BOBBIN STRESSES GENERATED BY WIRE WINDING

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ABSTRACT

The prediction of bobbin stresses generated by wire winding is now possible by combining a finite element structural code and a rigid body motion code. In this combination of computer codes, the bobbin and the individual wire wraps are considered to be axisymmetric. Each wire wrap is modeled with a single one-node element that has stiffness, mass and radius. The distributed radial load that a wire wrap exerts on the bobbin or other wires is calculated by using a relationship developed for a thin ring with a circular cross section.

In this analysis, a layer of wire wraps with a specified tension is applied to a bobbin. The bobbin contracts radially until an equilibrium position is reached. When a second layer is added, the bobbin and each wire in the first layer reach a new equilibrium position. The tensions and the distributed radial loads associated with each displaced wire in the first layer change accordingly. As additional layers are added, the tensions and the distributed radial loads for all the previously applied wires are adjusted to reflect their new positions. The stresses in the bobbin can be determined for any number of wire layers. Bobbin fixturing during winding can be simulated by imposing suitable boundary conditions on the bobbin's finite element mesh.

A simple test problem is presented, providing a comparison between the finite element results and a closed-form solution. Quantitative results for bobbin stresses

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and wire tensions are then presented for more realistic coils. The impact of bobbin fixturing and wire packing structure on the stresses in the bobbin are also discussed.

NOMENCLATURE

A	=	area of a wire cross section, mm^2
B_r	=	inner bobbin radius, mm
E	=	elastic modulus, GPa
G	=	shear modulus, GPa
F	=	cross sectional shape factor
I	=	area moment of inertia of a wire cross section, mm^4
L	=	axial length of bobbin, mm
n	=	number of discrete radial loads
r	=	radius of a wire cross section, mm
R	=	radius to the centroid of a wire wrap, mm
R_o	=	initial equilibrium value of R, mm
ΔR	=	change in R, mm
T	=	tension in a wire wrap, N
t_b	=	bobbin base thickness, mm
t_f	=	bobbin flange thickness, mm
T_o	=	initial tension in a wire wrap, N
W	=	discrete radial load, N
θ	=	half-angle between discrete radial loads, deg
ω	=	distributed radial load, N/mm

INTRODUCTION

Wire winding can produce stresses high enough to cause excessive deformation or cracking in a bobbin. However, predicting the stresses in a bobbin is difficult for several reasons. When a wire wrap under tension is added to the bobbin and any previous wraps, it exerts a distributed radial load which must be supported. The tension and the radial load associated with a given wire wrap will change as additional wraps are added. The analysis is complicated by the fact that the bobbin contracts less in the radial direction at the flanges than it does midway between the flanges. The bobbin's flanges also deflect nonuniformly. Furthermore, each wire wrap is in contact with and influenced by several other wraps. In this study, these difficulties were overcome by modeling the wire winding process axisymetrically using a computer code which combines finite elements and discrete elements.

The finite element code PRONTO2D (1) was used to model the bobbin. Discrete elements, based on the DMC (2) code, were used to model the individual wires in the winding. PRONTO2D is an explicit dynamics code that uses four-node, quadrilateral elements with single-point integration and hourglass control. DMC is a two-dimensional rigid body motion code developed to analyze the interactions of spherical particles. A quasi static solution was obtained by slowly ramping on the wire tension and damping the dynamic response.

Each wire wrap in the winding was modeled using a single discrete element, which treats the wrap as a ring with a circular cross section. Although these cross sections remain circular, some overlapping is allowed, based on the elastic moduli of the elements in contact. The displacement of a given wire wrap is determined by its contacts with the other wire wraps, and possibly the bobbin. Wire overlaps and reaction forces are calculated based on a Hertz contact algorithm (2). It should be noted that the bobbin's contact surface is defined by a group of discrete, circular elements inscribed in each of the quadrilateral elements on the surface of the bobbin's mesh. Therefore, the bobbin must be meshed with small, square elements to approximate a flat bobbin surface.

WIRE WRAP MODEL

In this study, wire winding is simulated by applying thin, circular rings to an axisymmetric bobbin. Each ring, or wire wrap, lies in a plane perpendicular to the bobbin's axis of symmetry. This precludes the consideration of any helix or crossovers present in the winding. Figure 1 shows a circular ring subjected to several equal, discrete, radial loads. The relationship between the loads and the radial displacement at each load point is given in (3).

$$\Delta R = \left(\frac{k_1(\theta - sc)}{4s^2} + \frac{k_2 c}{2s} - \frac{k_2^2}{2\theta}\right)\frac{WR^3}{EI} \tag{1}$$

where $s = \sin\theta$ and $c = \cos\theta$. For thin rings,

$$k_1 = 1 - \frac{I}{AR^2} + \frac{FEI}{GAR^2} \tag{2}$$

$$k_2 = 1 - \frac{I}{AR^2} \tag{3}$$

where F is a shape factor dependent on the ring's cross section. For a distributed radial load (θ approaches zero), the k_1 term in Equation (1) disappears. As the number of concentrated loads, n, gets large, a distributed radial load can be approximated with

$$\omega' = \frac{nW}{2\pi R} = \frac{W}{2R\theta} \tag{4}$$

A true distributed load, ω, is obtained by taking the limit of ω' as θ approaches zero. When the appropriate substitutions are made for I and A, the distributed radial load in a thin ring with a cross sectional radius, r, simplifies to

$$\omega = \lim_{\theta \to 0}\left(\frac{W}{2R\theta}\right) = \frac{4\pi E r^2}{4R^2 - r^2}\Delta R \tag{5}$$

Allowing for multiple load steps,

$$\omega = \omega_0 + \frac{4\pi E r^2}{4R_0^2 - r^2}\Delta R \tag{6}$$

The radial load that a wire wrap exerts on the bobbin or other wraps that support it can be updated as the wrap's radius, R, changes using Equation (6). When a wire wrap is first applied, the distributed radial load is just the initial tension , T_O, divided by the radius, R. In the computer code, a wire wrap with a specified tension is applied to the bobbin and any previously applied wires. The specified tension is maintained in the wrap until the new structure reaches equilibrium and the current wrap radius, R, is stored. If more wire wraps are applied or if the boundary conditions of the bobbin change, R will change. The distributed radial load that the wire wrap exerts on the bobbin or other wraps also changes, in accordance with Equation (6).

TEST CASE

A test case was devised to demonstrate that the computer code correctly applies wire wraps and allows the wire tensions (and the associated distributed radial loads) to relax. Figure 2 shows the axisymmetric finite element mesh used for the test case. The centerline of the flangeless bobbin and each wire wrap were prevented from translating in the axial direction. The first wire wrap was applied to the bobbin and the structure was allowed to reach equilibrium. The rest of the wraps were then applied in succession, allowing the existing structure to reach equilibrium before the next wrap was applied. Each wire wrap is applied with an initial tension of 0.3 N. For comparison, an approximate closed-form solution was obtained in which the flangeless bobbin is treated as a thin cylinder subjected to a uniform pressure and the wire wraps are modeled as rings which cannot overlap the bobbin or each other. Elastic moduli of 13.86 GPa and 117 GPa were used for the bobbin and the wire wraps, respectively. See Figure 2 for geometric parameters.

The bobbin hoop stress calculated by the code varies from -28.8 MPa to -30.4 MPa . The closed-form solution yields a uniform hoop stress of -27.7 MPa. Figure 4 shows how the tensions in all five wraps are ramped up to the specified values and then relax as additional wraps are added. The timesteps shown are nonphysical. Note that the tension in each wire wrap is not quite zero before it is applied. At the beginning of each run, all wire wraps are assigned an initial tension equal to one-percent of the values with which they will eventually be applied. These small initial tension values allow wire wraps to migrate towards their eventual equilibrium positions before they are applied and reduces the computer run time. The closed-form tension values for the first wire wrap are shown in parentheses in Figure 3. The tension relaxation calculated by the code is within 10 percent of the closed-form solution. Some difference was expected for several reasons. The closed-form solution requires several idealizations. Additionally, in the code, each wrap exerts a small load on the structure before it is actually applied. Given the differences in the solution techniques, the differences in the results appear reasonable. More complicated windings will now be considered.

RESULTS

Combining PRONTO2D and DMC provides a versatile tool for evaluating wire winding problems. Material properties, bobbin geometry and fixturing, the size and number of wire wraps, and the initial wire tension can be arbitrarily defined. In order to focus on a few parameters, however, the same bobbin geometry, wire size, and material properties were used for all of the cases considered in this paper. Additionally, the wire wraps in a given layer were turned on as a group rather than individually to simplify the process and significantly reduce the run times. Figure 4 shows the axisymmetric finite element mesh of the bobbin that will be used in this study. A thickness of 0.3 mm was used for both t_b and t_f. The bobbin radius, B_r, was 5.7 mm. Elastic moduli of 13.86 GPa and 117 GPa were used for the bobbin and the wire wraps, respectively.

Two types of bobbin fixturing were considered. In the first case, the bobbin's flanges were supported with rigid, flat washers. The washers prevent axial displacement but do not affect radial contraction. In the second case, the bobbin's hub was supported at either end with rigid cones. The cones allow the hub ends to contract radially only if the bobbin contracts axially. For both types of fixturing, the supports were removed after all of the wire wraps had been applied . Figures 5 and 6 show the von Mises stresses in the bobbin and the tensions in the wires for the flange-supported and hub-supported cases, respectively. The initial tension in each wire was 1.2 N. The wire wraps in both windings were completely hexagonally packed. The only difference was the bobbin fixturing. Clearly, bobbin fixturing can have a significant impact on the stresses in the bobbin as well as the final wire tensions.

The flange supported case was run again with an initial wire wrap tension of 0.6 N. The stress and tension contours (scaled by 0.5) are quite similar to those shown in Figure 5. This basically linear relationship between the initial wire tension and the final bobbin stresses and wire tensions was expected since the bobbin deformations were quite small. Another variation on the flange supported case is shown in Figure 7. The only change was in the wire packing structure, which deviates from the purely hexagonal packing considered in the other cases. The stresses in the bobbin with the slightly disordered packing are slightly higher than those with a totally hexagonal packing.

Table 1 shows a few of the outputs available from the computer code. The total radial load, P_t, that a coil with t wire wraps exerts on a bobbin is simply

$$P_t = \sum_{i=1}^{t} (2\pi R_i)\omega_i = 2\pi \sum_{i=1}^{t} T_i \tag{7}$$

Several items stand out in Table 1. Doubling the initial wire tension roughly doubled the maximum bobbin stresses and deformations. The bobbin whose flanges were supported by flat washers had lower stresses and less change in axial length than did the bobbin whose hub was supported by cones. Finally, it is interesting to note that even though the disordered coil exerted a lower radial load on its bobbin than the purely hexagonal coil did, the disordered coil produced slightly higher bobbin stresses. This occurs because a hexagonal packing minimizes the lateral loads the wire wraps exert on the flanges. A coil with a random packing structure could cause considerably higher bobbin stresses than a hexagonally packed coil .

SUMMARY

Winding wire on a bobbin can be modeled axisymmetrically using a computer code which combines finite elements and discrete elements, allowing the user to arbitrarily define material properties, bobbin geometry and fixturing, the size and number of wire wraps, and the initial wire tension. The code updates the tension and the radial load associated with a given wire wrap as additional wraps are added or bobbin constraints are removed.

Although only one bobbin and wire size were considered, the results presented show that bobbin fixturing and the wire packing structure can have a significant effect on the bobbin stresses and dimensional changes. It is evident from this study that bobbin stresses and dimensional changes can be minimized by supporting the flanges during winding. A hexagonally packed coil will also yield lower bobbin stresses, when compared to a disordered coil.

REFERENCES

1. Taylor, L. and Flanagan, D., "PRONTO2D A Two-Dimensional Transient Solid Dynamics Program," SAND86-0594, March 1987, Sandia National Laboratories. Albuquerque, NM.

2. Taylor, L. and Preece, D., "DMC - A Rigid Body Motion Code for Determining the Interaction of Multiple Spherical Particles," SAND88-3482, June 1989, Sandia National Laboratories. Albuquerque, NM.

3. Young, W.C. and Roark, R.J. Roark's Formulas for Stress and Strain, 6th ed., McGraw-Hill, New York, 1989, pp. 262, 268.

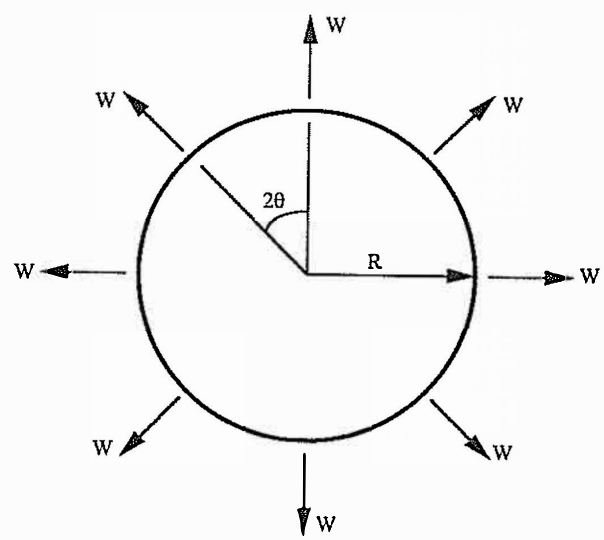


Figure 1. Circular Ring Subjected to Discrete, Radial Loads

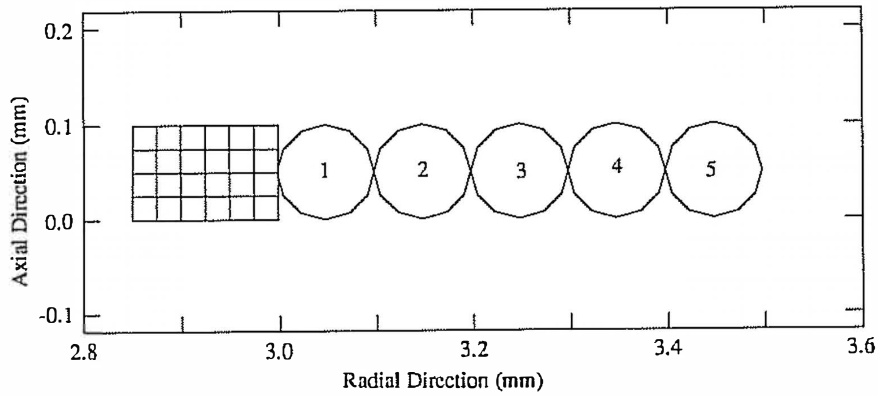
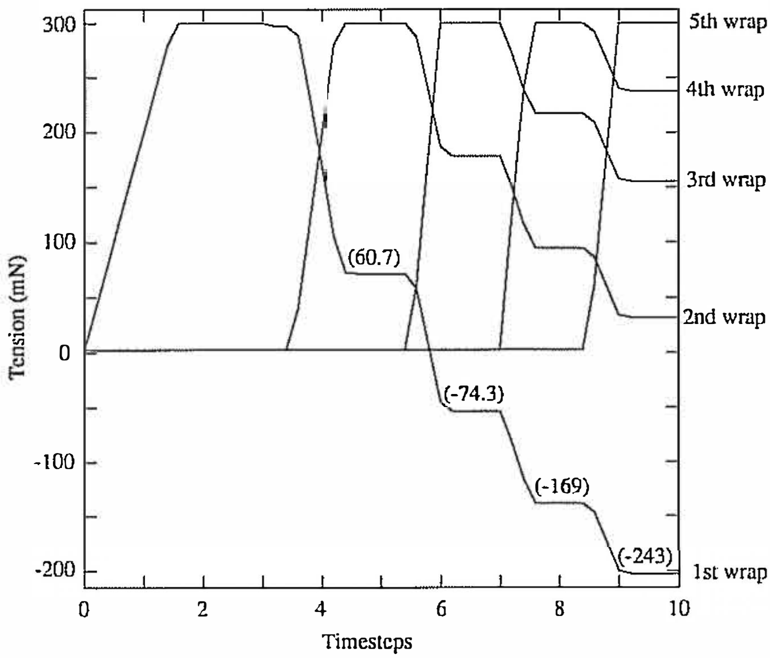


Figure 2. Finite Element Mesh for the Test Case



Note: Values shown in ( ) are closed-form solutions.

Figure 3. Computed Wire Tensions for Test Case

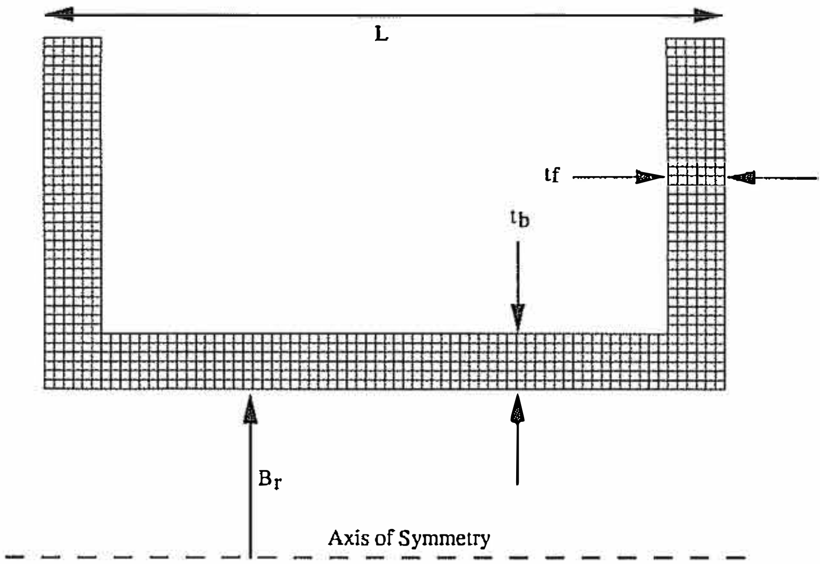


Figure 4. Bobbin Finite Element Mesh

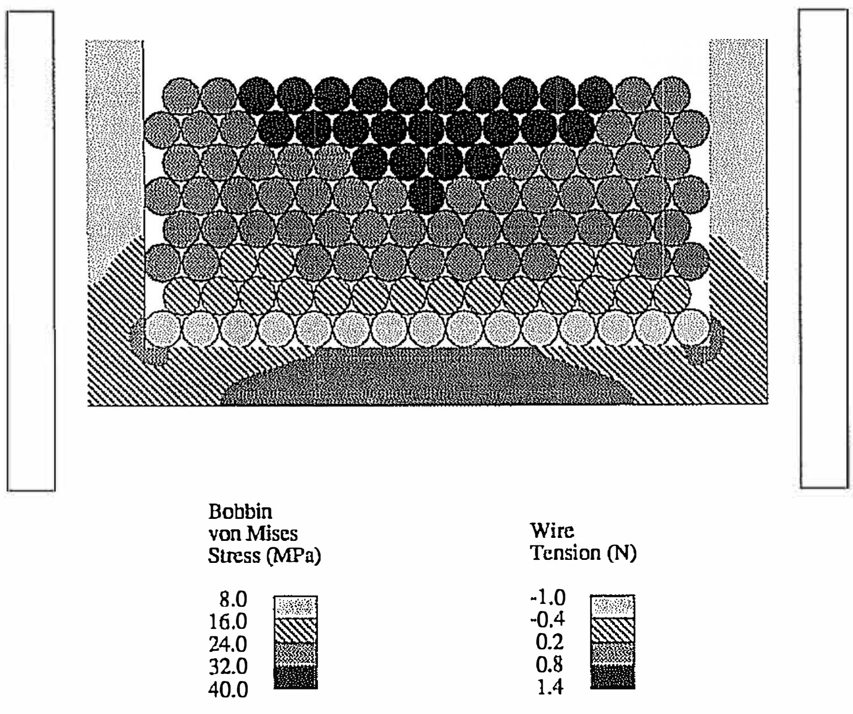


Figure 5. Flange Supported Bobbin with Hexagonal Wire Packing

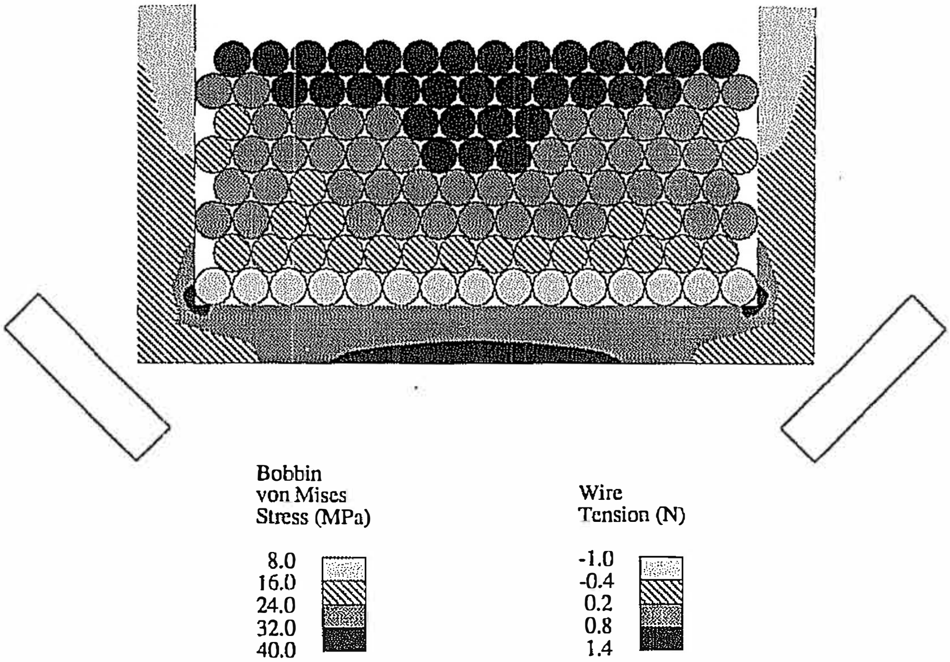


Figure 6. Cone Supported Bobbin with Hexagonal Wire Packing

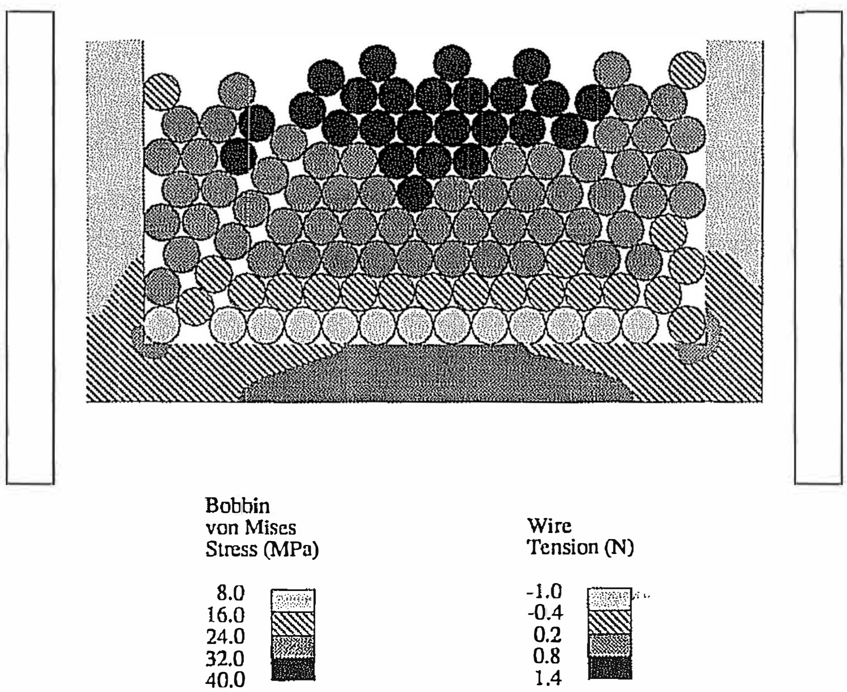


Figure 7. Flange Supported Bobbin with Disorderd Wire Packing

Case	Total Radial Load (N)	Maximum von Mises Stress in Bobbin (MPa)	Maximum Change in Axial Length (mm x 10^{-3})
Hexagonal Packing Flange Supports T_0=1.2 N	278	34.2	11.6
Hexagonal Packing Cone Supports T_0=1.2 N	309	41.1	19.8
Hexagonal Packing Flange Supports T_0=0.6 N	139	17.1	5.55
Disturbed Packing Flange Supports T_0=1.2 N	268	36.2	11.4

Table 1. Summary of Computer Code Outputs

BOBBIN STRESSES GENERATED BY WIRE WINDING
K. Metzinger

Would it be possible for you to taper the wire-winding tension?
Dave Pfeiffer, McGill

Yes. We apply each wire wrap as a ring. Each ring can have a different tension. We wouldn't even have to do it by layers. We could do it individually. That's not a problem. Well it's a problem if you've got to type them in, but it's not a problem for the code.

Maybe I should clarify it's a very generic technique. The finite element bobbin can be any size, any shape. We can use elastic properties, elastic plastic, we can use power law hardening. It's pretty flexible on what we can do. We have not explored all the possibilities.