variables in a particular potting situation, although considerable amounts of time and money are required. Much of this outlay could be avoided if reliable equations that could give us the same information for any prescribed situation, within certain defined limits, were available.

Of course, the basic physical and mechanical properties of the materials involved would have to be known, but this is far easier information to obtain than to actually implement and measure a complete potting process. Considerable difficulty arises, however, because of the complexity of the general type of equation required.

An attempt will not be made to write such equations, because to do so would require a large amount of time and research, and is beyond the present manpower capability. Because of their vast complexity, it may be unreasonable to expect that such equations could be written at all. However, some of the requirements that must be met in order to start writing them are shown. A very simplified example to illustrate something of the magnitude of such an undertaking is also shown.

To state the problem more precisely, it is necessary to set up equations that relate the pressure and stresses on any component within any given molded part to the following:

1. The geometries of both components and epoxies.
2. The physical and mechanical properties of all materials involved.
3. The temperature distributions on any interface or at any layer of interest.
4. The present state of the process and the history of the entire part.
5. Coupling interactions due to other nearby components.

6. External loads, whether steady-state, continuously changing, or pulsed.
7. The particular locations selected for study. (This is a simplifying step because we can predict ahead of time which areas are most likely to cause trouble).

These equations must be capable of determining the principal stresses in a part at any point to be studied, or in any uniform area. The equations must also be able to use these principal stress values to predict occurrence of localized bending or buckling and, if so, to determine the magnitude. They must also be able to predict plastic flow in any area, at least through an iterative process. Effects of wall-thickness, corners, depth, and interfaces should also be calculable from the equations. They should be in the form of a computer program which, at any key stage of a solution for a particular case will enable an engineer to direct the computer to a choice of next steps.

A SIMPLIFIED ANALYSIS

In order to set up such a set of equations, the problem must be broken down into logical groupings according to the actual requirements.

There are four different process stages in which the equations should be set up, each probably leading to certain boundary conditions for another. It is not necessary that the conditions of each stage be independent of the others. These four stages can be listed as pressure states, as follows.

Stage 1.

There are three different processes that are currently used at BKC, which could account for this first stage -- casting, transfer molding, and foam expansion molding. Other processes could be used here in other types of programs. In casting and foam expansion molding there is no real effect caused by filling the mold with the uncured material; therefore, the only thing to consider would be the amount of material deposited and the condition of the components and mold as boundary conditions for the second stage. Transfer molding, however, is a far different type of process. The epoxy is a visco-elastic fluid, and is injected into a preheated mold under controlled ram pressure. This process depends on the rate of flow of the epoxy being slow as compared to its cure rate. Flow condition is maintained until a significant degree of cure has been reached. This brings about a smooth transition into the second stage.

Stage 2.

There are differences in this stage in the different processes. In casting and foam-expansion molding, the materials

apply continually increasing pressure to the components within the mold during curing of the casting epoxy. This pressure is also due to material shrinkage around the components, and to its expansion to fill and pressurize the entire cavity. The transfer mold materials do just the opposite, i.e., they contract in such a manner as to gradually relieve the pressures on the components caused by the ram pressure from stage 1. This pressure can fall off to a zero final pressure.

Stage 3.

The different materials behave in an essentially similar fashion in this stage. Since they are cured, the potting materials can apply pressures on the components due to thermal cycling. Because all of the potting materials have high coefficients of thermal expansion, as compared to the coefficients of the components, there can be a large effect caused by differential thermal contraction when parts are taken from a high to a low temperature. This is usually the most significant stage.

Stage 4.

The pressures caused by part encasement materials, by localized loading on the epoxy, or by thermal or ultrasonic shocks can become other sources of difficulty in the parts. Such pressures may not be steady-state conditions, but they may cause a lack of repeatability in certain areas of the part, due to highly localized concentrations at some time during other processing. This stage may occur concurrently with some of the other stages.

Before beginning an attempt to write the equations, there are several parameters of each situation and some properties of the materials that must be known or defined. Some of the most likely parameters are shown below, with the stages in which they should be defined also listed:

<u>Parameter</u>	<u>Used in Stages</u>
Ram Pressure* (Transfer Molding)	1
Mold Temperature Distribution*	1, 2, 3
Viscosity of Uncured Epoxy*	1
Residual Pressure Gradient*	2, 3
Exothermic Effects in Epoxy*	2
Shear Effects at Interfaces*	2, 3, 4
Visco-elastic Modulus of Epoxy*	1, 2
Component Temperature Distribution*	1, 2, 3, 4
Cured-Epoxy Temperature Distribution*	3, 4
Residual Deformations*	3, 4
Special Boundary Conditions Caused by History of Non-Repeatable Effects	3, 4

<u>Parameter (Continued)</u>	<u>Used in Stages</u>
Residual Pressure on Epoxy from Encapsulation Material	4
External Temperature Gradients*	4
Local Loading Functions*	4
Shock Functions (Ultrasonic or Thermal)*	4
Geometry of the Components	1, 2, 3, 4
Geometry of the Epoxy	1, 2, 3, 4
Geometry of Mold	1, 2, 3, 4

*This indicates that these parameters could also be time-dependent functions depending on conditions of the particular problem.

In addition to these parameters, there are the material properties that must be taken into account. These properties could vary according to the stage; however, they obviously apply generally to the solid materials in any of the stages. These properties are listed below, together with the form they would have to take in order to be generally useful:

<u>Material Property</u>	<u>Form</u>
Coefficient of Linear Thermal Expansion (c,e)**	Polynomial*
Elastic Modulus (c,e)**	Polynomial
Compressive Modulus (c,e)**	Polynomial
Flexural Modulus (c,e)**	Polynomial
Poisson's Ratio (c,e)**	Constant
Yield Strength (c,e)**	Constant
Ultimate Tensile Strength (c)	Constant
Density (c,e)	Constant
Thermal Conductivity (c,e)**	Constant
Specific Heat (c,e)	Constant

*The polynomials will take the form $a_0 + a_1T + a_2T^2 \dots$ where the number of terms depends on each separate material and the maximum temperature range to be experienced. Probably no case should require accuracy above a fourth order term for any property.

**These are directional effects in most epoxies and can, therefore, have different values for each component equation.

The actual variables of the problem must be defined at this point. Some of these variables have been implied in previous statements. Each of these must be carefully defined and related to the conditions of the problem, as shown on the following page.

<u>Variable</u>	<u>Function Of</u>
Position	Coordinate System
Time	Starting Points
Temperature	Position, Time and Material Properties
Pressure	Temperature, Time, Position and Material Properties
Principal Stresses	Pressure, Temperature, Position, Time and Material Properties
Strain	Principal Stresses, Pressure, Temperature Position, Time and Material Properties

In setting up the equation, care must be taken in the selection of a coordinate system. Different geometries lead to different best selections. The possible geometrics that would be encountered would suggest cartesian, cylindrical, and spherical coordinates for various cases. An equation set up in generalized orthogonal coordinates would possibly be the best. If any one system had to be selected, cylindrical coordinates would probably have the widest application.

Another item that should be considered before setting up the equations is the existence of some interaction effects in a real process. These can lead to errors and inaccuracies. Some of these are listed, as follows:

1. Components made of more than one material, (eg. a transistor can with an epoxy sealed base). This could lead to high stress on the bonded interfaces. It could also lead to irregular expansion characteristics, causing excessive localized stresses.
2. Lead wires and other connections to components would be another source of stress concentrations.
3. Spacing of components, if small, could lead to a coupling interaction between stress parts.

Such interaction effects would have to be well defined, and significance-tests would be necessary to determine which, if any, of these could be neglected in a given case.

The final item needed to set up the equations is a list of the requirements of what must be gotten from the equations. Some of these have been mentioned previously i.e., principal stresses, bending, buckling, plastic flow, depth effects, etc.

Besides these, there are certain other prescriptions that must be satisfied. All functions must be clearly delineated, together with methods of solution, forms of solution analysis, and criteria for judging the significance of cross terms. Also, there must be some inherent way to evaluate the accuracy of the calculations.

Obviously, the magnitude and complexity of this problem is vast. It would help in evaluating the magnitude of it if an example was available to serve as a guide. As a result, only a somewhat simplified derivation of a general equation for just one small aspect of the entire problem can be presented here, based on some rather straight-forward assumptions.

It is assumed that a right, cylindrical, uniform, solid-metal component is potted in uniform epoxy in a right cylindrically shaped mold. By making the can of solid metal, the effects of wall thickness, corner, and bending effects are of little consequence. By assuming a right cylinder, the geometry can be simplified. Further simplification can be accomplished by considering the part only in the condition where the epoxy is completely cured and only thermal cycling effects exist, with no outside forces of any kind and no other components. It is also assumed that there are no residual stresses or pressures on can or epoxy, and that the shear force between the two is zero. It is assumed, however, that in the initial condition, there is zero separation between can and epoxy at all points, and that all of the properties of both metal and epoxy are isotropic. This will eliminate consideration of bothersome cross-term interactions in the properties. Only the circular end of the component, as an interface with the epoxy, is considered.

This results in the situation shown in Figure 1, on the following page, where $2L_e$ and $2L_c$ are the lengths of the cylinders of the epoxy and the component respectively:

R_e and R_c are the respective radii

α_e and α_c are coefficients of linear thermal expansion

E_e is the elastic modulus of the epoxy

E_c is the compressive modulus of the components

T_0 is the initial temperature of the system after it has had time to achieve equilibrium

Figure 1

A Representative Component Encapsulated in an Epoxy Potting Compound

L_e , L_c , R_e and R_c are constant. The values of α_e , α_c , E_e and E_c are functions of temperature in the following forms:

$$\begin{aligned}\alpha_e &= a_0 + a_1T + a_2T^2 + a_3T^3 \\ \alpha_c &= b_0 + b_1T + b_2T^2 \\ E_e &= c_0 + c_1T + c_2T^2 + c_3T^3 \\ E_c &= d_0 + d_1T + d_2T^2\end{aligned}\tag{1}$$

Where the a's, b's, c's and d's are constants and T is the temperature.

The temperature, however, has some sort of distribution that can be described as a polynomial function of position. Although the temperature is considered to be in equilibrium, it is not a good assumption that it is uniform. It is assumed, however, that the temperature distribution is a simple function of the coordinate variables so that there are no cross terms that signify interaction effects. It is further assumed that the process can be adequately described in terms of the end states only, so that a differential equation that can be solved by straight forward integration can be set up.

With these assumptions, the only cross terms that will arise will be through multiplication products and will not indicate any intrinsic effects. As a result, such cross terms will only affect total magnitude values. Distribution is also limited to third order terms of the coordinates. Therefore:

$$T = \tau + \mu_1 r + \mu_2 r^2 + \mu_3 r^3 + v_1(r\theta) + v_2(r\theta)^2 + v_3(r\theta)^3 + \pi_1 z + \pi_2 z^2 + \pi_3 z^3 \quad (2)$$

where

$\tau = \mu_0 + v_0 + \pi_0$ is a reference temperature

μ 's, v 's and π 's are coefficients

and

r, θ, z are cylindrical coordinates

r = radial (distance)
 z = axial (distance)
 θ = tangential angle (radians)

If temperature undergoes a change from an initial equilibrium state T_0 to a cooler final equilibrium state T_1 , a differential thermal contraction will occur ($\alpha_e < \alpha_c$). This is assured in virtually every epoxy metal interface and in some foam-metal contractions. This differential contraction is caused by the fact that interface adhesion tends to restrict interface contraction only to position 1 in Figure 2; whereas, the epoxy, because of its higher coefficient of thermal expansion, tends to contract to position 2. As a result, the can goes into compression and the epoxy into tension, and a compromise is reached at position 3. If uniformity is assumed, it is necessary to consider differential contraction in only one direction. The axial (z) direction is selected in this instance.

Figure 2

Differential Thermal Contraction Stresses When Cooling)

(Positions noted are discussed in the text)

Figure 3

Cross-Sectional View of Stresses Shown in Figure 2

(Mathematical expressions are noted)

At position 0 the contraction is zero.

At position 1 the contraction in the components due to the thermal expansion is $L_c \alpha_c \Delta T$.

At position 2 the contraction in the epoxy due to the thermal expansion is $L_e \alpha_e \Delta T$.

Complete symmetry has been assumed, so that one-half of the effect is seen at one end of the can, and the other half occurs at the other end. In order for the two materials to come to equilibrium at position 3, a differential strain force comes into play. This force is the force of the epoxy on the can (ΔF_c) and the force of the can on the epoxy (ΔF_e). Equilibrium demands that these two forces be equal and opposite, i.e., $\Delta F_c = -\Delta F_e$. Since these forces are related to strains, the distances through which they act are given by

$$L_c (\Delta F_c / E_c A_c) \quad (3)$$

and

$$L_e (\Delta F_e / E_e A_e)$$

where

$$A_c = \pi R_c^2$$

and

$$A_e = \pi R_e^2$$

It is assumed here that the overall dimensions of the epoxy cylinder are short enough to be uniformly affected by this strain-ing force. Otherwise, it would be necessary to write L_e and A_e as functions dependent on the distance away from the edge of the component. They would also have to take into account fall-off with distance from the edges of the can. Such functions would probably be something like decreasing second order exponentials of distance. Such terms would involve significant cross-product interactions, which this example is designed to avoid.

Assuming continuity and regularity, an equation can be written in differential form for the total strain at position 3 of Figure 2.

$$L_c \alpha_c dT + L_c (dF_c / A_c E_c) = L_e \alpha_e dT + L_e (dF_e / A_e E_e) \quad (4)$$

It is known that

$$dF_e = -dF_c = -dF$$

Therefore,

$$L_c \alpha_c dT + L_c (dF / A_c E_c) = L_e \alpha_e dT - L_e (dF / A_e E_e) \quad (5)$$

This is a differential equation with variables that can be separated, because all functions except force are written in terms of temperature. This equation reduces to the form:

$$dF = \frac{A_c A_e E_c E_e (L_e \alpha_e - L_c \alpha_c)}{L_c A_e E_e + L_e A_c E_c} dT \quad (6)$$

Substituting the values from Equation 1 ,

$$dF = \frac{A_c A_e [c_0 + c_1 T + c_2 T^2 + c_3 T^3] [d_0 + d_1 T + d_2 T^2] [L_e [a_0 + a_1 T + a_2 T^2 + a_3 T^3] - L_c [b_0 + b_1 T + b_2 T^2]]}{L_c A_e [c_0 + c_1 T + c_2 T^2 + c_3 T^3] + L_e A_c [d_0 + d_1 T + d_2 T^2]} dT \quad (7)$$

Carrying out the indicated operations and grouping like terms, Equation (6) reduces to the form of Equation 8.

$$dF = \frac{M_0 + M_1 T + M_2 T^2 + \dots + M_3 T^3}{N_0 + N_1 T + N_2 T^2 + N_3 T^3} dT \quad (8)$$

Where M and N are the appropriate products of L_c , L_e , A_c , A_e , a's, b's, c's and d's.

Mathematically, $1/N_0 + N_1 T + N_2 T^2 + N_3 T^3$, can be closely approximated by a real polynomial of the form $Q_0 + Q_1 T + Q_2 T^2 + \dots$, where the Q's are functions of the N's. This is a convergent series. For this type of problem, and in the temperature ranges that we expected, the terms above the fourth power can probably be neglected. Thus obtaining:

$$dF = (M_0 + M_1 T + \dots + M_3 T^3)(Q_0 + Q_1 T + \dots + Q_4 T^4) dT \quad (9)$$

If the approximation is accurate enough, this can be restated in the form of Equation 10.

$$dF = (R_0 + R_1 T + R_2 T^2 + \dots + R_{12} T^{12}) dT \quad (10)$$

where the R's are functions of M's and Q's.

Integrating Equation 10 between appropriate limits results in Equation 11.

$$\int_{F_0}^{F_z} dF = \int_{T_{z0}}^{T_{z1}} \sum_{i=0}^{12} R_i T^i dT \quad (11)$$

or

$$F_z = \sum_{i=0}^{12} R_i' (T_{z_1}^i - T_{z_0}^i) \quad (12)$$

Where T_z , the temperature on the end surface, and the R_i' have been properly adjusted.

As shown in Equation 2 ,

$$T = \tau + \mu_1 r + \mu_2 r^2 + \mu_3 r^3 + v_1(r\theta) + v_2(r\theta)^2 + v_3(r\theta)^3 + \pi_1 z + \pi_2 z^2 + \pi_3 z^3 \quad (2)$$

For T_z , z is a constant, therefore,

$$T_z = \tau' + \mu_1 r + \mu_2 r^2 + \mu_3 r^3 + v_1(r\theta) + v_2(r\theta)^2 + v_3(r\theta)^3 \quad (13)$$

Where τ' has been adjusted to include the constant z terms.

Substituting Equation 13 into Equation 12, a solution evolves for the original expression, as follows:

$$F_z = \sum_{i=0}^{12} R_i' \left\{ \sum_{j,k=0}^3 \sum_{l,m=0}^n \sum_{q,p=s}^t \left[s_{jk}^{lm} \left(r_{lm}^{qp} \right)_1^j \left(\theta_{lm}^{qp} \right)_1^k - s_{jk}^{'lm} \left(r_{lm}^{qp} \right)_0^j \left(\theta_{lm}^{qp} \right)_0^k \right] \right\} \quad (14)$$

where:

the set j, k represents the appropriate powers and coefficient array numbers,

the set l, m represents particular points on the surface,

the set q, p represents the area of the surface enclosing the points l, m , and

the numbers s through t designate particular increments of area on the surface.

The limits of $s \rightarrow 0$, $t \rightarrow \infty$, and $n \rightarrow \infty$ of Equation 14 represent the total force on the end of the cylinder caused by this one dimensional consideration.

Since the force over a particular area is known (i.e., pressure), and since the situation is completely regular, standard equations can be used to calculate the principal stresses caused by this pressure at that location. If there should be

enough metal compression to cause plastic flow, then, an iteration can be repeated until a satisfactory point to convergence is reached.

It is now possible to make some sort of assessment as to the magnitude of the problem that has been described. The solution, equation (13), to the present example is of such proportions that it is quite unwieldy. The $r\theta$ products, alone, reach the 78th degree, and involve about 1,000 different terms (eg. $r^{40}\theta^{38}$) for each boundary condition $(r\theta)_1$ and $(r\theta)_0$. The number of each of these variables eg., $(r\theta)_{lm}$, needed to study a given area could be quite large, depending on the significance of slight changes of position or the fluctuation of properties due to temperature. The next step is to consider the complexity of the coefficients S_{jk}^m , each of which can involve as many products as the highest degree of the $r\theta$ products (78) in any term, and as many terms as the expansion ($\approx 1,000$). Besides these, there are the R_i terms which include all of the original polynomial coefficients (a's, b's, c's, and d's) in a cross term array to a high degree of complexity.

The net result of all of this is that it would take a vast computer program just to study this very simplified case. This brings up another problem, namely, that the actual size of a computer is a limiting factor on the capability to solve and analyze such equations, even if it were possible to get them set up. Such a problem involves many thousands of bits of information that a computer would have to handle.

Comparing the complexity of this example to the dimensions necessary to satisfy all of the conditions of such a problem, it can be seen that a number of man-years of work would be required, without any guarantee, at this stage, that a satisfactory set of equations could be derived.

APPENDIX B - PART 2

THERMAL STRESS ANALYSIS OF A TYPICAL ELECTRONIC COMPONENT BURIED IN A POTTING COMPOUND

1. INTRODUCTION

1.1 General Goal

The Test Laboratory was asked to develop an analytical approach which could be used to predict the compatibility of the materials in a potted electronic component assembly. The stress analysis was to predict the maximum stresses encountered at any location and indicate whether the detail electronic parts would function satisfactorily (mechanically) with the potting compound used.

A "virgin model" was used, meaning that the potted assembly would be assumed to consist of materials essentially in the "as-received" condition, without regard to the effects of later processing and any residual stresses which might result therefrom.

The request was couched in the above manner because results obtained would be used to make decisions on an existing component assembly at Bendix. Since the decisions might possibly involve changes in material or processing, it was desirable to keep the analysis fairly general.

It was also believed that the stress analysis would be a fair representation of the particular assembly.

1.2 PRODUCTION UNIT INVESTIGATION

Due to interest in the assembly, it was decided to survey the assembly and to use it as a pilot in developing a representative "stress model". The investigation consisted of (1) talks with engineers and technicians involved in design, development, processing, and testing of the unit, (2) a demonstration of the transfer molding process for individual modules, and (3) examination of an assembly (cutaway view), as well as individual modules, both potted and unpotted.

The following observations are presented as results of this investigation, since they are considered pertinent to the development of the force system existing in the assembly due to the potting, curing, differential thermal contraction, etc. It is important that the stress analysis, as developed in this report, be examined in the light of the following remarks to ensure agreement in the minds of stress analyst, writer, and reader when interpreting the stress analysis. The remarks should be borne in mind when reading the assumptions, discussion, results, conclusions, and recommendations.

1.2.1 FAILED PARTS

No record of mechanical failure in parts which had been produced by mechanical processing could be found. Some transistor cans had been crushed under an injection mold pressure of 600 psi; however, manufactured parts are molded at a mold reservoir pressure of only 300 psi. Manufactured unpotted assemblies have withstood an external hydraulic pressure of 1,000 psi on the shell.

1.2.2 ELECTRONIC COMPONENTS

Components have external shells of many different sizes and shapes. The shells are made of a variety of materials such as nickel, glass, tantalum, etc. Some shells consist of several pieces loosely assembled and most have leads passing through them.

1.2.3 POTTING COMPOUND USED IN THE MODULES

This material is a glass-reinforced, mineral-filled epoxy (E-3824C Fiberite Corp. Spec.). It is a grainy powder of various sizes and shapes. The largest grains are approximately 0.125 by 0.125 by 0.063-inch.

Flowability, or plasticity, in standard spiral-flow-control molds appeared excellent, i.e., material is very soft at molding temperatures. However, the comments on flowability, (Paragraph 1.2.5, Manufacturing Process) further explain the behavior of the compound.

1.2.4 THE ELECTRONIC MODULE

This subassembly consists of a large number of electronic components which are fastened to a pinboard. Two very thin layers of rubber (RTV), vulcanized at room temperature, are sprayed and baked on all components before potting. The purpose of this coating is to eliminate local stress concentrations on the shell housings. The module is then potted by transfer molding. A row of pins, for leads, protrudes through one end.

1.2.5 MANUFACTURING PROCESSES

This is a transfer molding process. A cross-section of the mold cavity is shown in Figure 1, Addendum A. A pinboard with the RTV coated components is positioned in the mold cavity, a row of pins are assembled in the mold, and the mold is then closed. After the mold reaches 240°F, the potting compound powder is poured into the reservoir. The powder quickly melts, and is then immediately transferred to the mold cavity.

Filling of the mold cavity takes approximately 1-1/2 to 2 minutes. Figure 2 describes how the potting compound enters the

mold. Leaving the gate, it enters the cavity as a thick flat sheet through a slit, 0.010-inch high which is as wide as the cavity. The compound then begins to fill the mold at the gate end. The space between the pin board and cavity wall is very narrow, therefore, this space occasionally does not fill. Air vents underneath the board at each end allow air to escape as the mold fills.

Several partially molded modules were examined. The filling was deliberately stopped before molding was completed. Figure 3A and 3B are top and side views respectively. Note that the flow front (the free surface, or boundary) has a steep slope, indicating a high resistance to gravity and, hence, a high viscosity.

A laboratory study (Reference 1*) shows that curing raises the temperature, after molding, to about 300°F. Pressure in the cavity after the mold has filled is approximately 250 psi.

1.3 Problem Definition

This type of problem frequently occurs in engineering designs and involves the interaction of two parts, the potting compound and an electronic component. However, the many different components and their orientation in respect to each other would require considerable time for a complete analysis.

The project completion date was set for thirty days after start, which obviously eliminates a complete analysis of assembly.

Study of an individual module indicates that parts, although they are apparently not failing mechanically, could have a high stress in several localized areas on detail components. The high stress can be caused by temperature, pressure, and electrical effects, as well as stress concentration. All of these effects act on each component in the module.

A detailed examination of these thousands of interactions is not practical, regardless of the amount of effort expended; therefore, the best approach to the problem is to select a representative local area of failure in the module, and to examine it in detail. However, even in this approach, several thousand individual calculations are required. Therefore, a number of simplifying assumptions must be made to increase the chances of attaining a representative analysis of a typical part.

*All references are listed in Addendum I.

2. THE PILOT MODEL

The pilot model is shown in Addendum A, Figure 4. It is a transistor can, Bendix Part Number, which has been altered by the assumptions shown below to produce a realistic design for analysis (see Figure 5). It is composed of a thin cylindrical shell with two thin end plates, or disks. The actual thickness, diameter, and length of the can has been used, but the flange has been eliminated, as shown in Figure 5A. It is surrounded completely and is in intimate contact with a potting compound.

2.1 Assumptions for the Pilot Model

The following simplifying assumptions were selected to produce a pilot model which would give an indication of the general level of reliability of all parts in the module:

- (1) The transistor can is a typical electronic component which will "feel" stress levels in the module which are essentially the same as the highest stress levels applied to any component.
- (2) The greatest increment of stress contributing to the stress level is the differential thermal expansion between the transistor can and the potting compound, and occurs when the module cools after curing.
- (3) The molding pressure, 200 psi (Reference 1), has negligible effect, because it has been removed before the thermal stresses are fully developed.
- (4) The model part will see typical stress concentrations as high as any encountered in any diode, transistor, capacitor, etc., since the layers of RTV rubber will partially dampen out extremely high stress rises. The sharp corners on the model part will effectively represent a typical stress concentration. Stress concentrations considered here are general ones which cause strain gradients, rather than shorter fatigue life.
- (5) Shrinkage of the potting compound due to curing is insignificant. If the stress analysis shows a high level of strain (plastic strain), a decision will be made at that time on whether or not it should be considered.
- (6) Normally encountered electrical effects are of small magnitude. These are assumed to be manifested in the form of a small quantity of heat. Except for sudden electrical surges, heating would actually tend to lower the thermal stresses.
- (7) The pilot model is of one piece homogeneous construction, with welded, brazed, or soldered joints.

(8) There are no holes in the shell, no reinforcing flanges, or other stress raisers.

(9) The flow pressure exerted on the can by the viscous liquid is negligible. This assumes the time required to fill the mold is due to the relatively large mold cavity and the narrow 0.010 inch filling slot for the molding compound.

2.2 Material Constants

The following materials are currently used in the production modules and were also used in the pilot model:

(1) TRANSISTOR CAN

Material, A-Nickel. (1)

Elastic modulus at 70°F, E	30 x 10 ⁶ (Reference 4).
Coefficient of thermal expansion at 70°F,	7.4 x 10 ⁻⁶ (in/in/°F).
Poisson's Ratio, elastic	0.3
Poisson's (2) Ratio, plastic	0.5
Stress strain curve for plastic analysis	(See Figures 13, 14, 15; Addenda D and E).
Stress strain curve for hardness comparison	(See Figure 9 and Reference 2).
Tensile Strength (hot rolled, annealed)	

Low Temperature	Ultimate	0.2% Yield Strength
70°F, longitudinal	72,700 (psi)	23,100 (psi)
70°F, transverse	72,400 (psi)	23,600 (psi)
-100°F longitudinal	81,400 (psi)	25,300 (psi)
-100°F transverse	79,800 (psi)	26,800 (psi)

(Continued on the following page)

(1) All nickel properties unless noted otherwise are from Reference 2. A-Nickel is equivalent to Nickel 200, which is an International Nickel Corporation specification.

(2) Plastic flow assumes constant volume.

High Temperature	Ultimate	0.2% Yield Strength
70°F	67,000 (psi)	21,500 (psi)
200°F	66,500 (psi)	22,300 (psi)
300°F	66,700 (psi)	21,700 (psi)
Compressive Strength (hot rolled)		23,150 (psi)

(2) **POTTING COMPOUND**

(Unless otherwise noted, all properties are from Reference 3)

Material, E-3824 glass fiber filled epoxy

Elastic modulus, E	1.4×10^6 (psi)
Tensile strength	10,500 (psi)
Flexural strength	15,500 (psi)
Molding temperature	300°F
Molding pressure	250 (psi)
Coefficient of thermal expansion (3)	27×10^{-6} (in/in/°C).
Elastic modulus (See Reference 5)	
-45°F	1.59×10^6 (psi)
70°F	1.24×10^6 (psi)
150°F	1.58×10^6 (psi)
190°F	1.28×10^6 (psi)
Flexural modulus (See Reference 5)	
-45°F	1.43×10^6 (psi)
70°F	1.21×10^6 (psi)
200°F	1.11×10^6 (psi)
300°F	1.01×10^6 (psi)
Coefficient of thermal expansion, α , (See Reference 6)	
$\Delta T, ^\circ C$	α , in/in/°C
188 - 138	36.8×10^{-6}
138 - 188	29.3×10^{-6}
88 - 38	28.4×10^{-6}
38 - (-12)	27.8×10^{-6}
-12 - (-62)	21.4×10^{-6}
-62 - (-112)	18.5×10^{-6}

2.3 Assumptions for Properties of Materials

The stress analysis of a thin shelled pilot model part embedded in a potting compound by molding requires considerable data on the variation of material properties with temperature. These are difficult to obtain without considerable lead time.

- (3) For elastic analysis, which assumes material properties are constant:

A number of attempts to obtain such data were unsuccessful. Representatives of the transistor manufacturer (Fairchild) and International Nickel could not help us, although we were assured that the extremely flat slope of the plastic portion of the reconstructed stress-strain curve for A-Nickel (Figure 13) was correct.

The following assumptions for material properties were made:

- (1) The potting compound, E-3824C is isotropic, with all material properties the same in all directions within the material.
- (2) The modulus of elasticity and the coefficient of thermal expansion of the potting compound are constant from -65 to +300°F in the plastic stress analysis.
- (3) The material of the pilot model, A-Nickel, is assumed to have a modulus of elasticity and a coefficient of thermal expansion which are constant from -65 to +300°F.
- (4) Since a stress-strain curve is not available, it will be assumed that the potting compound is elastic enough to deflect in the same manner as the can, although at a different strain rate.
- (5) The elastic modulus of the potting compound is constant and the elastic limit of the compound is high.
- (6) The potting compound contains no voids.
- (7) The A-Nickel material is isotropic.

3. ANALYTICAL PROCEDURES

3.1 General Assumptions

To properly evaluate the existing stress condition of an actual transistor can of a specific material embedded in a specific potting compound requires a stress analysis which predicts the actual stress level, rather than the maximum level which can be expected. The designer usually allows for all discrepancies in data or formulations to fall on one side of a specified limiting condition. The stress analyst cannot do this when predicting a stress in an existing part. To obtain a realistic prediction of this type, the analyst delicately balances the assumptions and variations in constants and material data to "zero out" as many of these effects as possible. It is the intent of all assumptions to accomplish this, thus reducing the amount of calculating required. The following assumptions apply to this analysis:

- (1) There is no bonding of the potting compound to the thin shell of the pilot model.
- (2) Both potting compound and pilot model have no residual stresses in them at the start of curing and molding.
- (3) The part is assumed to be molded and cured at 300°F at a zero stress throughout and that stresses then develop as it cools.
- (4) The end plates tend to act as diaphragms.
- (5) The temperature at any given instant is uniform throughout the pilot model and the potting compound in all analyses.
- (6) The nickel shell is made of hot-rolled, annealed sheet. It is drawn and then completely stress relieved by annealing (see discussion).
- (7) The pin board does not bend.
- (8) The volume of the potting compound is so large that the outer surface produces no edge effect.
- (9) The longitudinal axis of the shell is an axis of symmetry.

(10) The normal assumptions of bending stresses hold in elastic analysis and also where applicable in plastic analysis. (Such as plain sections remain plain after bending, etc.).

(11) The usual assumptions for principal stresses hold unless otherwise stated.

(12) The principal stresses in the disk (even at the edge) are not as large as the maximum principal stresses in the cylinder.

(13) A uniform stress field exists in the potting compound. The pressure is considered to be a force acting perpendicular to the surface at all points. This means that at the corner the biaxial stress field which would normally develop is considered to be uniaxial and perpendicular to the edge (at 45° to the axial of the can).

(14) There are no stress effects on the epoxy surrounding the can from neighboring electronic components.

3.2 The Load Picture

Figure 6, Addendum A, shows differential elements of the end plates and cylinder which make up the pilot model. The cylinder has a pressure load distributed over its entire surface. The external pressure load also transfers a compressive force to the edge of the disk. The end plates are dished by an external pressure load which is transferred to the cylinder as an axial compression.

Figures 7A and 8A, Addendum A, are differential elements taken from the cylinder and dish respectively. Each element is taken from the edge of its parent member. The edges are considered as relatively fixed, even though they have some elasticity. This fixity reverses the bending moments and shear forces near the edge (Figures 7B and 8B). The stress pattern at the edge is considerably more complicated and the stresses are highest there. The arrows and circles represent forces and moments, respectively, acting on the elements.

Figure 9, Addendum A shows the joint arrangement which is assumed in this analysis. Note that this results in some shear planes on the disk, for example: forces 13 and 14 in Figures 7A and 8A) being located slightly inward radially from the edge. The forces and moments developed due to (1) the pressure of the potting compound and (2) the interaction between the edges of the disk and the cylinder are listed below. The numbers correspond to those shown in Figures 7A and 8A.

- (1) A local axial buckling tendency due to Item (2) and (7), below, acting on the cylinder near the ends.
 - (2) A circumferential bending moment on the cylinder due to surface curvature and the axial bending moment of Item (3). This moment is equal to the moment of Item (3) multiplied by Poisson's Ratio. It is due to the geometry of the component.
 - (3) Axial bending moment due to external pressure on the cylinder. This acts on both the cylinder and the disk.
 - (4) Hoop compression forces due to the external pressure on the cylinder.
 - (5) An axial compression, acting on the cylinder, due to the external pressure on the disk.
 - (6) Vertical shear force, interacting between the disk and cylinder, due to the external pressure on the cylinder. This is related to the axial bending in Item (3).
 - (7) Vertical shear forces in the cylinder due to the external pressure on the cylinder. This force is related to Item (2) and is a result of surface curvature.
 - (8) A tangential tensile force in the disk due to bending. The external pressure on the disk generates this force, which is a general membrane stretching caused by curvature of deflection. This is related to Items (18) and (3).
 - (9) A radial⁽⁶⁾ compression due to the external pressure on the cylinder acting on the edge of the disk. This acts on the disk only.
 - (10) Circumferential bending moment due to external pressure on the disk. This is also an effect of Poisson's Ratio and is related to Item (11) by Poisson's Ratio and geometry.
 - (11) Radial bending moment due to external pressure on the disk.
 - (12) Shear at the joint due to external pressure on the disk as a result of the elastic action of the disk. (See Figure 9.) Note: Items (12), (13), (16) and (17) as shown in Figure 7A, actually tend to relieve existing effects and, hence, can be considered as reverse shears and bending moments which are interactions, or edge effects. They are due to the assumption of complete edge
-
- (6) A radial compressive moment causes a radial stress, a circumferential moment causes a circumferential stress.

fixity (rigid joints) and represent the effect of some elasticity in the joints. Normally these are eliminated by assuming a certain fixity (80%, 90%, etc.) of the joint in each stress calculation. The disk is the foundation, which contracts radially to a slight degree. This force is an interaction.

(13) Shear at the joint due to external pressure on the disk. (See Figure 7A.) This is the same type of edge fixity as that of Item (12), except the cylinder is the foundation and contracts axially.

(14) Vertical shear at the joint due to the external pressure on the disk and Poisson's Ratio. This force is related to Item (10) and is an interaction of disk and cylinder.

(15) The external pressure exerted by the potting compound on the part. This is a uniformly distributed load and acts on both disk and cylinder.

(16) Bending moment at the joint due to the elastic action of the cylinder when acting as a fixed foundation. This is derived from Item (13) and affects both disk and cylinder.

(17) Bending moment at the joint due to the elastic action of the disk as it assumes the role of a fixed foundation.

(18) Radial tension due to the external load on the disk as the middle plane of the disk stretches to produce the deflection curve.

(19) Pure shear across the edge of the joint, acting on the cylinder and disk, due to pressure on the cylinder.

(20) Pure shear across the thickness of the disk at the inside surface of the edge of the joint due to the external pressure on the disk.

3.3 The Stress Analyses

To determine the suitability of a part using different shell and potting material combinations, a series of stress analyses should be made, each one relating to particular types of loading and/or material properties. The analysis covers the 20 types of loading listed under the section entitled "The Stress Picture."

3.3.1 BUCKLING ANALYSIS

An analysis for buckling, Item (1) under the description of loads, was made using the normal assumptions for inelastic stability and a formula from Reference 7.

The formula gives a critical buckling load (pressure) on the pilot model (thin shell) for an external lateral and longitudinal pressure on a thin shell with closed ends, as follows:

$$P = \frac{E(t/r)}{\left[1 + \frac{1}{2} \left(\frac{\pi r}{nL}\right)^2\right]} \left\{ \frac{1}{n^2 \left[1 + \left(\frac{nL}{\pi r}\right)^2\right]^2} + \frac{n^2 t^2}{12 r^2 (1 - \mu^2)} \left[1 + \left(\frac{\pi r}{nL}\right)^2\right]^2 \right\} \quad (1)$$

where:

- P = external pressure required to buckle the shell longitudinally (psi).
- E = modulus of elasticity of shell (psi)
- t = thickness of shell (inches)
- r = radius of shell (inches)
- L = length of shell (inches)
- n = number of nodes formed by the tube in buckling

This formula was used on a computer to determine the critical buckling pressure as a function of the number of nodes. (See Table I, Addendum A.) The minimum pressure which will buckle the shell is 97,748 psi, and will cause a buckle having six nodes. While this is a minimum load in the table, it is the upper end of a range of loads. (See discussion below.)

3.3.2 ELASTIC STRESS ANALYSIS

Since there was a possibility of low stresses in the pilot model (see paragraph on failed parts under Historical Background), an elastic analysis was made. It must be made in any event, however, for if plastic flow is encountered, the elastic analysis forms the base for the plastic analysis.

3.3.2.1 Assumptions for the Elastic Stress Analysis

- (1) The pressure exerted at the interface of the two materials is uniformly distributed, but unlike a hydrostatic pressure, falls off if the shell deflects.
- (2) Plane sections remain plane after bending.

(3) Any change in material mechanical properties due to a change in temperature is small. The same is true for thermal properties.

(4) The potting compound surrounding the thin shell has a large volume, so that its outer surface is a free boundary which has no principal stresses in any direction.

3.3.2.2 Explanation of Symbols Used

α_c = coefficient of linear thermal expansion of transistor can, (in/in/°F).

α_e = coefficient of linear thermal expansion of epoxy, (in/in/°C).

E_c = modulus of elasticity of can (psi).

E_e = modulus of elasticity of epoxy (psi).

R_c = outside radius of transistor can (inch).

R_e = inside radius of epoxy (same magnitude as R_c) (inch).

μ_c = Poisson's Ratio of the can.

S_1 = axial (or meridional or longitudinal) stress due to a uniform pressure (psi).

S_2 = hoop (or circumferential) stress due to uniform pressure (psi).

t_c = wall thickness of transistor can (thin shell), (inch).

t_d = thickness of the disk (inch).

$\delta\Delta T_c$ = incremental change in radius due to temperature change of can (inch).

$\delta\Delta T_e$ = incremental change in radius due to temperature change of epoxy (inch).

δp_c = incremental change in radius due to pressure, can (inch).

δp_e = incremental change in radius due to pressure, epoxy (inch).

δ_B = beam deflection due to bending of the shell, (psi).

R_o = outside radius of the epoxy, (inch).

Δt = temperature change from molding (or curing to whatever equilibrium temperature is desired), °F.

P = uniform (but no hydrostatic) pressure on the shell (psi).

S_{d1} = axial discontinuity compressive stress due to the external pressure (Figure 7, Item 5). A prime (') symbol indicates bending.

S_{d2} = hoop discontinuity compressive stress due to the external pressure (Figure 7, Item 4). A prime (') symbol indicates bending and a double (") indicates shear.

D_d = flexural modulus of the disk.

D_c = flexural modulus of the cylinder.

λ_c = flexural rigidity and foundation elasticity characteristic for the cylinder.

λ_d = flexural rigidity and foundation elasticity characteristic for the disk.

V_o = the vertical shear due to discontinuity stresses (pounds per linear inch of circumference).

M_o = bending moment due to discontinuity stresses (inch-pounds per linear inch of circumference).

S_a = maximum combined stress in the axial direction (psi).

S_c = maximum combined stress in the circumferential (hoop) direction (psi).

x = length of beam from one end to section under analysis (psi).

3.3.2.3 Geometric and Materials Constants

$\alpha_e = 27 \times 10^{-6}$ (in/in/°C).

$\alpha_c = 7.4 \times 10^{-6}$ (in/in/°F).

$R_c = 0.1516$ inch.

$R_o = \infty$

$R_e = 0.1516$ inch.

$$t_c = 0.0114 \text{ inch.}$$

$$E_e = 1.4 \times 10^6 \text{ (psi).}$$

$$E_c = 30 \times 10^6 \text{ (psi).}$$

$$\mu_e = 0.5$$

$$\mu_c = 0.3$$

3.3.2.4 Shrinkage Profiles

Figure 10, Addendum A, is a series of cross-section of the pilot model showing shrinkage profiles, etc. Since the stresses in the transistor can are due to shrinkage of the epoxy on cooling, some arbitrary state must be assumed at which point the epoxy just starts to shrink. In this state, there are no stresses existing in the can or epoxy. It is then assumed that the can and epoxy are subjected to a temperature change of 365°F (from 300 to -65°F) and allowed to reach an equilibrium state of stress and temperature.

Since the epoxy has a higher thermal expansion coefficient, it will tend to compress the transistor can to a size smaller than it would assume due to its own thermal shrinkage. Hence, at the equilibrium state, a pressure exists at the interface of the epoxy and the can. This equilibrium pressure has restricted the shrinkage of the epoxy in proximity to the can.

Figure 10 illustrates this action. The largest can profile (Line A) represents the interface between the epoxy and the can in the original stress-free state. After a temperature change of 365°F, the radius of the transistor can would be decreased some small amount ($\delta \Delta t_c$) due to thermal compression, and the can's outside surface would be relocated at the profile labelled "B". In a similar manner, the surface of the epoxy cavity would tend to be relocated at profile "C" by an amount, $\delta \Delta T_e$, since its thermal shrinkage is greater. Since the can cannot shrink this far, it is compressed slightly, but exerts a pressure against the epoxy which pushes the interface out to the equilibrium position of profile "D".

An equilibrium equation can be written, from this figure relating the pressure at the interface (Figure 7A, and 8A, Item 15) to the temperature change which has occurred.

Summing deflections:

$$\delta \Delta T_c + \delta p_c = \delta \Delta T_c - \delta p_c \quad (2)$$

Relating strain to thermal expansion:

$$\begin{aligned}\delta\Delta T_c &= \alpha_c R_c \Delta T; \text{ also:} \\ \delta\Delta T_c &= \alpha_e R_e \Delta T\end{aligned}\tag{3}$$

Relating interface pressure to strain:

$$\delta_{pc} = \frac{R_c}{E_c} \left[\frac{PR_c}{t_c} - \frac{\mu_c PR_c}{2t_c} \right]\tag{4}$$

This formula is from Reference 8, Case 1, page 268. It has been modified by substituting for S_1 and S_2 as follows:

$$S_2 = \frac{PR}{t} \text{ and } S_1 = \frac{PR}{2t}\tag{5}$$

(See Reference 8 for an explanation of these).

$$\delta_{pe} = \frac{PR_e}{E_c} \left[\frac{R_o^2 + R_e^2}{R_o^2 - R_e^2} - \mu_e \left(\frac{R_e^2}{R_o^2 - R_e^2} - 1 \right) \right]\tag{6}$$

This formula is for a thick-walled cylinder (the epoxy) and is derived from Reference 8, Case 29, page 276.

Symbols were switched as follows:

$$\begin{aligned}R_o &= a \\ R_e &= b \\ \delta_{pe} &= \Delta a\end{aligned}\tag{7}$$

Since " R_o " is so large that the epoxy's outer surface is stress free, there is no effect of " R_o " on strain; therefore, from Equation 6:

$$\frac{R_o^2 + R_e^2}{R_o^2 - R_e^2} \text{ must approach one, and} \quad (8)$$

$$\frac{R_e^2}{R_o^2 - R_e^2} \text{ must approach zero. Then:} \quad (9)$$

$$\delta_{pe} = \frac{PR_e}{E_e} (1 + \mu_e) \quad (10)$$

Substituting Equations 4, 10, and 3 into Equation 2 yields:

$$\alpha_c R_c \Delta T + \frac{PR_c^2}{E_c t_c} \left[1 - \frac{\mu_c}{2} \right] = \alpha_e R_e \Delta T - \frac{PR_e}{E_e} (1 + \mu_e) \quad (11)$$

Gathering Terms:

$$\left\{ \frac{R_c^2}{E_c t_c} \left(1 - \frac{\mu_c}{2} \right) + \frac{R_e}{E_e} (1 + \mu_e) \right\} = \alpha_e R_e \Delta T - \alpha_c R_c \Delta T \quad (12)$$

Since $R_c = R_e$ solving for "P" yields:

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T}{\frac{R_c}{E_c t_c} \left(1 - \frac{\mu_c}{2} \right) + \left(\frac{1 + \mu_e}{E_e} \right)} \quad (13)$$

This is the equation derived for the pressure acting at the interface of the epoxy and the transistor can (Figure 7A and 8A, Item 15).

This pressure is the external pressure exerted on the cylindrical surface and ends of the can. It is used to find the magnitudes of the various forces and moments outlined in the section labelled "The Load Picture." These forces and moments are then combined into various formulas to find the stresses they set up in the material. It should be noted that the elastic interactions of the two materials which were described above have been accounted for in the above equation.

This pressure is not a hydraulic pressure, but is used to solve for the hoop compressive stress (Figure 7A, Item 4).

$$S_2 = \frac{PR_c}{t_c} \text{ psi} \quad (14)$$

(Case I, page 268, Reference 8.)

The axial compressive stress (Figure 7A, Item 5) is found by:

$$S_1 = \frac{PR_c}{2t_c} \text{ psi} \quad (15)$$

(Case I, page 268, Reference 8.)

If the walls of the can are relatively thin and there are no abrupt changes in the curvature or the longitudinal slope of the can, the longitudinal and hoop stresses are uniform throughout the can. But in this analysis, the abrupt change in slope at the junction of the wall of the cylindrical shell with the ends (Figure 9) induces local bending and shear on this section and also on those sections adjacent to it. These "discontinuity stresses" are superimposed upon the uniform longitudinal and hoop stresses which occur due to changes in pressure and temperature. Had no abrupt change occurred, the maximum axial stress would be S_1 , (Figure 11, Addendum A) and the maximum hoop stress, S_2 . These two stresses are results of the uniform pressure (Figure 7A, Item 15). Because of the moment and shear force induced by the change in slope at the junction of the can wall and the disk, a new set of stresses must be imposed upon the already existing axial and hoop stresses. (See Reference 8, Chapter 7, pages 31 to 38; Reference 9, page 88; and Reference 10, pages 124 to 134). These discontinuity stresses are as follows:

(1) An axial bending stress due to the moment, M_o , (Figure 9). This stress is due to Items 3, 11, 16 and 17 of Figure 7A. (See Reference 8, Chapter 12, Cases 11 and 24, pages 271, 276 and Reference 9, page 35.)

(2) A hoop stress due to the vertical shear force, V_o , in Figure 9. This stress is due to Items 6, 12 and 13 of Figure 7A. (See Reference 8, Cases 10 and 24.)

(3) A hoop stress due to axial bending moment, M_o , in Figure 9. Items 3, 11, 16 and 17 in Figure 7A cause this stress. (See Reference 8, Case 11, page 271; Case 24, page 276; and Reference 9, pages 35 and 36.)

(4) A hoop bending stress due to the moment. M_o . This stress results from Items 2, 7, 16 and 17 in Figure 7A, by virtue of Poisson's Ratio. See Reference 8, Case 11, page 271, and Case 24, page 276, and Reference 9, pages 31, 32, 35 and 37).

(5) The bending moment and shear forces for discontinuity stresses are calculated as follows from Reference 8, Case 24, page 275.

$$M_o = \frac{\frac{PR_c^3 \lambda_c^2 D_d^2}{4D_d(1+\mu_c)} + \frac{2PR_c^2 \lambda_c^3 E_c t_c D_d}{E_c t_c^2 (1 - \frac{\mu_c}{2}) [E_c t_d + 2R_c D_c \lambda_c^3 (1-\mu_c)]}}{2\lambda_c + \frac{2R_c \lambda_c^2 D_c}{D_d(1+\mu_c)} - \frac{\lambda_c E_c t_d}{E_c t_d + 2D_c \lambda_c^3 R_c (1-\mu_c)}} \quad (16)$$

$$V_o = M_o \left[2\lambda_c + \frac{2R_c \lambda_c^2 D_c}{D_c(1+\mu_c)} \right] - \frac{PR_c^3 \lambda_c^2 D_c}{4D_d(1+\mu_c)} \quad (17)$$

$$D_c = \frac{E_c t_c^3}{12(1-\mu_c^2)} \quad (18)$$

$$\lambda_c = \sqrt[4]{\frac{3(1-\mu_c^2)}{R_c^2 t_c^2}} \quad (19)$$

Since:

$$R_c = R_d$$

$$t_c = t_d$$

$$\mu_c = \text{is for both, cylinder and disk}$$

$$E_c = \text{is for both cylinder and disk}$$

Then:

$$D_c = D_d \text{ and,}$$

$$\lambda_c = \lambda_d$$

Note that λ includes the flexural rigidity of the beam as well as the elasticity of the foundation. (To better understand this, see Reference 9, page 4, 30, 31 and 88).

Since the two symbols λ_c and λ_d are equal, the above formulations for M_o and V_o can be rearranged as follows:

$$M_o = \frac{\frac{PR_c^3 \lambda_c^2}{4(1+\mu_c)} + \frac{2PR_c^2 \lambda_c^3 D_c}{(1 - \frac{\mu_c}{2})[E_c t_c + 2R_c D_c \lambda_c^3 (1-\mu_c)]}}{2\lambda_c + \frac{2R_c \lambda_c^2}{1+\mu_c} - \frac{\lambda_c E_c t_c}{E_c t_c + 2D_c \lambda_c^3 R_c (1-\mu_c)}} \quad (20)$$

$$V_o = M_o \left[2\lambda_c + \frac{2R_c \lambda_c^2}{1+\mu_c} \right] - \frac{\lambda_c E_c t_c}{E_c t_c + 2D_c \lambda_c^3 R_c (1-\mu_c)} \quad (21)$$

Note that this vertical shear is directed radially outward and one result will be a hoop tension at the edge.

Using Equations 20 and 21, the discontinuity equations can now be calculated as follows:

The axial stress due to the bending moment, M_o , Paragraph (1), above:

$$s'_{d1} = \frac{\sigma M_o}{t_c^2} \quad (22)$$

The hoop stress due to the vertical shear force, V_o , Paragraph (2), above:

$$s''_{d2} = \frac{2V_o \lambda_c}{t_c} \quad (23)$$

This is the maximum vertical shear and occurs at the mid-radius of the cylinder.

The hoop stress due to axial bending moment, M_o , Paragraph (3), above:

$$s_{d2} = \frac{2M_o \lambda_c^2 R_c}{t_c} \quad (24)$$

This is the maximum hoop stress and occurs at the joint.

A hoop bending stress due to the bending moment, M_o , Paragraph (4), above:

$$\begin{aligned} S'_{d_2} &= \mu_c S'_1 \\ &= \frac{6\mu_c M_o}{t_c^2} \end{aligned} \quad (25)$$

(Reference 8, Case 11, page 271.)

The above stresses must now be combined into maximum stresses in given directions. The three dimensions of interest are radial, circumferential, and axial. Since a radial stress would be mainly due to sheer, and would be small, only stresses in the circumferential (hoop) and axial directions will be combined.

Combining Equations 15 and 22, above, the maximum stress in the axial direction is:

$$S_a = S'_{d_1} + S_1 \quad (26)$$

Signs of these principal stresses should be ascertained when actual calculations (see sample calculations) are carried out.

The maximum circumferential (hoop) stresses can be found by the following combination:

$$S_c = S_2 + S''_{d_2} + S'_{d_2} \quad (27)$$

from Equations 14, 23 and 24 and 24 with 25. This equation has two values (tension and compression) due to bending.

This concludes the equations developed for the elastic analysis. A set of the mathematical calculations are included in Addendum B.

The above calculations show that all three elastic stresses are very large. An examination of the stress-strain curves for A-Nickel (Figure 12) shows these stresses to be considerably above the elastic limit. Therefore, the effect of stress relief due to bending of the shell wall as a beam should be considered next.

3.3.2.5 Elastic Beam-Deflection Analysis

While the effects of elasticity of the nickel and the epoxy were taken into account in the above analysis, the effects of elastic deflection due to beam-effects (bending) was not. This can be shown to be small by incorporating it (δ_B) into the interface pressure and comparing:

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T - \delta_B}{\frac{R_c}{E_c t_c} (1 - \frac{\mu_e}{2}) + \frac{1 + \mu_e}{E_c}} \quad (28)$$

The deflection as a beam would tend to relieve the interface pressure and its effect would be greatest at the center of the beam (can). The deflection equation is from Reference 10, page 127, Equation 16 and the following calculations need be considered only as they affect the hoop stress and axial stress at the center of the can:

$$\delta_B = \frac{V_o \cos \lambda x - \lambda_c M_o (\cos \lambda x - \sin \lambda x)}{2 \lambda_c^3 D_c e^{\lambda x}} \quad (29)$$

Where $V_o = P_o$; $\lambda_c = B$ in the reference, and $x =$ the beam or can length.

A set of sample calculations is shown in Addendum C. Note that the interface pressure is reduced only 0.3-percent (1,900 psi to 1.890 psi) by considering the bending deflection; for this reason, a plastic analysis must be made.

3.3.2.6 Plastic Stress Analysis

A stress analysis occasionally yields a stress or a strain so high that it is considerably above the 0.2-percent yield strength. (See Figure 12, Addendum A.) In some applications of materials to specific designs, this is considered acceptable. Normally, this is when a very ductile material is being used, creep is absent, and the number of cyclic loads is low (below several hundred cycles).

Stress analysis normally calculate an "elastic stress," since the modulus of elasticity, E , is used in the calculations. This is the slope of the straight line portion of the curve in Figure 12, Addendum A. Note that the stress strain curve departs from this straight line at the proportional limit, which is slightly below the elastic limit, and is considerably below the 0.2-percent yield strength.

A high elastic stress such as that calculated in Addendum B (237,700 psi compressive) will lie at a point "W" on the modulus curve, well above the stress/strain curve of the material (A-Nickel) as shown in Figure 12, Addendum A. Since the material can only exhibit strain (stress being an engineering concept for convenience), the elastic stress analysis must be modified to yield the strain actually obtained in the material.

3.3.2.6.1 Assumptions for Plastic Analysis

(1) Due to the amount of time available, the plastic analysis will be greatly simplified by analyzing the stresses on a section at the end of the cylinder, at the joint of the cylinder and disk. All sections of the can will be assumed to have the same strain-hardened reduced modulus of elasticity as the section being analyzed. (See the discussion for the effects of this).

(2) The apparent-stress/apparent-strain curves shown in Figure 23, Addendum D, and Figure 24 and 25, Addendum E, will be assumed to be true-stress, true-strain curves up to the ultimate stress, since data on the latter are not available.

(3) Stable or uniform plastic flow (uniform elongation) occurs below the elastic limit and the ultimate strength. (See Reference 11, page 371-3).

(a) Local necking does not occur until the ultimate strength is reached.

(b) Only pure tension exists on the specimen's cross-section until local necking starts.

(4) Since (3) holds, Poisson's Ratio, μ_c , is 0.5, because the volume of the material under going plastic flow remains constant.

(5) The stress-strain curves of Figure 23, 24, and 25 are representative of both tensile and compressive curves.

(6) The maximum stress theory holds, and a maximum combined stress can be compared with a point (stress) on the stress-strain curve, Reference 10, pages 372 to 375.

(7) Due to the design of the can, the cylinder can have only three possible section locations for plastic hinges (sections at each end and at the middle). (See Reference 10, pages 354 to 366 and Reference 11, pages 404, 407 and 416, for plastic hinges.)

(8) The end plates, while allowing the joints to act as elastic foundations, reduce the tendency to form a plastic hinge to a negligible amount (at both ends).

(9) The external pressure acting on the thin circular shell produces a type of bending action which generally results in stretching (tension) and tends to prevent the formation of plastic hinges at any location.

(10) Based on assumptions (7) and (8), above, the principal of superposition still holds, even though the stresses are above the elastic limit. However, it is recognized that biaxial stresses are involved.

(11) The axial end loads due to pressure and modulus of elasticity of the compound are low enough to prevent kinematic collapse of the cylinder. (Reference 11, page 406.) The material behaves elastically when the load is released, although the Bauschinger effect (small inelastic strain due to microstresses affecting subcrystals) may be present. (Reference 10, pages 412 to 413.)

(12) The ratio of any stress in the circumferential (hoop) direction to any stress in the longitudinal (axial) direction is constant.

(13) In a region in which plastic flow exists, plastic flow occurs over the entire cross-section of the cylinder.

(14) The modulus of elasticity of the potting compound (epoxy) is assumed constant.

(15) The stresses in the potting compound during plastic flow in the nickel cylinder are assumed to be elastic.

(16) The modulus of elasticity of the can is constant with change in temperature.

3.3.2.6.2 Materials Properties for Plastic Analysis

The material properties used in the elastic analysis will be used for the plastic analysis with the exception of Poisson's Ratio and the modulus of elasticity of the A-Nickel.

Poisson's Ratio, μ_c , for the cylinder is 0.5, since during plastic flow the volume of metal involved is constant until necking occurs. (Reference 13 and 14.)

The modulus of elasticity, E_c , will be replaced by a "modified", or plastic modulus. (The method by which this is determined is described below).

Stress-strain curves are required for the plastic analysis. Figure 23, Addendum D, has been reproduced from Figure 13 and Table VI, Reference 2; however, the strain scale has been increased. This curve is for an annealed sheet stock.

Later, two other curves, Figures 24 and 25, Addendum E, were reconstructed using the same data. These two curves were reconstructed using the 0.2-percent yield strength, the ultimate strength, the percent elongation, percent reduction of area, and an estimate of the rupture strength based on Figure 13, Reference 2 (a group of stress-strain curves). Figure 15 of the reference work is an approximation for a three-quarters hard material and Figure 14 is an approximation for a fully hardened material.

3.3.2.6.3 Plastic Analysis Procedure

The stress/strain curve of the material for the metal can be reconstructed at room temperature, as described below.

Recalling from the elastic analysis that the pressure exerted on the epoxy by the transistor can be:

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T}{\frac{R_c}{E_c t_c} \left(1 - \frac{\mu_c}{2}\right) + \frac{1 + \mu_e}{E_e}} \quad (30)$$

and noting that:

$$\alpha_e \Delta T - \alpha_c \Delta T = \text{a constant (say } K_1)$$

and letting:

$$\frac{R_c}{E_c t_c} \left(1 - \frac{\mu_c}{2}\right) = \frac{K_2}{E_c} \quad (31)$$

$$\frac{1 + \mu_e}{E_e} = \frac{K_3}{E_e} \quad (32)$$

$E_e = K_4$, etc., the equation for "P" can be reduced to:

$$P = \frac{K_1}{\frac{K_2}{E_c} + \frac{K_3}{K_4}} \quad (33)$$

This can be further simplified to:

$$P = \frac{K_1 K_4 E_c \times 10^{-6}}{K_2 K_4 + K_3 E_c \times 10^{-6}} \quad (34)$$

For the plastic analysis, E_{cp} is substituted for E_c , where E_{cp} is a reduced or plastic modulus which is explained below.

The maximum combined elastic stress which was calculated is S_a , which is an axial stress. For a pressure of 1,900 psi on the can, a stress of 238,000 psi (Addendum A) is obtained in the cylinder wall, or a ratio of:

$$S_a = \frac{238,000}{1,900} = 125P, \text{ psi} \quad (35)$$

To find the reduced or plastic modulus:

(1) The elastic modulus is extended above the elastic limit (Line S, Figure 23, Addendum D)

(2) The elastic stress, S_a , is then plotted (Point J) on the graph.

(3) Drop a vertical line (Line A, Figure 23) until it intersects the stress-strain curve (Point K).

(4) Draw Line B. The slope of Line B is now the estimated plastic modulus, E_{cp} , and Point K is the new strain-hardened elastic limit.

(5) Using the value found for E_{cp} above, find a new external pressure, P , on the can from Equation 35, above.

(6) Using this value of P , find a plastic stress from:

$$S_a = 125P, \text{ (Point L, Figure 23).} \quad (36)$$

(7) Repeat the above six steps, starting with Line B and finding a new plastic modulus (Line D, Figure 23), and a new P and S_a , (Point M).

(8) Continue this interaction until a plastic stress (Point R on Line G) is found, which resets on the stress-strain curve. This is the assumed maximum stress in the cylinder at the joint, and $S_a = 29,600$ psi.

(A set of calculations is shown in Addendum D.)

4. DISCUSSION OF ANALYSES

4.1 Stress

The above outline represents a short range stress analysis of the stresses to be expected in a typical electronic component due to the pressure exerted by the thermal contraction of the potting compound. It is a sophisticated analysis, but still, due to expediency, it is a "diamond in the rough," and needs considerable cutting and polishing. To provide a clear picture of what was done and to interpret the results, the three basic sections of the analysis (buckling, elastic analysis and plastic stress analysis) are discussed in detail.

The assumptions for the pilot model appear to be good, especially since the effects of elastic foundations (the end plates) were taken into consideration. However, the maximum stresses occur at the joint, and it should be recalled that rather simply anchored joints of good material were assumed. A welded joint would probably be 5 to 15 percent weaker, and a joint with screw threads would be worse. A built-in stress concentrator such as a right angle flange or a recess in the cylinder wall would be even worse, since these types of stress concentrations would further restrict relative movement.

On individual components, such items as the above should be superimposed on the above pilot model, and should be examined individually if they appear severe.

4.2 Buckling

The buckling analysis on the shell shows the external pressure which can be expected to cause failure to be 97,148 psi, in six nodes. Calculations of pressure to be expected are several orders of magnitude lower than this. Actually, this pressure, while the lowest calculated in Table I, Addendum A, is the maximum which can be resisted. The minimum buckling pressure for six nodes could conceivably be as low as 40,000 to 50,000 psi; however, this still provides a very large safety factor in local buckling. Thin tubular-shells normally will buckle locally before the load is fully plastic. (See Reference 12, page 575.) However, the pressure falls off rapidly on our shell if buckling occurs.

4.3 Elastic

The elastic analysis formulations are much more sophisticated and exact than they appear. The discontinuity stresses for the joints include the effects of elastic foundations,

wave effects in the deflection due to the geometry of the joint, and the increase in the moment of inertia (rigidity) of the joint.

The elastic stress analysis shows that extremely high stresses (237,000 psi axial compression, Addendum B, for example) can exist in the pilot model. This stress occurs in the thin cylinder (shell) at the joint. This is so large it cannot be ignored. Hence, the first assumption of the elastic analysis which considered the pressure to be uniformly distributed (but not hydrostatic) was examined more closely.

It was apparent that the pressure in the middle of the cylinder would be lower if the effect of shell deflection on the epoxy were considered. This is the bending deflection. Therefore, it was incorporated into the formula for external (interface) pressure. (See Addendum C). However, this effect was shown to be negligible (0.3%). For this reason, a plastic flow analysis was required.

4.4 Plastic

The plastic flow analysis is more an iterative procedure than an analysis. It assumes that the material follows the stress-strain curve and the problem then is to find a strain that produces a stress which is on the stress-strain curve when a given pressure is applied to the cylinder. Since the only connections between the pressure and the stress are the modulus of elasticity and Poisson's Ratio, they were altered in an iterative process to find a point on the stress-strain curve. This appears to be a crude process and, while not desirable for a long range analysis, it will be shown by examining its assumptions to be a fairly usable approximation for a short range analysis.

In the assumptions for the plastic analysis, Figures 13, 14, 15, 16, 17, 18, and 19 of Addendum A show the effects of assumption (1), which assumed that all parts of the can have the same strain-hardened reduced modulus of elasticity, E_{cp} . The term "reduced modulus of elasticity" means that our iterative process has assumed a steeper slope for the elastic portion of the stress-strain curve, (Line A-C, instead of A-B, Figure 13, Addendum A). This means that we have assumed the nickel material in the shell to be less stiff. The effect of this is best understood by considering the actual effects of plastic flow.

Figure 14A shows the pilot model in its original shape, profile "O". Profile "E" shows its shape (greatly exaggerated) after elastic deformation (just before plastic flow). Profile "p" shows the changes due to plastic flow. Note that this is

a localized effect, occurring only at the ends. It should be recalled that a three dimensional object is considered here and the cylinder is shrinking along each circumference simultaneously with bending along its length. (See Figure 14B.)

Our assumption (1) is now inserted into this picture and we will attempt to evaluate its affect. Figure 15A is a diagram showing how the pressure is distributed along the surface of the cylinder. The vertical heights of the arrows represent their magnitudes (Line "O"). Figure 16A shows the pressure distribution after elastic stresses have been applied. The dotted line "O" is the original level of magnitude of the pressure distribution. Line "e" represents the new lower level of load magnitude, after elastic stresses have caused the cylinder to bend and contract. Line "S" represents (greatly exaggerated) the surface profile of the cylinder. The numbers one, two, and three are locations along the cylinder's longitudinal axis. Number 1 is in the center, the 3's are points on the ends, and the 2's are on the main deflection curve of the surface before reverse curvature due to end effects begins.

If the load magnitudes of Figure 16A are replotted to a roughly relative scale with a flat base we obtain Figure 16B. Note that the pressure is higher at the ends, where deflection is smallest. Figure 17A shows the effect of plastic flow at the ends. Line "P" shows the effects on pressure distribution. Figure 17B is a replot of Figure 17A to a flat base. Figure 18A shows the effect of the assumption that a lower elastic or reduced modulus results in more deflection in the center and, hence, lower pressure. Figure 18B is a replot of Figure 18A showing a pronounced drop in load at the center. Using the above load picture, the cylinder's deflection can now be ascertained.

Figure 19A is an exaggerated sketch of the deflection curve of the cylinder's surface, Line "O" being the original shape, Line "e" the shape after elastic stresses are applied, and Line "p" the effect of plastic flow. Figure 19B shows the effect of our assumption on the center. The actual plastic flow in Figure 19A is not shown in Figure 19B. Figure 19C shows the assumed final composite slope. The ends of the three profiles have been moved together for easier comparison. We now ascertain that the effect of our assumption has been to increase the curvature of the deflection curve to cause greater bending. The effect of this bending is shown in Figure 13. In the elastic analysis, we found Points 1 and 2 (Figure 13) to be on the elastic modulus curve. Points 1 and 2 may be of different sign than Points 3 (i.e. tension), but they can be plotted on the same curve in Figure 13 for convenience.

A rigid plastic analysis would find Points 3 to be re-located at "C" (Figure 13) on the stress-strain curve, while

Points 2 might be slightly higher, and Points 1 would be at about the same location. These points represent elastic stresses.

Our assumption would move Points 1 and 2 to the reduced modulus line (Figure 13), increasing their strains. Points 3 would be relocated at "D", since the bending stresses have been increased. This means our assumption has increased the stress (and strain) at Points 3 (the ends of the cylinder at the joints) and we have calculated a stress that is too high.

From a cursory look at the formulations which have been used, it appears that the assumption discussed above might cause the maximum stress to be about 10-percent too high, possibly even 20-percent.

Assumption (2) uses the apparent stress-strain curve and, since elongation is uniform and plastic strain is only about 3.5-percent, the stress due to this assumption is probably about 3-percent too low.

Assumption (3) is considered valid, since nickel has a very low slope stress-strain curve.

Assumption (5), claiming compression stress-strain curves are the same as tension, probably results in a stress about 5-percent too low, because compressive yield strengths are usually at least 5-percent higher for nickel.

Assuming that the maximum stress theory holds, assumption (6) is correct, even though triaxial loads are present on a differential element. However, there is an error due to the method of calculation. It is included in the estimate for assumption (11).

Assumption (10), that the "theory of superposition" holds, is a correct one, since it is normally valid unless plastic or kinematic collapse occurs. Since plastic hinges cannot develop, (see assumption (7), (8) and (9)) this cannot occur at the pressure levels used. The thin cylinder could become unstable in biaxial compression (Reference 11, pages 385 and 386) if the slope of the stress strain curve above the yield point were steep (a high strain hardening exponent). However, it is very shallow, and the applied load thus falls off rapidly. Also, large hoop stresses are developed from the bending stresses; hence, biaxial collapse cannot occur, because plastic flow in bending again relieves the external pressure. There can be some error in biaxial stresses, since plastic flow occurs in a similar manner to that described in Reference 11, pages 385 and 386. This, however, is included in the estimate for assumption (1).

Due to limitations of time, Section 2, assumption (1) must be used. However, a "long range" program should eliminate this assumption by analyzing several cycles of stress application, finding the total plastic stress range, and applying newer methods of analysis for thermal fatigue. (See Reference 15, Chapter 3 for example.) Since processing is apt to leave a compressive stress in the can, this assumption is necessary.

Assumption (12) was also used for expediency. Its estimated error is included in the estimate for assumption (11). It should be eliminated in a long range analysis.

Assumption (13), which assumes that plastic flow occurs across the entire cross-section, is valid because the shell is so thin. In addition, a comparison of a beam (Reference 8, Case 32, page 108) similar to the one analyzed indicates that even a thicker shell would be fully plastic at these loads. The bending moment above was estimated for a fully plastic beam as $20/27 W = M$ ($W = \text{load}$, $M = \text{moment}$). This is considerably larger than required by limit design (Reference 11, page 407), which is $M = 3/26 W$.

Assumptions (14) and (15) should not be used in a long range analysis, and should be eliminated by finding moments through the use of form factors and interaction curves (such as shown in Reference 12, Chapters 17 and 18). The largest error encountered here is due to the relatively low stress value at which the modulus changes slope (i.e., the low proportional limit). This assumption can result in calculating a higher stress than that which actually exists. This error is of the order of 10 to 15-percent. The above stress analysis has shown that the material property assumptions have several areas which should be examined more closely. The stress/strain curve used for the nickel is for an annealed material. Figure 21, Addendum A, shows an overlay of several curves of Reference 2, Figure 5, page 4, and how greatly material properties vary with heat treatment. Since even trace elements in the chemical composition can affect the heat treatment of nickel (Reference 2, Figures 6 and 7), a chemical analysis of a typical transistor can was made, using an optical emission spectrograph. Table II, Addendum A compares the specification published by the manufacturer (from Reference 2), with the results of the analysis. The underlined value in the analysis indicates the probable concentration of the element. This analysis appears satisfactory, but it should be noted that the presence of 0.6-percent, Mg can raise the recrystallization temperature several hundred degrees.

Since the plastic stresses were so much lower than the elastic stresses, it was concluded that the effect of heat treatment must be checked. Hence, a plastic analysis was made, using a stress-strain curve (Figure 24, Addendum E) for a hardened nickel sheet.

4.5 Plastic Analysis of a Hardened Nickel Can

The same procedure and formulations were used for this analysis as for the first plastic analysis. The analytical results are shown in Table III, Addendum E, and the iteration is shown graphically on Figure 24, Addendum E.

This analysis indicates a plastic stress of 96,988 psi at a strain of 2.9-percent. Since the ultimate strength for hardened nickel sheet is 105,000 psi (Reference 2), this represents a stress level which is much too high. If the nickel can is manufactured as a drawn cup, both the bottom and the top of the can at the corner will be heavily cold worked. The effect of this on ductility is shown in Figure 20, Addendum A. To ascertain how serious the effect of a hard shell material can be, a third plastic analysis was made, assuming a yield strength of 31,500 psi and Rockwell B hardness of 68 (Reference 2, Figure 10).

4.6 Third Plastic Analysis

This analysis is similar to the first one. Calculations are shown in Table IV, and Figure 25, Addendum E, revealing a plastic stress of 48,300 psi and 2.95-percent strain.

A plot of the three stress-strain curves (soft, medium, and hard material) and the plastic stresses calculated with them is shown in Figure 22, Addendum A. The dashed vertical line shows the effect of hardening the nickel can. The shaded area shows the probable approximate value of the actual stress. When the final heat treatment is established, this area will be reduced to an increment (length A-B for example) of the stress-strain curve for that heat treatment.

The effect of hardening is to raise the stress considerably, while the strain reduction is small. This can have a degrading effect on fatigue life, since the stress is more than tripled.

4.7 Elastic and Plastic Analysis of Curing Stresses

Since the differential thermal expansion could cause large stresses (depending on the heat treatment), the question arises: "Would the curing stresses, if added to the thermal stresses, cause failure of our Pilot Model?" To answer this question, an elastic and plastic stress analysis which considered the combined effect of thermal and curing stresses was made. (See Addendum E.)

The following additional symbols were used in this analysis:

- E_{cp} = reduced or plastic stress-strain modulus (psi).
 P_c = external pressure exerted on the can by the epoxy due to curing (psi).
 P_T = total pressure due to both differential thermal expansion and curing (psi).

Figure 26, Addendum E shows how the temperature and curing pressure vary with time during actual injection molding of modulus. Modules contained a diaphragm type transducer, and tests were conducted as described in Reference 1. Note that the curing pressure exists at the time of application (40 to 45 hours) of the thermal stresses. Therefore, the elastic stresses were calculated exactly as before, except the curing pressure, P_c (250 psi), was added to the thermal pressure, P , to give a total pressure, $P_T = P + P_c$.

Table V, Addendum E, shows the results for the elastic analysis, and compares them with the elastic analysis for thermal stresses only. Note that the curing stresses cause approximately a 15-percent higher longitudinal bending stress, etc.

A plastic analysis was then made, using an annealed stress-strain curve, (Figure 27, Addendum E). The iteration is shown in Table VI and Figure 27, Addendum E. The plastic stress is 29,400 psi with 3.10-percent strain. The plastic stress and strain are compared in Table VII with the analysis for thermal stresses. Note that the curing pressure had no effect. This is due to two causes:

(1) The final reduced modulus (assumed) allows enough shrinkage in the can to compensate for a large amount of the differential thermal expansion.

(2) The discussion on assumption (1) above showed that the effect of that assumption, (Figure 19B) was to add deflection (hence, strain) to the main deflection curve. While this raised the maximum plastic axial bending stress (Point D, Figure 13, Addendum A), eventually, as a lower and lower modulus is assumed, a condition will be approached at which any added (spurious) strain will actually reduce the bending stresses by reducing the unit longitudinal strain.

It appears that the above two effects have reached an upper limit at about 3.05 to 3.10-percent strain on the annealed nickel sheet, (Point D, Figure 22, Addendum A). However, the first limitation above now enables us to say that, if we are approaching it, we are also approaching a maximum plastic

stress for this pressure level. Hence, it appears our actual stress due to the thermal expansion and curing lies in the area CDEF in Figure 22, Addendum A and, when the final heat treated condition of the nickel is established, it will lie on a segment (Line A-G for example) of the stress-strain curve.

4.8 Pure Shearing-Stress Analysis

In the above analyses, we have not considered the effect of pure shearing stresses at the joint of the cylinder and can (Item 6 and 13, Figure 7A and 8A, Addendum A). As a quick check, the calculations shown in Addendum F were made. This stress was combined with the original axial bending stress of 237,000 psi as shown. It is apparent that this effect is insignificant. The method of analysis used is greatly simplified, but is on the conservative side, and this shear stress is a maximum value.

4.9 Elastic Stress Analysis of Varying Material Properties

We still do not have enough information to render an engineering judgement on the compatibility of the two materials of our pilot model, because our calculations show that high stresses can exist. Therefore, one more auxiliary analysis must be considered. The properties of nickel will not vary greatly for the temperature change encountered during manufacture; however, those of the potting compound merit study.

Table VIII, Addendum G lists variation with temperature of the modulus of elasticity and the coefficient of thermal expansion for the epoxy potting compound. Figure 28, Addendum G is a plot of these two properties vs temperature.

In addition to the above effects, the assumption that the potting compound is of infinitely large size was eliminated and an outside diameter of one-half inch was assumed. Calculations for the elastic stresses are also in Addendum G. Note that the maximum combined axial stress is more than 100,000 psi lower (45-percent).

A cursory investigation of the formulations indicates the following effects of varying the material properties:

- (1) Effect of a finite epoxy radius mold width on the pressure is negligible.
- (2) Effect of varying the epoxy elastic modulus on the pressure is a 15-percent reduction in stress.

(3) Effect of varying the thermal expansion coefficient on the pressure is a reduction of 25-percent.

(4) The pressure change due to the effects of (3) above reduces as follows:

(a) The bending moment, M_o , 25-percent.

(b) The vertical shear, V_o , 25-percent.

(c) The longitudinal compressive stress due to end loading, S_1 , 25-percent.

(d) The maximum combined axial stress, S_a , 25-percent.

The above analysis represents a considerable reduction in stress level. A similar reduction can be expected for the elastic stress level for a hardened material. However, the method of calculating this stress assumes that each increment of strain, which was calculated with a lower elastic modulus at high temperature, will not be reduced in magnitude when the temperature is reduced to room temperature, at which time the stresses are highest. (This has the same effect as assuming that a dead-loaded metal rod at high temperature will not contract due to a change in stiffness when its temperature is lowered and its modulus increases.) That is, its contraction is considered to be thermal only. Actually, there is an elastic contraction, also. Hence, the stress levels calculated by this method are 15-percent too low. In other words, the effect of considering variation in epoxy material properties due to temperature change, only, lowers stresses about 10-percent. A rough estimate of the plastic stresses which would result from this lower elastic stress indicates they would only be reduced moderately, since the knee of the stress-strain curve changes slope rapidly above the yield strength (of the order of 10,000 psi plastic stress).

The above analysis of our pilot model, as well as the discussion of possible errors, have not considered some effects such as the following:

(1) Does the potting compound bond to the metal can? This would not affect the elastic analysis but would affect the plastic.

(2) Are residual stresses present?

(3) Does the thin RTV rubber coating reduce the stress concentrations to a negligible value? Probably not.

(4) What is the effect of plastic flow of the potting compound?

5. CONCLUSIONS

Until the last set of calculations above was made, nothing could have been predicted about the compatibility of the two materials being studied, except possibly to say that they were marginally unacceptable. In fact, an opinion based on the first elastic and plastic analysis would have had a very damaging result in that we would have rejected most metal/solid-plastic combinations.

The decision to use a constant strain approach for the plastic analysis was a sound one, because of the flat stress-strain curve of nickel (Reference 15, page 104). Plastic strain analyses appear to be extremely crude because of the large quantities of strain involved. However, due to the low slopes of stress-strain curves in high stress regions, they can be pretty accurate (Reference 15, page 110).

Parading all the above assumptions and their possible errors, etc., in front of the reader can create the impression that an answer is unobtainable. This is not entirely true. Many assumptions tend to offset each other, and many are second order effects.

This overall method of analysis, by carefully considering the larger physical effects and noting all the assumptions as the study proceeded, should be quite useable for practical applications; however, the actual material properties of the metallic can must be ascertained more closely than has been done. As shown in Figure 22, Addendum A, they can have a very large effect.

A valid, fairly accurate method of stress analysis has been developed for quick estimation of the compatibility of pairs of materials in potted electronic assemblies.

The method described will be suitable for accurate predictions of the compatibility of material pairs, but only if the plastic stress analysis is modified considerably and several assumptions are eliminated.

The epoxy-nickel combination which was analyzed is compatible, but only if the nickel is stress relieved after drawing and receives a post heat treatment providing a lower hardness than Rockwell B70. Any higher hardness will probably cause a small percentage of cans to fail.

6. RECOMMENDATIONS

It is recommended that the above method be modified to develop a stress analysis computerized program which can be used to screen out noncompatible material pairs for potted electronic components. (See Addendum H.)

It is also recommended that the best heat treatment for the nickel transistor-can be ascertained and made a required specification for acceptance of parts.

Addendum A

Illustrations and Tables

Figure 1

Cross-Section of Mold With Pinboard in Place - Ready to Mold

Figure 2

Detailed View of Section A in Figure 1 - Showing Flow of Potting Compound

Figure 3

A Partially-Molded Module - Showing Shape of the Flow-Front

Figure 4

Transistor Can

Figure 5

Transistor Can as Altered for the Analysis

Figure 6

Typical Locations of Differential Elements

Figure 8
Differential Element at Edge of Disk

Figure 9

Tension and Compression at Joint of Pilot Model Can

TABLE I
*RESULTS OF A CRITICAL BUCKLING
ANALYSIS OF THE SHELL OF THE PILOT MODEL

<u>BUCKLING NODES, N_1</u>	<u>CRITICAL BUCKLING PRESSURE, psi</u>
2	118,343
3	123,165
4	109,403
5	99,861
6	97,748
7	102,160
8	111,880
9	125,980
10	143,850
11	165,057
12	189,300
13	216,400
14	246,300
15	278,900
16	313,900
17	351,500

*Bendix Computer Program No. 143, Quality Control

Figure 10

Shrinkage Profiles

Figure 11

Pressure Stresses
Hoop and Axial

Figure 12

Typical Stress-Strain Curve for Annealed A-Nickel

TABLE II

COMPARISON OF MATERIAL PROPERTIES OF A TRANSISTOR CAN
WITH SPECIFICATIONS OF THE MANUFACTURER

ELEMENT PERCENT		
<u>ELEMENT</u>	<u>MANUFACTURER'S SPECIFICATION</u>	<u>AS ANALYZED</u>
Nickel	99 minimum	Balance
Manganese	0.35 maximum	<u>0.1</u> - 1.0
Calcium	---	<u>0.001</u> - 0.01
Titanium	---	<u>0.01</u> - 0.1
Copper	0.25 maximum	0.01 - <u>0.1</u>
Silicon	0.1 maximum	<u>0.1</u> - <u>1.0</u>
Magnesium	---	0.01 - <u>0.1</u>
Sulfur	0.01	---
Iron	0.4 maximum	<u>0.01</u> - 0.1
Carbon	0.15 maximum	---

Figure 14
Pilot Model Profiles Showing Deflections

Figure 15

Pressure Distribution at Instant Before Deformation

Figure 16

Effect of Elastic Deformation on Pressure Magnitude and Distribution

Figure 17

Effect of Plastic Flow on Figure 16

Figure 18

Effect of Assumption (1) on Plastic Flow of Figure 17

-A-
Elastic Deformation

-B-
Plastic Deformation

-C-
Composite Overlay

Figure 19.

Effect of Assumed Plastic Flow on Deformation Curve of Cylinder Surface

Figure 20

Heat Treating Nickel

Figure 21

Effects of Cold Working and Hardness

Figure 22

Comparison of Stress-Strain Values for Different Hardnesses

Addendum B

Example

Elastic Analysis Calculations

EXAMPLE OF ELASTIC ANALYSIS CALCULATIONS

1. Equilibrium pressure acting at the interface of the transistor can and the potting compound (epoxy). (Figure 7A, Item 15.)

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T}{\frac{R_c}{E_c t_c} (1 - \frac{\mu_c}{2}) + 1 + \frac{\mu_e}{E_e}}$$

$$= \frac{(27)(202.7)(10^{-6}) - (7.4)(365)(10^{-6})}{\frac{0.1516}{(30)(10)^6(0.0114)} (1 - \frac{0.3}{2}) + \frac{1 + 0.5}{(1.4)(10)^6}} = 1,900 \text{ psi}$$

2. Uniform axial compressive strength due to P. (Figure 7A, Item 5.)

$$S_1 = \frac{PR_c}{2t_c} = \frac{(1,900)(0.1516)}{(2)(0.0114)}$$

$$= 12,700 \text{ psi, comp.}$$

Parallel to axis
of cylinder

3. Uniform hoop compressive stress due to P. (Fig. 7A, Item 4.)

$$S_2 = \frac{PR_c}{t_c}$$

$$= \frac{(1,900)(0.1516)}{(0.0114)}$$

$$= 25,000 \text{ psi comp.}$$

Parallel to axis
of cylinder

4. Flexural modulus of the cylinder, D_c :

$$D_c = \frac{E_c t_c^3}{12(1 - \mu_c^2)}$$

$$= \frac{30(10)^6(0.0114)^3}{12[1 - (0.3)^2]} = 4.57 \text{ inch pounds}$$

5. Flexural rigidity and foundation elasticity characteristic:

$$\lambda_c = \sqrt[4]{\frac{3(1 - \mu_c^2)}{R_c^2 t_c^2}} = \sqrt[4]{\frac{3[1 - (0.3)^2]}{(0.1516)^2 (0.0114)^2}} = \frac{30.255}{\text{inches}}$$

6. Bending moment, M_o , due to discontinuity stresses:

$$M_o = \frac{\frac{PR_c^3 \lambda_c^2}{4(1+\mu_c)} + \frac{2PR_c^2 \lambda_c^3 D_c}{(1 - \frac{\mu_c}{2}) [E_c t_c + 2R_c D_c \lambda_c^3 (1 - \mu_e)]}}{2 \lambda_c^2 + \frac{2R_c \lambda_c^2}{1 + \mu_c} - \frac{2E_c t_c}{E_c t_c + 2D_c \lambda_c^3 R_c (1 - \mu_c)}}$$

Substituting values and solving:

$$M_o = \frac{\frac{(1,900)(0.1516)^3(30.255)^2}{(4)(1+0.3)} + \frac{(2)(1,900)(0.1516)^2(30.255)^3(4.57)}{(1+\frac{0.3}{2})[(30)(10)^6(0.0114) + (2)(0.1516)(4.57)(30.255)^3(1-0.3)]}}{(2)(30.255)^2 + \frac{(2)(0.1516)(30.255)^2}{(1+0.3)} - \frac{(30.255)(30)(10)^6(0.0114)}{(30)(10)^6(0.0114) + (2)(4.57)(30.255)^3(0.1516)(1-0.3)}}$$

Solution:

$$M_o = 4.88 \text{ (inch-pounds)}$$

Note that the direction of this moment tends to decrease the diameter at the ends of the cylinder, although it reverses as it recedes from the edge.

7. Vertical shear, V_o , due to discontinuity stresses:

$$\begin{aligned}
 V_o &= M_o \left[2\gamma_c + \frac{2R_c \gamma_c^2}{1 + \mu_c} \right] - \frac{\gamma_c E_c t_c}{E_c t_c + 2D_c \gamma_c^3 R_c (1 - \mu_c)} \\
 &= 4.88 \left[2(30.255) + \frac{(2)(0.1516)(30.255)^2}{(1 + 0.3)} \right] - \frac{(1,900)(0.1516)^3 (30.255)^2}{30(10)^6 (0.0114) + 2(4.57)(30.255)^3 (0.1516)(1 - 0.3)} \\
 &= 171.8 \text{ lb/in}^2 \quad (\text{Upward at the edge})
 \end{aligned}$$

8. Calculation of discontinuity stresses:

(a) Axial bending stress due to M_o :

$$S'_{d1} = \frac{6M_o}{t_c^2} \quad \begin{array}{l} \text{Parallel to} \\ \text{Axis of} \\ \text{Cylinder} \end{array}$$

$$= \frac{(6)(4.88)}{(0.0114)^2}$$

$$= 225,300 \text{ psi}$$

This stress causes tension on the outside surface at the edge and compression on the inside surface.

(b) Hoop stress due to vertical shear, V_o :

$$S''_{d2} = \frac{2V_o \gamma_c}{t_c}$$

$$= \frac{2(771.8)(30.255)(0.1516)}{0.0114}$$

$$= 138,000 \text{ psi (tension)}$$

(c) Hoop stress due to bending moment, M_c :

$$\begin{aligned}
 S_{d_2} &= \frac{2M_o \lambda_c^2 R_c}{t_c} \\
 &= \frac{2(4.88)(30.255)^2(0.156)}{0.0114} \\
 &= 118,800 \text{ psi, (compression)}
 \end{aligned}$$

(d) Hoop bending stress due to bending moment, M_o :

$$\begin{aligned}
 S'_{d_2} &= \frac{6 \mu_c M_o}{t_c^2} = \mu_c S'_1 \\
 &= (0.3)(225,300) \\
 &= 67,600 \text{ psi}
 \end{aligned}$$

This stress causes tension at the outer surface of the cylinder and compression at the inner surface.

9. Maximum axial combined stress:

$$\begin{aligned}
 S_a &= S'_{d_1} + S_1 \\
 &= -225,000 + (-12,700) \\
 &= -237,700 \text{ psi}
 \end{aligned}$$

This stress is compressive, as shown below, and occurs on the inner surface at the edge of the cylinder.

10. Maximum compressive hoop stress:

$$\begin{aligned} S_c &= S_2 + S''_{d_2} + S_{d_2} + S'_{d_2} \\ &= -25,000 + 138,000 - 118,800 - 67,600 \\ &= -73,000 \text{ psi} \end{aligned}$$

This stress occurs on the inner surface at the edge.

11. Maximum tensile hoop stress:

$$\begin{aligned} S_c &= S_2 + S''_{d_2} + S_{d_2} + S'_{d_2} \\ &= -25,000 + 138,000 - 118,800 + 67,600 \\ &= +61,800 \text{ psi} \end{aligned}$$

This stress appears as tension on the outer surface.

12. Stress Diagram

The three maximum stresses which occur are shown in the following sketch.

These stresses exist in the wall of the transistor can

Addendum C

Example

Calculations of Stress Relief due to Bending

CALCULATIONS OF STRESS RELIEF DUE TO BENDING DEFLECTION IN THE CAN WALL

1. CALCULATE DEFLECTION

For deflection of $x = 0.125$ at the center of the beam (cyl):

$$\delta_B = \frac{V_O \cos \lambda x - \lambda_c M_O (\cos \lambda x - \sin \lambda x)}{2 \lambda_c^3 D_c e^{2x}}$$

Since:

$$\lambda = 30.255 \text{ inch}^{-1}$$

$$D = 4.57 \text{ lb-in.}$$

$$\lambda_c x = 3.782$$

$$e^{\lambda x} = 43.82$$

$$\cos \lambda x = -0.802$$

$$\sin \lambda x = -0.597$$

$$\cos \lambda x - \sin x = -0.205$$

$$V_O = 143 \text{ lb/in and } M_O = 4.046 \text{ lb-in/in.}$$

Therefore:

$$\delta_B = \frac{(143)(-0.802) - (30.255)(4.046)(-0.205)}{(2)(30.255)^3(4.57)(43.82)} = -8.077 (10)^6 \text{ inch.}$$

The minus sign indicates that deflection is radially inward.

2. CALCULATE INTERFACE PRESSURE

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T - \delta_B}{\frac{R_c}{E_c t_c} \left(1 - \frac{c}{2}\right) + \frac{1 + \mu_e}{E_c}}$$

$$P = \frac{(27)(202.7)(10^{-6}) - (7.4)(365)(10^{-6})}{\frac{0.156}{(30)(10^3)(0.114)} \left(1 - \frac{0.3}{2}\right) + \frac{1 + 0.5}{(1.4)(10^3)}} = -1,890 \text{ psi (Compressive)}$$

3. CALCULATE PERCENT CHANGE

$$\frac{1900 - 1890}{1900} \times 100 = 0.3 \text{ percent}$$

Addendum D

Example

Plastic Stress Analysis Calculations

EXAMPLE OF PLASTIC STRESS ANALYSIS CALCULATIONS

1. Find the external pressure on the can by using the plastic modulus, E_{cp} , represented by Line B of Figure 23, page B-93:

$$E_{cp} = \frac{S}{\epsilon} \text{ psi} = \frac{65,000 \text{ psi}}{0.026 \text{ in/in}} = 2.50(10^6) \text{ psi}$$

Where S = stress increment (psi) and ϵ = strain increment (in/in)

$$P = \frac{K1 K4 E_{cp} (10^{-6})}{K2 K4 + K3 E_{cp} (10^{-6})} \text{ psi} \quad (\text{from Equation 34, page B-41}).$$

$K1 \times K4 = 9.97$ and $K3 = 1.071$; therefore;

$$P = \frac{2,774 E_{cp} (10^{-6})}{9.97 + 1.07 E_{cp} (10^{-6})} = \frac{2,774 (2.5) (10^6) (10^{-6})}{9.97 + 1.07 (2.5) (10^6) (10^{-6})}$$

$$P = 548 \text{ psi}$$

2. Determine the maximum combined stress, S_a :

$$S_a = 125 P = 125(548) = 68,500 \text{ psi}$$

This stress is Point L on Line B of Figure 23.

3. Drop a vertical line (Line C) to the stress-strain curve at Point N, and draw a new plastic modulus curve (Line D).

4. Determine a new E_{cp} :

$$E = \frac{\Delta S}{\Delta \epsilon} = \frac{55,000}{0.052} = 1.05 (10^6) \text{ psi}$$

5. Calculate a new pressure, P :

$$P = \frac{(2,774) (1.05) (10^{-6}) (10^6)}{9.97 + 1.07 (1.05) (10^6) (10^{-6})} = 263 \text{ psi}$$

$$S_a = 125 (263) = 32,900 \text{ psi}$$

6. Locate S_a on the modulus curve, Line D, at Point M.

7. Drop a vertical line (Line B) from Point M to the stress-strain curve at Point O on Figure 23.

8. Draw a new plastic modulus curve (Line F), and determine a new modulus, E_{cp} :

$$E_{cp} = \frac{45,000}{0.047} = 0.95 (10^6) \text{ psi}$$

9. Calculate a new pressure:

$$P = \frac{2,774 (0.95)}{9.97 + 1.07 (0.95)} = 240 \text{ psi}$$

10. Calculate a new plastic stress, S_a :

$$S_a = 125 (240) = 30,000 \text{ psi}$$

11. Locate S_a on the plastic modulus curve at Point P (not shown).

12. Draw a vertical line (not presently shown in Figure 23) to the stress-strain curve at Point O.

13. Draw a new plastic modulus line (Line G), and find the final modulus, E_{cp} :

$$E_{cp} = \frac{40,000}{0.0428} = 0.94 (10^6) \text{ psi}$$

14. Calculate a new pressure:

$$P = \frac{2,774 (0.94)}{9.97 + (1.07) (0.95)} = 237 \text{ psi}$$

15. Calculate a plastic stress, S_a :

$$S_a = 125 (237) = 29,000 \text{ psi}$$

16. Locate S_a on the plastic modulus line, Point R. Since this point coincides with the stress-strain curve, it is the solution:

$$S_a = 29,600 \text{ psi.}$$

Figure 23.
Plastic Stress Analysis
Annealed Nickel Alloy-200 Sheet

Addendum E
Results of Stress Analyses

TABLE III

RESULTS OF PLASTIC ANALYSIS FOR THE
HARDENED NICKEL SHEET*

ITERATION No.	REDUCED MODULUS, E_{cp} (psi x 10^6)	EXTERNAL PRESSURE, P (psi)	PLASTIC STRESS (psi)
1	10.9	1,400	175,000
2	4.53	848	106,000
3	4.17	801	100,200
4	4.03	782	97,700
5	3.98	776	96,988

*See graphic plot on the following page.

Figure 24
Plastic Stress Analysis
Hard Nickel/Steel Sheet

TABLE IV

RESULTS OF THE THIRD PLASTIC STRESS ANALYSIS
(MEDIUM HARDNESS MATERIAL) *

ITERATION No.	REDUCED MODULUS, E_{cp} (psi x 10^6)	EXTERNAL PRESSURE, P (psi)	PLASTIC STRESS (psi)
1	4.54	850	106,181
2	2.0	458	57,200
3	1.7	400	50,000
4	1.6	386	48,300

*See graphic plot on the following page.

Figure 25
Plastic Stress Analysis
Medium-Hard Nickel Sheet
(R_B 70 with 36,000 Y.S.)

TABLE V

COMPARISON OF ELASTIC THERMAL STRESSES
FOR CURED AND UNCURED POTTING COMPOUND

<u>TYPE OF STRESS</u>	<u>THERMAL STRESSES ONLY (P=1,900 psi)</u>	<u>CURING PRESSURE AND THERMAL STRESSES (P_T= 2,150 psi)</u>
Uniform Longitudinal Stress, S ₁	12,700 psi	14,400 psi
Uniform Hoop Stress, S ₂	25,000 psi	28,300 psi
Bending Moment, M _O	4.88 in-lb/in	5.52 in-lb/in
Shear Force, V _O	171.8 lb/in	194.4 lb/in
Longitudinal Bending Stress, S' _{d1}	225,300 psi	255,000 psi
Direct Hoop Stress due to V _O , S'' _{d2} : : : : .	138,000 psi	156,200 psi
Direct Hoop Stress due to M _O , S' _{d2} : : : : .	118,800 psi	134,400 psi
Hoop Bending Stress due to M _O , S' _{d2}	67,600 psi	76,500 psi
Maximum Combined Longitudinal Stress, S _a	-238,000 psi	-269,400 psi
Maximum Combined Com- pressive Hoop Stress, S _c	-73,000 psi	-83,000 psi
Maximum Combined Tensile Hoop Stress, S _c	61,800 psi	70,000 psi

Figure 26. Curing Pressure and Temperature versus Molding Time

TABLE VI
PLASTIC ANALYSIS FOR COMBINED
THERMAL AND CURING STRESSES

ITERATION No.	REDUCED MODULUS, E_{cp} (psi x 10^6)	EXTERNAL PRESSURE * P_T (psi)	PLASTIC STRESS (psi)
1	2.35	520	65,300
2	1.02	257	32,100
3	0.95	240	30,000
4	0.93	235	29,400

* $P_T = P + P_C = 1,900 + 250 = 2,150$ psi (See Figure 27)

TABLE VII
COMPARISON OF PLASTIC STRESSES
FOR A CURED AND UNCURED POTTING COMPOUND

PARAMETER	MATERIAL CONDITION	
	CURED	UNCURED
Maximum Longitudinal Stress, S_a	29,600 psi	29,400 psi
Total Strain	3.10 %	3.10 %

Figure 27
Plastic Stress Analysis for Curing Stress
and
Differential Thermal Expansion Stress

Addendum F

Example

Derivation of Pure Bending Stress

DERIVATION OF PURE BENDING STRESS

1. Assuming that the cylinder is being sheared inward at the ends due to the total pressure on its surface:

$$\text{Pure shear stress} = \frac{\text{total pressure on the cylinder}}{\text{total shear area}}$$

or symbolically:

$$S_s = \frac{PA_c}{2A_s}$$

Since the cylinder is thin:

$$S_s = \frac{P\pi ld}{2\pi dt} ; = \frac{Pl}{2t} \text{ psi}$$

where:

S_s = pure shearing stress (psi)

P = unit external pressure (psi)

A_c = surface area of the cylinder (in.²)

A_s = cross-sectional area of cylinder, or joint

l = length of the cylinder (inches)

d = diameter of cylinder (inches)

t = thickness of cylinder (inches)

2. Using $P = 1,900$ psi, from Addendum B, and $l = 0.250$ inch:

$$S_s = \frac{1,900 (0.250)}{2(0.0114)} ; = \frac{475}{0.0228} ; = 20,833 \text{ psi}$$

3. Combining this with the maximum combined axial stress for a new maximum stress:

$$S_a = \sqrt{(237,700)^2 + (20,833)^2} ;$$

$$= 1,000 \sqrt{56,500 + 433.9} ;$$

$$= 1,000 \sqrt{56,934} ;$$

$$= 237,700 \text{ psi}$$

Addendum G

Example

Effects of Varying Materials Properties (Data and Calculations)

CALCULATIONS TO DETERMINE THE EFFECTS OF TEMPERATURE UPON MATERIAL PROPERTIES OF EPOXY E-3824 COMPOUND

1. External Pressure on the Cylinder

- a. Using the original formulation for the external pressure and modifying it to include a finite outer radius for the epoxy:

$$P = \frac{\alpha_e \Delta T - \alpha_c \Delta T}{\frac{R_c}{E_c T_c} \left(1 - \frac{\mu_c}{2}\right) + \frac{1}{E_e} \left[\frac{R_o^2 + R_e^2}{R_o^2 - R_e^2} - \mu_e \left(\frac{R_c^2}{R_o^2 + R_c^2} - 1 \right) \right]}$$

- b. Let $R_o = 0.250$ inch and calculate pressure, P , for a temperature drop of 50°F (220 to 170°F):

$$P = \frac{[(36.8)(50)(5/9) - (7.4)(50)](10^{-6})}{\frac{(0.1516)(1 - 0.3/2)}{(30)(10^6)(0.0114)} + \frac{1}{1.21(10^6)}} \quad (\text{continued})$$

$$= \frac{[(0.25)^2 + (0.1516)^2]}{[(0.25)^2 - (0.1516)^2] - (0.5) \left(\frac{(0.1516)^2}{(0.25)^2 - (0.1516)^2} - 1 \right)} (10^{-6})$$

$$P = 279 \text{ psi}$$

- c. Using the formula in "a," calculate the increase in pressure - in increments down to -80°F . (Results only are shown.)

TEMPERATURE INCREMENT (degrees F)	PRESSURE INCREMENT (psi)
170 to 120	195
120 to 70	193
70 to 20	191
20 to -30	110
-30 to -80	71

- d. Summing pressure increments:

$$P = \Sigma \Delta P = 279 + 195 + 193 + 191 + 71 = 1,039 \text{ psi}$$

2. Calculations of the elastic stresses in the transistor can proceed in the same manner as the previous elastic stress calculations.

- a. Uniform Longitudinal Stress:

$$\frac{PR_c}{2t_c} = 6,900 \text{ psi.}$$

- b. Uniform Hoop Stress:

$$\frac{PR_c}{t_c} = 13,800 \text{ psi.}$$

- c. Moment, M_o , due to discontinuity = 2.67 in. pounds.

- d. Vertical shear, V_o , due to discontinuity = 94 pounds.

- e. Axial bending stress:

$$S'_{d_1} = \frac{6M_o}{t_c} = 123,200 \text{ psi.}$$

- f. Direct hoop stress due to V_o :

$$S''_{d_2} = \frac{2V_o \lambda R_c}{t_c} = 75,700 \text{ psi.}$$

- g. Direct hoop stress due to M_o :

$$S_{d_2} = \frac{2M_o \lambda^2 R_c}{t_c} = 65,000 \text{ psi.}$$

- h. Hoop bending stress due to M_o :

$$S'_{d_2} = \frac{6\mu M_o}{t_c} = 37,000 \text{ psi.}$$

- i. Maximum combined axial stress, $S_a = 130,100 \text{ psi.}$

- j. Maximum combined compressive hoop stress, $S_c = 40,100 \text{ psi.}$

TABLE VIII
 VARIATION OF MATERIAL PROPERTIES WITH TEMPERATURE
 EPOXY E-3824 POTTING COMPOUND*

<u>TEMPERATURE CHANGE ΔT, °F</u>	<u>LINEAR COEFFICIENT OF THERMAL EXPANSION (in/in/°F x 10⁻⁶)</u>	<u>MODULUS OF ELASTICITY (psi x 10⁻⁶)</u>
220 to 170	36.8	1.21
170 to 120	29.3	1.295
120 to 70	28.4	1.37
70 to 20	27.8	1.43
20 to -30	21.4	1.48
-30 to -80	18.5	1.51

*Source, Reference 5.

Figure 23. Variations of Material Properties With Temperature Changes - Fiberite E-9405

Addendum H

A Proposal for a Computerized Cyclic Stress Analysis

This project would create a stress analysis with an accompanying computer program. The stress analysis would be an extension of the present one, but would be considerably more complex. It would simulate the environment of a potted electronic component as it is potted, heated, cured, cooled, and then thermal cycled.

This computer program could be used to give material pairs a final screening for compatibility, before an assembly is designed. It would compute the stresses encountered during various parts of the thermal cycle, and permit study of the changes in stresses from cycle-to-cycle, as well as stress reversals during each cycle.

The project would require the following:

A. Expansion and modification of the present stress analysis, as follows:

1. Buckling

Use the present analysis, but add an analysis for circumferential buckling to be used when extremely high pressures are encountered.

2. Elastic Analysis

The existing analysis would be used, without change.

3. Plastic Analysis

a. Develop a set of strain equations which would consider plastic flow in all areas of the transistor component simultaneously.

b. Develop principal stress equations which would consider plastic flow as biaxial stress fields develop.

c. Calculate the different stress levels as the temperature is varied.

d. Calculate changes in stress levels as temperature reverses, from high to low to high. Example: from cure temperature to -65 to 165 and back to -65°F, etc.

e. Refine the pure shear analysis.

f. Add the effects of a finite size potting compound.

g. Material Properties Considerations

- (1) Determine the effects of variations of material properties.
 - (a) Modulus of elasticity, with direction (potting compound only).
 - (b) Heat treatment (metallic material only).
 - (c) Different tensile and compressive stress/strain curves (both types of materials).
- (2) Determine the effects of:
 - (a) Bauschinger effect.
 - (b) Curing shrinkage (potting compound only).
 - (c) Potting compound residual stresses.

h. Incorporate a prediction for thermal fatigue life.

4. Determine Material Properties

The above analysis would require determination of many material properties which are not now obtainable commercially. These would be required for each pair of materials to be examined. Properties would be required from -100 to 400°F in 100-degree increments. Some of these are listed below:

a. Absolutely Required:

- (1) True-stress/true-strain curves in tension and in compression.
- (2) Modulus of elasticity for potting compound, in fourteen directions (15 apart in two planes 90 degrees apart).
- (3) Poisson's Ratio.
- (4) Coefficient of Thermal Expansion.
- (5) Coefficient of Thermal Conductivity.
- (6) Bauschinger Effect.
- (7) Thermal fatigue strain cycle constants.
- (8) Yield Strength, 0.2-percent (metal only).
- (10) Fatigue Life - low cycle.
- (11) Heat treatment effects on material properties.

b. Desired:

- (1) Fatigue (S-N) curve.
- (2) Residual curing stresses (potting compound only).
- (3) Viscoelastic properties (potting compound only).

B. Write a computer program incorporating the entire analysis.

C. Cost Estimate:

1. Rewrite and expand stress analysis	3,320 man-hrs.		
2. Study material properties	450	"	"
3. Develop material properties, per pair:			
a. Required properties	1,140	"	"
b. Desired properties	700	"	"
4. Incorporate analysis into a computer program	800	"	"
Total man hours	6,410*		
5. Total elapsed time would be 9 to 15 months, if priority is available.			

D. Project Manpower

For consultation purposes, more than one engineer would be required approximately 50-percent of the time. Approximately 90-percent of Item C.3, above, would be technician time. About 90-percent of Item C.4 would be programmer's time.

The above estimate does not include charges for computer time nor test samples.

*Note that for the computerized stress analysis only, 4,120 man-hours are required.

Addendum I

References

REFERENCES

1. Thompson, C., "Interim Report on the Study of Transfer Molding Electronic Assemblies," Bendix Test Laboratory, July, 1967.
2. "Engineering Properties of Nickel 200 and 201," Technical Bulletin T-15, Huntington Alloy Products Division, Int. Nickel Co., Inc.
3. Fiberite Molding Compounds, Bulletin, Fiberite Corporation.
4. "Materials in Design Engineering," Materials Selector Issues, Reinhold Publishing Corporation, Mid-October, 1966-67.
5. Clocbough, William, Fiberite Corporation.
7. Roark, R. J., Formulas for Stress and Strain, 3rd Edition, McGraw-Hill, Case "R", p. 318, 1954.
8. Roark, R. J., Formulas for Stress and Strain, 3rd Edition, McGraw-Hill, 1954.
9. Hetenyi, Beams on Elastic Foundations, University of Michigan Press, 1955.
10. Timoshenko, "Advanced Theory and Problems," Strength of Materials, Part II, D. Van Nostrand, 1960.
11. Taupel, J. H., Engineering Design, John Wiley and Sons, Inc., 1964.
12. Seely and Smith, Advanced Mechanics of Materials, John Wiley and Sons, Inc., 1959.
13. Nadai, A., Theory of Flow and Fracture of Solids, McGraw-Hill Book Company, 1950.
14. Proger, W., Introduction to Plasticity, Addison-Wesley Publishing Company.
15. Manson, S. S., Thermal Stress and Low Cycle Fatigue, McGraw-Hill Book Company, 1966.