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B. R. Dewey

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THERMAL STRESS ANALYSIS OF AN ALUMINUM-MYLAR TRANSFORMER COIL

B. R. Dewey

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THERMAL STRESS ANALYSIS OF AN ALUMINUM-MYLAR TRANSFORMER COIL

B. R. Dewey

ABSTRACT

Linear finite-element modeling techniques were used to study temperature and stress distribution in a Mylar-aluminum transformer coil. We first investigated approaches to dividing the structure into discrete elements: a simplified model was adequate for predicting maximum stresses and displacements. Stress distributions produced with stainless steel and with fiberglass outer coverings were compared. The fiberglass-covered coil showed a peak interlayer pressure of about 4.3 MPa (600 psi) and a maximum hoop stress of about -20 MPa (3000 psi). Corresponding stresses in the stainless-steel-covered coil were about one-third higher. In both cases the maximum interlayer pressure is at the center plane of the coil about halfway between the inner and outer surfaces. Simplifying these assumptions should provide results adequate for identifying problem areas within the coil.

INTRODUCTION

A new, economical electrical transformer design utilizes a coil wound from alternate layers of aluminum foil and Mylar* film. Mechanical properties tests on these materials and on a prototypic coil have been started at ORNL. Guidance for the placement and interpretation of the instrumentation on the test coil is presented in this finite-element thermal stress analysis.

ANALYSIS TECHNIQUE

The finite-element technique[1] is well established for heat transfer and elasticity analysis. In this study we investigated three different basic models to establish the validity of making simplified meshes.

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*Trademark, E. I. DuPont de Nemours and Co., Wilimington, Del.

1. A 2 1/2-dimensional (D) model is an axisymmetric representation of the coil with the center plane taken as a plane of symmetry. This model is more expensive to analyze than are the two methods below.

2. A plane-stress 2-D model is also an axisymmetric representation, but has no constraint in the axial direction. Because the coil is free to displace axially from the effect of temperature and load, the axial stresses are zero. This model also gives lower radial and tangential stresses than does the 2 1/2-D model.

3. A plane-strain 2-D model is also an axisymmetric one, but the axial direction is fully constrained, the axial stresses are no longer zero, and the radial and tangential stresses are higher than in the 2 1/2-D model.

Our first assumption was that models 2 and 3 tended to bound the solution. It is much more economical to analyze both models 2 and 3 than it is to determine the full 2 1/2-D solution.

Limitations and assumptions in this study should be noted. We decided not to model the material as orthotropic because no previous work exists to establish the orthotropic elastic properties for this layered construction. The layers may or may not be bonded together; thus, it is uncertain if continuity of tangential and axial stress distribution exists. The interlayer pressure, which is the negative of radial stress and is thus important in this study, is continuous because radial stress is transmitted from layer to layer. The finite-element model gives continuity of radial displacements but not exactly of stresses because the stresses are calculated from displacements, and some numerical inaccuracy is inevitable. The finite-element analysis includes descriptions of the geometry, material, and loading of the structure.

Geometric Model

Dimensions and construction of the particular coil[2] analyzed are shown in Fig. 1. The extension of the inner core above and below the coil (shown in Fig. 1) and the excess Mylar that overhangs the aluminum layers (not shown in Fig. 1) were not included in the finite-element model. Because of symmetry, only the top one-half of the coil is considered in

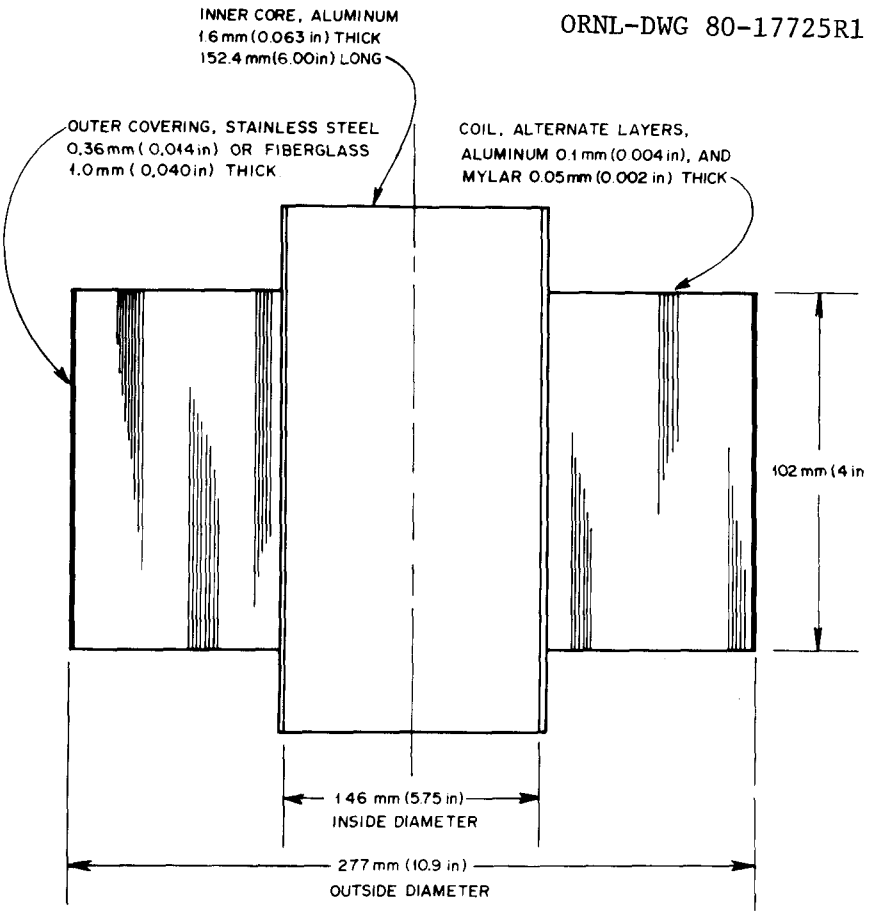

Fig. 1. Aluminum-Mylar transformer coil.

the 2 1/2-D model and, of course, axisymmetric finite elements can be used. The eight-node isoparametric quadratic element [PAFEC element 36210 (ref. 1)] was selected for its great tolerance to distortion of the height-width ratio.

The finite-element model did not use elements as thin as the individual layers of foil and film because that would require about 700 elements in the radial direction, which would lead to a very large solution. The 2 1/2-D model would also take about 50,000 to 100,000 degrees of freedom (d.f.), depending on aspect ratio. The 2-D analyses would use from 350 to 1400 d.f., where the lower number is for plane strain. Furthermore, the refinement afforded by this large number of d.f. cannot be expected to improve accuracy. Hence, we decided to use a more moderate number of elements — 520 in the 2 1/2-D analysis (a mesh of 65 elements in the radial direction and 8 elements in the axial direction) and 65 in the

2-D analyses. These meshes are small to moderate in size but are believed to give good results for our problem, although the layers are thicker and fewer than in the actual coil.

The meshes are shown in Figs. 2 and 3. The 2 1/2-D mesh in Fig. 2 shows the thinner elements (Mylar) distorted in height-to-width ratio to about 10:1. Although this amount of distortion is in the "warning" area recommended by PAFEC,[1] it effects a more economical solution and is adequate for models having small shear stresses. On the other hand, the mesh in Fig. 3 for 2-D analysis contains elements that are distorted only 2:1, well within the normal range allowed.

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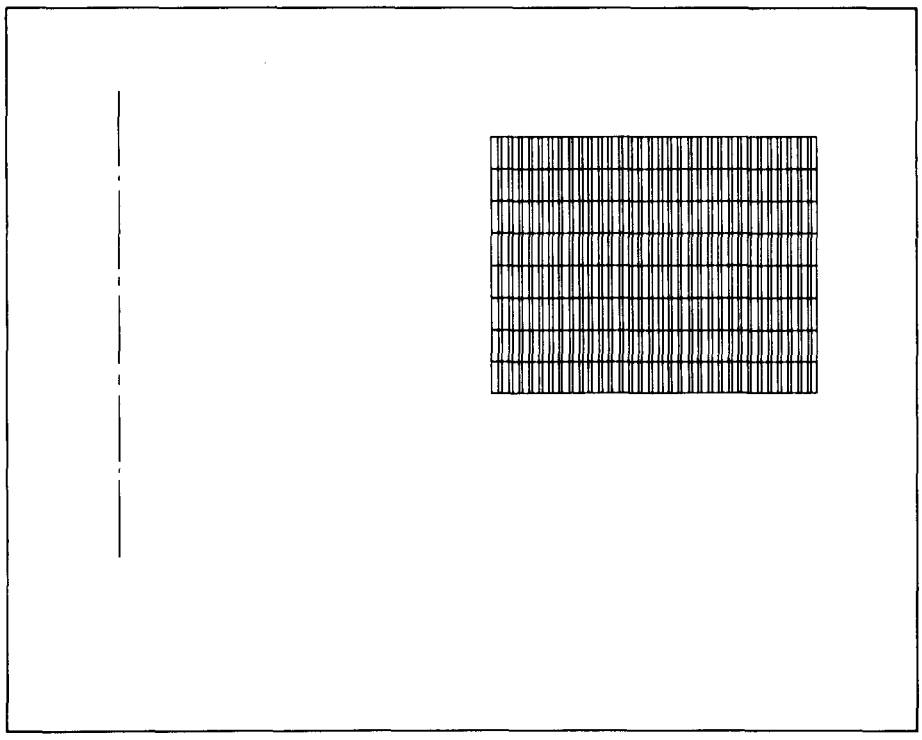

Fig. 2. Mesh for 2 1/2-D analysis. Bottom of the mesh is restrained from vertical displacement, which is the symmetry condition permitting use of one-half the coil.

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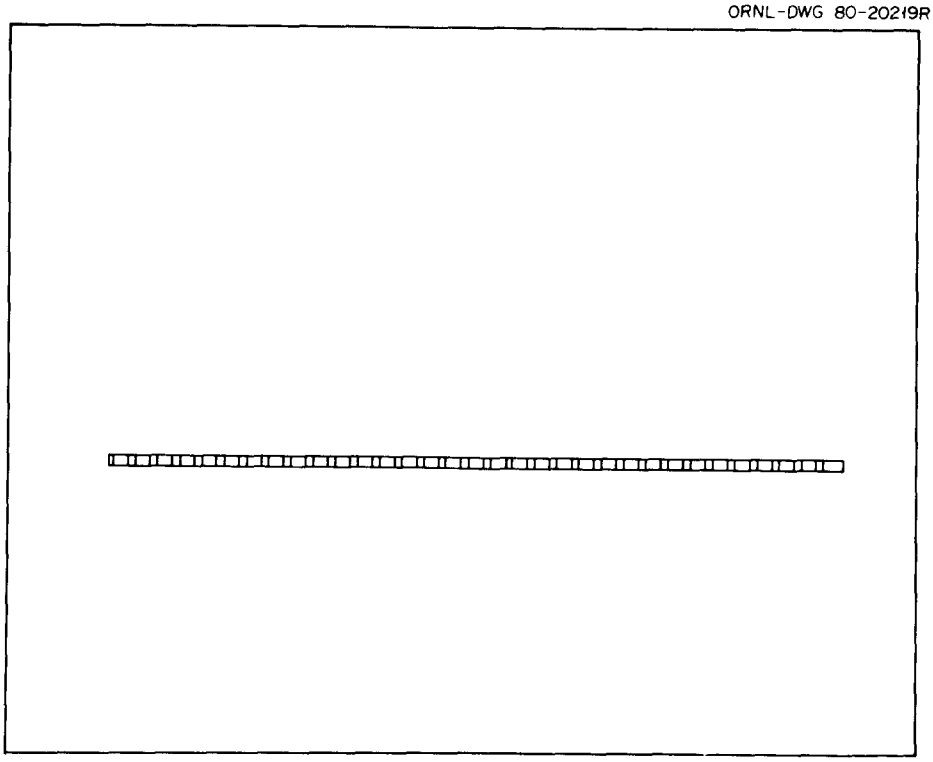

Fig. 3. Mesh for 2-D analysis. For plane strain, all freedoms are restrained in the vertical direction. For plane stress, only the bottom planes of nodes are vertically restrained.

Material Properties

Mechanical and thermal properties of the aluminum, Mylar, stainless steel, and fiberglass used were obtained from various sources. Definite mechanical properties for Mylar cannot be quoted; Mylar is anisotropic because of the long-chain molecules in the plane of the film, and its properties vary from one lot to another. Futhermore, Mylar probably is not linearly elastic, although we have assumed that it is for this analysis. Because hoop stress in the fiberglass predominates, the circumferential properties of the fiberglass were selected for an isotropic approximation.

Selected values of properties used in the analysis are given in Table 1. If it is later found that some of the properties are greatly different from the ones used here, repeating the analysis is not a difficult task.

Table 1. Material properties used in thermal-elastic analysis of transformer coil

	Al-1350	Mylar	300-Series stainless steel	Fiberglass
Modulus of elasticity, GPa	69	3.8	195	34
Poisson ratio	0.33	0.45	0.27	0.25
Thermal expansion, $C^{\circ -1}$	24×10^{-6}	27×10^{-6}	12×10^{-6}	6.5×10^{-6}
Thermal conductivity, W/(m·°C)	238	0.16	24.5	0.34

Loading

Two effects of loading are considered: (1) the temperature distribution in the coil and (2) the geometric restraints on the boundaries of the mesh representing the coil.

Two runs are needed to analyze thermal stress. In the first, the heat transfer problem is solved and the nodal temperatures are stored in a data set. The second run starts with the stored temperatures and performs the elastic stress analysis. The temperature distribution and the mechanical restraints give rise to the particular thermal stress distribution. The same meshes and the same order of elements are used for the two runs. Thus, there is a one-to-one correspondence of both meshes.

We suspected that the coils could also have residual stresses caused by winding tension. No method currently exists for determining the residual stresses in the coil, but it is reasonable to assume that they are somewhat smaller than the thermal stresses. The coil is also loaded by gravity. Distortion from the weight of the coil may be evident in the experiments, depending on the method of support and winding tension. However, in this finite-element analysis, the effects of residual stress and gravity have been ignored.

With time and temperature, the materials tend to creep, and a stress redistribution can occur. This type of nonlinear analysis can be routinely performed with the finite-element method, but we have not addressed it here.

RESULTS

Meshes for the 2 1/2-D and 2-D analyses were prepared. After thermal and mechanical analyses, results from the three basic approaches (models 2 1/2-D, 2-D plane stress, and 2-D plane strain) were compared.

Thermal Analysis

Identical temperature distributions exist for the conditions assumed in the 2 1/2-D and 2-D models. For simplicity and for matching the desired experimental conditions, we assumed that the entire inner core of the coil

had a temperature of 100°C, that the outer covering had a temperature of 70°C, and that the edge surfaces (top and bottom in Fig. 1) were insulated. In the actual coil, the Mylar layers hang out the top and bottom for a considerable distance, which supports the insulated-edge assumption.

The stepped-solid line in Fig. 4 shows the resultant temperature distribution derived from finite-element analysis. Because aluminum is a good conductor compared with Mylar, the aluminum layers are at nearly constant temperature. In contrast, the Mylar layers, which tend to be insulators, show temperature drops. The solutions do not change materially when the covering is changed from stainless steel to fiberglass.

Solution of the Fourier heat condition equation for these conditions is routine and leads to a logarithmic temperature distribution. For comparison, this temperature distribution is shown in Fig. 4 as a dashed line and labeled as the temperature distribution T, which would occur in a homogeneous cylinder. The nodal point temperatures calculated here are used as input conditions for the stress analysis.

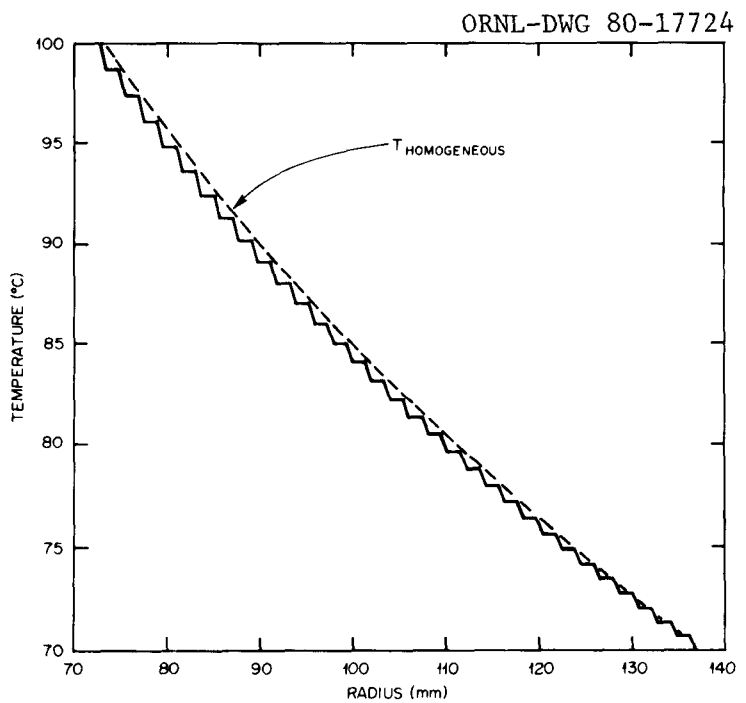

Fig. 4. Results of thermal analysis.

Elastic Stress Analysis for Coil with Stainless Steel Outer Covering

In the finite-element method used, displacements are first calculated, and stresses are calculated on the basis of the solution for displacements. Because of the axisymmetry, there are only radial displacements u and axial displacements w. The most important strain, the hoop strain, is given by the formula u/r, and the radial strain is found from the partial derivative $\partial u/\partial r$, where r is the radius. Axial strain is $\partial w/\partial z$, where z is the axial coordinate. The geometry of the cross section of the transformer coil and the radially only varying temperature render the shear stresses small and unimportant. The 10:1 distortion of the elements in the mesh of Fig. 2 was therefore assumed to be unimportant.

The most direct way to compare the efficacy of the 2 1/2-D against the 2-D solutions is to compare the radial displacements u (Fig. 5). For the 2 1/2-D analysis, u is plotted as a function of radial position for the centerline, for a line halfway between the centerline and outside edge

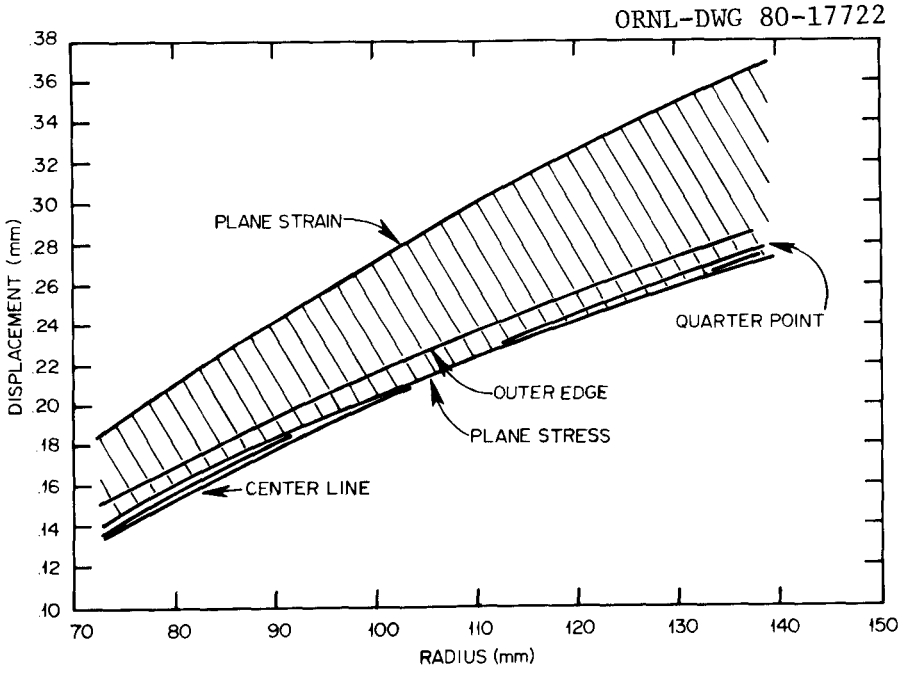

Fig. 5. Radial displacements computed for transformer coil with stainless steel outer covering.

(quarter point), and for a line on the outer edge. For the 2-D analysis, u for plane strain (w = 0) and for plane stress (w free to displace) is also shown in Fig. 5. The cross-hatched area represents the scatter band predicted by the two 2-D assumptions. In view of the uncertainty in Mylar properties, linear elasticity, and residual stresses, we consider this scatter band to be acceptable. The plane stress analysis predicts u better. Also, u is not greatly different at any particular axial location in the coil in the 2 1/2-D analysis. On the basis of displacements computed, the extra expense of a 2 1/2-D analysis is not justified.

The strains in finite-element analysis follow from the strain-displacement equations that, except for the hoop strain, involve derivatives of the displacements. Stresses are then computed from the strains and the elastic properties. In this analysis, as in most cases, the hoop stress is the predominant stress.

Figure 6 compares the hoop stresses computed by the 2 1/2-D and 2-D analyses. In the case of the Mylar, the hoop stresses are small (as expected for a low-modulus material) and are bounded well by the two 2-D analyses. The Mylar has compressive hoop stresses only. On the other hand,

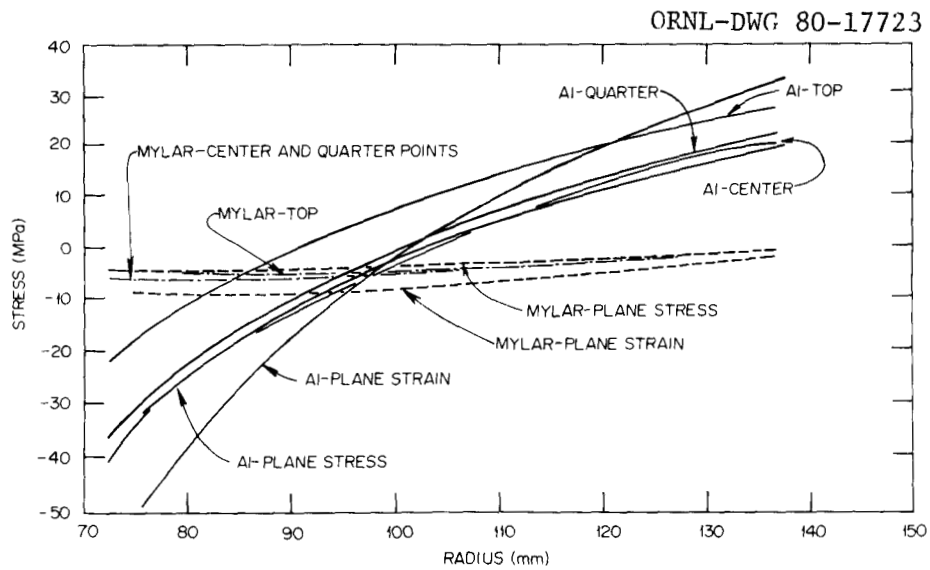

Fig. 6. Hoop stresses comparisons for transformer coil with stainless steel outer covering.

the hoop stresses in the aluminum are not bracketed by the plane stress and plane strain lines, particularly on the compressive side. However, the maximum values of hoop stress are overestimated by the plane strain analysis, and this is on the safe side for design work. We take the comparisons in Figs. 5 and 6 as evidence that the 2 1/2-D analysis is not really justified in subsequent work.

To clarify differences between the various approaches, the stresses in Fig. 6 do not include a large hoop stress in the outer stainless steel covering. Drawn to larger scale, Fig. 7 shows hoop-stress predictions from the plane stress and the plane strain approaches, including the outer covering. The stainless steel, with the thermal expansion coefficient mismatch and the higher modulus of elasticity, becomes highly stressed. The large stress in the stainless steel band has some limited effect on the rest of the structure.

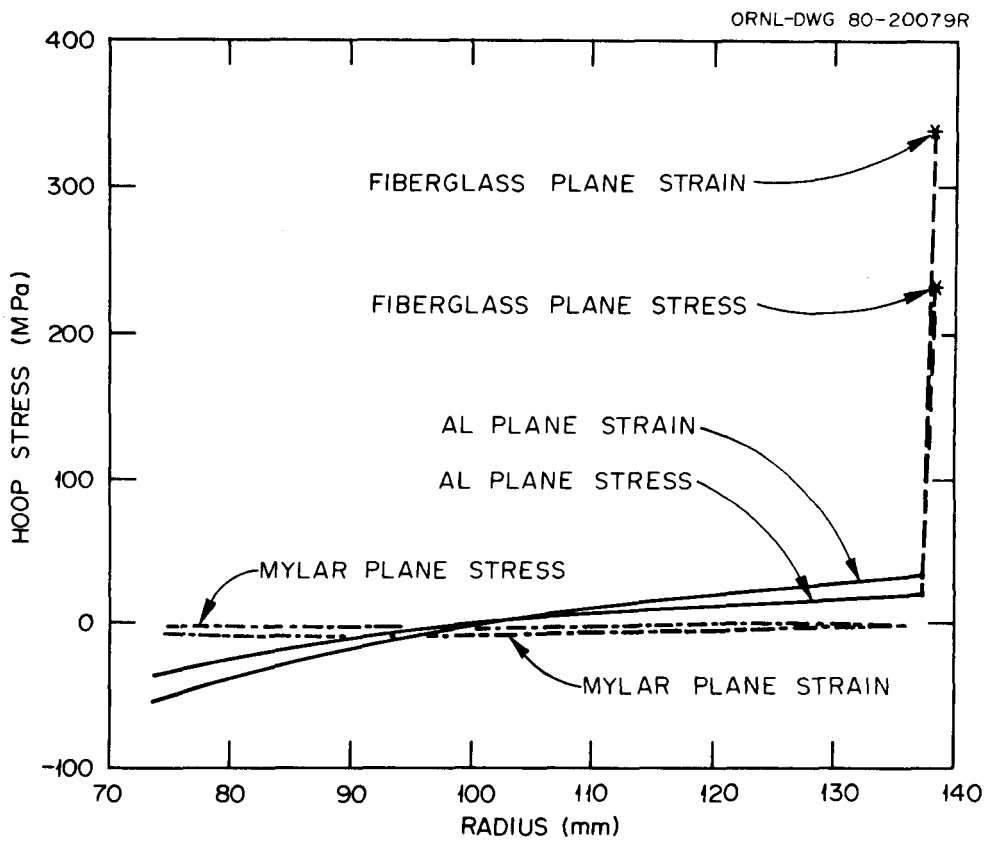

Fig. 7. Two-dimensional analysis hoop-stress predictions for coil with stainless steel outer covering included.

Another important result is the interlayer pressure distribution between the Mylar and aluminum foil. Figure 8 shows results from the 2-D analysis, where the pressure between layers is the negative of the radial stress. The lack of axial restraint in the plane stress analysis causes a slightly lower radial stress in the Mylar. With the axial restraint in plane strain, the radial stresses are essentially equal in the two materials. The plots in Fig. 8 can be assumed to bracket the radial stress distribution in the entire coil; the plane strain predictions correspond to the radial stresses in the interior of the coil, whereas those for plane stress correspond to the free surface of the coil.

The analysis in the preceding section for plane stress and plane strain conditions has been repeated with the stainless steel outer covering replaced by fiberglass.

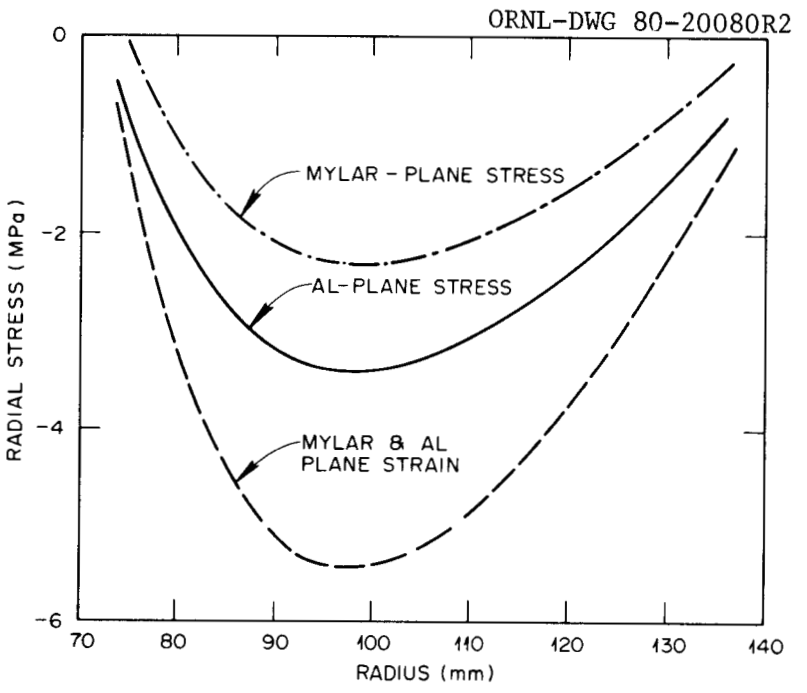

Fig. 8. Two-dimensional predictions for radial stress in coil covered with stainless steel.

Comparison of Fig. 9 with Fig. 5 does not show a significant difference in the predicted radial displacements at the inside of the coil. At the outside, however, the plane strain solution for the glass-covered coil shows slightly less displacement than for the stainless steel.

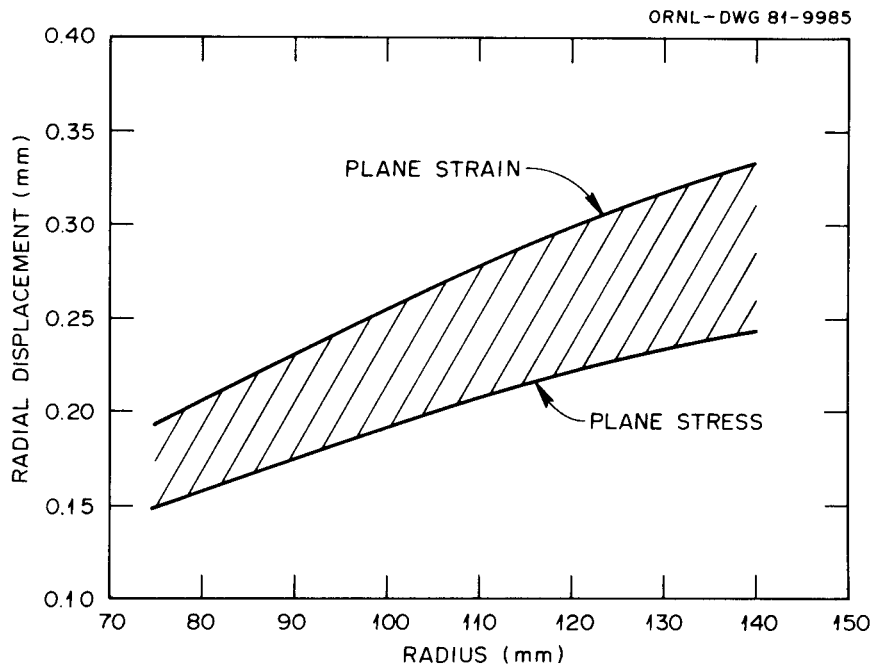

Fig. 9. Radial displacements computed for coil with fiberglass covering. Shaded area shows upper and lower bounds on the solution; lower bound is better predictor.

The hoop stresses in the aluminum layers of the coil with the fiberglass covering are somewhat less (Fig. 10 compared with Figs. 6 and 7). This is a favorable result, because the maximum hoop stresses in the aluminum are below the yield stress, which is taken as approximately 28 MPa (4 ksi). As before, a region of zero hoop stress is predicted at a radius of about 115 mm (4.5 in.). The hoop stresses in the Mylar are even smaller than in the previous analysis, being, for all practical purposes, zero.

Accompanying the decreased hoop stresses are decreased interlayer pressures. Figure 11 shows that this pressure has an upper-bound value of 4.3 MPa compared with 5.5 MPa (Fig. 8) where the outer covering is stainless steel. The maximum radial pressure occurs, as before, where the hoop stresses are essentially zero in the aluminum layers of the coil.

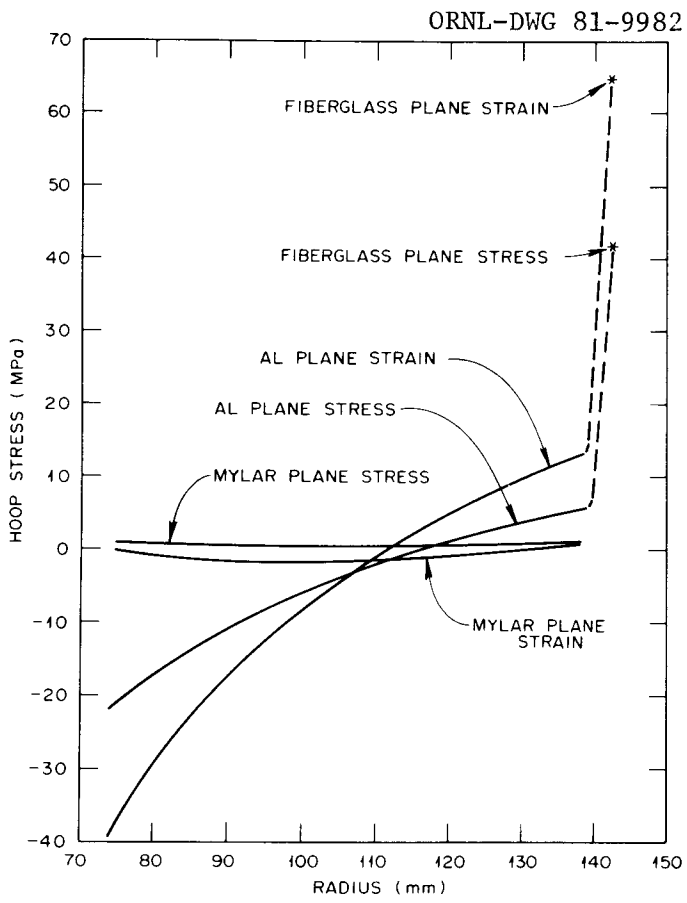

Fig. 10. Hoop stresses for coil with fiberglass covering. Plane-stress plot tends to be better predictor for compressive stress, whereas the plane-strain plot is better for tensile stress. When compared with Fig. 7, the hoop stresses here are much lower.

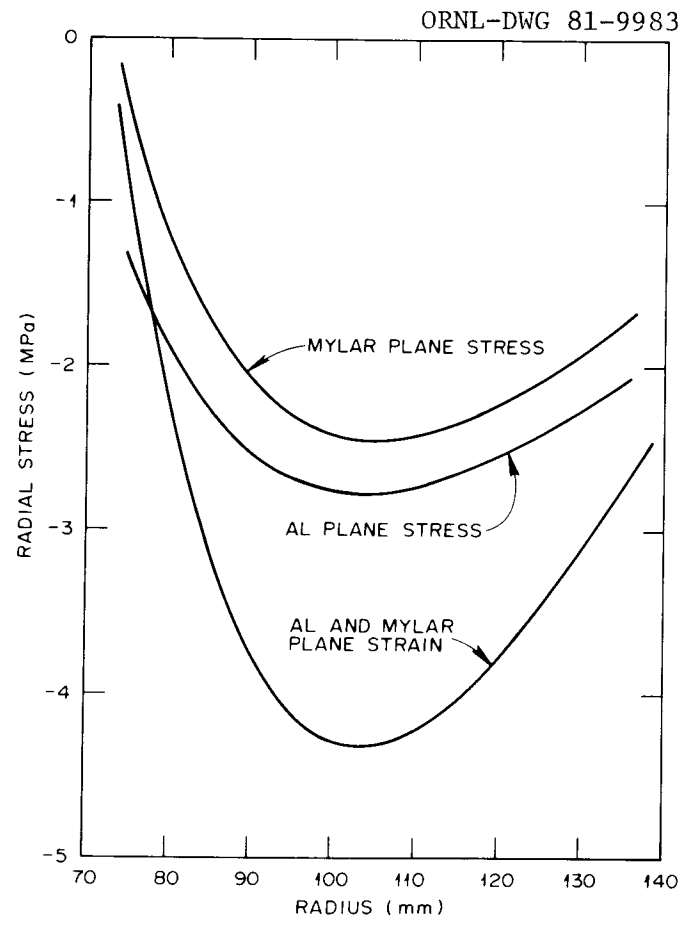

Fig. 11. Radial stress (interlayer pressure) for coil with fiberglass covering.

CONCLUSIONS AND RECOMMENDATIONS

These finite-element results should aid in the design of the transformer coils and in the placement of transducers for measurement of pressure and strain. The comparisons between the more complete 2 1/2-D analysis and the less expensive 2-D analyses verify the approach of bracketing the displacements and stresses with plane strain and plane stress results. The large stress in the stainless steel outer cover and the larger stresses in the coil indicate that he fiberglass covering is a better choice.

The results are limited by assumptions of isotropic mechanical properties for Mylar, by the assumption of linear elasticity, by the neglect of residual stress, and by the use of layers of aluminum and Mylar that are thicker than actual. For the results of this type of analysis to have a higher accuracy, better mechanical property data are needed. A more detailed and expensive inelastic analysis may be attempted at a future time when creep equations can be established for the Mylar-aluminum sandwich material. Creep testing of directionally oriented samples of Mylar could be attempted to improve property information and perhaps set the basis for a finite-element creep analysis.

A worthwhile project would be to correlate finite-element modeling (which predicts deformation versus time for a coil) with laboratory experiments. If confidence in such a model could be established, the changing of various design parameters could be economically evaluated by finite-element studies. Changes might include different operating temperatures, different thickness layers of Mylar and/or aluminum, changing the geometry of the coil, and substituting different materials. Such a procedure would aid in a more rapid cycle through design, testing, and production.

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REFERENCES

1. Program for Finite Element Calculations, Ltd., *PAFEC-75 Data Preparation*, Nottingham, England, 1978.
2. C. R. Brinkman, ORNL, letter 0903-183-80 to B. R. Dewey, 1980.

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