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POLOIDAL FIELD SUPERCONDUCTING RING COIL CASE* AND SUPPORT STRUCTURE DESIGN

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ABSTRACT

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Conceptual design studies have been carried out at the Fusion Engineering Design Center (FEDC) on a succession of tokamak devices. In order to quickly assess the technical feasibility and to estimate the costs of competing poloidal field (PF) systems, it is desirable to develop simple, but accurate, rules for the structural design of these coils. In this paper we describe the rules developed for superconducting ring coils and illustrate their application to a specific example. This methodology may be easily adapted to resistive ring coil design.

INTRODUCTION

Since October 1979, conceptual design studies have been carried out at the FEDC on a succession of tokamak devices - the Engineering Test Facility (ETF) Designs 1 and 2, the International Tokamak Reactor (INTOR), and the Fusion Engineering Device (FED) baseline. During the course of these studies, simple though accurate computational structural models have been developed for poloidal field (PF) ring coils. Coils EF1, EF2, EF3, and EF4 of Fig. 1 (equilibrium field coils in the baseline PF configuration¹) are typical of such coils; these coils, although they carry very large MAT, are beam-like in proportion, with cross-section dimensions being small in comparison with coil diameter. In the FED baseline, ring coils internal to the toroidal field (TF) coils (coils EF1 and EF4 in Fig. 1) are copper coils, whereas those external to the TF coils (coils EF2 and EF3) are superconducting. The discussion in this paper is in the context of superconducting coils. The concepts described can, however, be applied to resistive coils as well, with the only difference being that the

Fig. 1. FED baseline PF system configuration.

individual turns in resistive coils can be designed to be free standing (i.e., the conductor is capable of equilibrating the magnetic running load) under the action of the magnetic loads whereas for a superconducting coil it is necessary to provide a structural case for the equilibration of magnetic running loads.

WINDING PACK LOAD PATHS

The superconductor for the FED baseline PF ring coils and the winding layout for coil EF2 are shown in Figs. 2 and 3. The conductor is a

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MASTER

Fig. 2. Poloidal field superconductor.

Fig. 3. Coil EF2 winding pack layout.

flat twisted cabled conductor with a stainless steel strap used to provide conductor with strength for handling and winding. The conductor is co-wound into pancakes with a thin sheet of G-10 to provide turn-to-turn insulation. The dimensions, number of turns, and number of pancakes will vary from coil to coil; Figs. 2 and 3 are typical ring coil concepts.

Axial loads on individual turns are statically determinate, being transmitted in bearing to the axial support plates (see Fig. 3) and then in bending of the support plates to the support blocks, which transmit the axial loads in bearing to the case.

The radial load transmitted to the case may also be determined from statics, with loads on individual turns being transmitted by compression of the winding and accumulating from turn to turn with the total radial force on each pancake transmitted to the case at the outermost turn (or innermost turn, if the radial load is directed inwards) of the pancake. This can be shown by a simple load deflection analysis of the radial loading on a pancake. Consider a radial force balance on the k th strap in a pancake (Fig. 4).

Fig. 4. Radial force balance on strap.

The strap of thickness t_s and height w has a hoop force $\sigma_\theta w t_s$ acting on each end, radial pressures p_i and p_o on its inner and outer faces, respectively, due to compression of the superconducting cables between straps (with the strap being relatively rigid in radial compression compared with the cables), and a radial magnetic

running load f_r on the strap. The hoop stress σ_θ and pressures p_i and p_o are related to deflections of the k th strap u_k and of its neighbors to the inside and outside by

$$p_o = E_c \frac{u_k - u_{k+1}}{t_c} ,$$

$$p_i = E_c \frac{u_{k-1} - u_k}{t_c} ,$$

$$\sigma_\theta = E_s \frac{u_k}{R} ,$$

where E_s is the modulus of the strap and E_c and t_c are the modulus in compression and radial thickness, respectively, of the conductor layer between straps. A radial force balance on the elemental length $R d\theta$ produces, after rearrangement, the first-order difference equation:

$$u_{k+1} - 2 \left(1 + \frac{t_s t_c E_s}{2R^2 E_c} \right) u_k + u_{k-1} = - \frac{f_r t_c}{E_c w} . \quad (1)$$

For a typical practical example (ring coil EF2), the second term in parentheses is of order 10^{-5} and may be neglected in comparison to unity. With this simplification, Eq. (1) has the solution

$$u_k = \frac{t_c f_r}{2E_c w} \left[n_L (n_L + 1) - k(k - 1) \right] ,$$

where n_L is the number of layers in a pancake and the boundary conditions are $p_i = 0$ on the first turn ($u_0 = u_1$) and $u_{n_L + 1} = 0$ (the case is rigid). The pressure transmitted to the case will be p_o at $k = n_L$ or

$$p_o (k = n_L) = \frac{n_L f_r}{w} \quad (2)$$

The pressure [Eq. (2)] is simply the solution that would have been obtained if curvature had been neglected, with the radial force on the turn being equilibrated by the differential pressure $p_o - p_i$ across each strap.

FORCE DISTRIBUTION

The radial (in-plane) and axial (out-of-plane) magnetic running loads on a ring coil may be written

$$F_r(\theta) = F_{ro} + \bar{F}_r(\theta) , \quad (3)$$

$$F_z(\theta) = F_{zo} + \bar{F}_z(\theta) ,$$

where F_{ro} and F_{zo} are the axisymmetric components and $\bar{F}_r(\theta)$ and $\bar{F}_z(\theta)$ are the components which vary with circumferential angle θ . F_{ro} and F_{zo} are caused by the interaction of the magnetic field produced by the other poloidal field coils with the current in the ring coil, whereas $\bar{F}_r(\theta)$ and $\bar{F}_z(\theta)$ are attributable to the interaction of the fringing of the toroidal field between the discretely spaced TF coils with the ring coil current. The interaction of the self-field of the coil itself with its own current produces clamping forces that are self-equilibrating over the cross-section of the coil and that therefore produce no stresses in the coil case and are not specifically dealt with here.

Because of the symmetry of a tokamak, $\bar{F}_r(\theta)$ and $\bar{F}_z(\theta)$ are odd, periodic functions of θ , with the period equal to the spacing of the TF coils. In the practical configurations considered at the FEDC, it has been observed that

$$F_r = F_{ro} + F_r^* \sin N\theta , \quad (4)$$

$$F_z = F_{zo} + F_z^* \sin N\theta ,$$

are satisfactorily accurate approximations to the actual running load distributions [Eq. (3)], where N is the number of TF coils. In Eqs. (3) and (4), radial running loads are positive when directed radially outward and the axial running loads are positive upward.

RING COIL SUPPORT

At the FEDC, we have chosen a ring coil support scheme designed to structurally decouple the ring coils from the TF system as much as is practical. The ring coils are supported from the TF system by supports located at each TF coil location that are free to slide radially. Consequently, the running load components F_{ro}^* , F_r^* , and F_z^* are equilibrated by dilation and bending of the coil and transmit no force to the TF system at the supports. The running load component F_{zo} is not self-equilibrating in character; unlike the other force components, F_{zo} produces a net force when integrated over the coil. Static equilibrium requires that the force component F_{zo} be equilibrated by the supports.

COIL STRESS ANALYSIS

The axisymmetric component F_{ro} of the radial running load causes uniform dilation of the coil, which produces a hoop membrane stress

$$\sigma_{\theta m} = F_{ro}/A, \quad (5)$$

where A is the coil structural cross-sectional area. The conductor concept adopted for the tokamak studies at the FEDC is a flat cabled conductor with the cable twisted on a stainless steel strap to provide a conductor strength sufficient for handling and winding under tension. It has been assumed that the cables themselves provide no hoop strength to the conductor, so the area A is the sum of the stainless steel case area and the area of the stainless steel straps in the n turns in the winding.

The circumferentially varying component $F_r \sin N\theta$ of the radial running loads causes bending of the coil with no dilation (neglecting axial shortening due to bending). By symmetry of loading, the radial deflection and bending moment due to this component of load will be zero at the locations of the TF coils (the support points).

Evaluating the arc of radius R and included angle $2\pi/N$ between adjacent coil supports as a straight beam^a of length $L = 2\pi R/N$, the bending moment will be

$$M(x) = \frac{F_r R^2}{N^2} \sin \frac{2\pi x}{L}, \quad (6)$$

$$M_{\max} = \frac{F_r R^2}{N^2}.$$

Similarly, the maximum moment produced by the circumferentially varying component $F_z \sin N\theta$ of the axial load will be $F_z R^2/N^2$. Superposing the hoop membrane stress and the two components of hoop bending stress results in the following equation for maximum stress:

$$\sigma_{\theta, \max} = \frac{F_{ro}}{A} + \frac{R^2}{N^2} \left(\frac{|F_r^*| C_1}{I_1} + \frac{|F_z^*| C_2}{I_2} \right), \quad (7)$$

where C_1 , I_1 and C_2 , I_2 are the coil case section properties for bending due to F_r^* and F_z^* , respectively.

The maximum stress [Eq. (7)] occurs at the 1/4 and 3/4 points of the span of coil between the TF coils. At these locations, the contribution to hoop bending from the force component

^aNote: The straight beam results [Eqs. (6) and (7)] are accurate to within about 1.5% for eight or more TF coils.

F_{zo} is small (the bending moment due to F_{zo} passes through maxima at the support points and midspan) and is neglected.

APPLICATION TO FED BASELINE

The equations developed above are applied to ring coil EF2 of the FED baseline as an example. The conductor and winding pack details for this application are illustrated in Figs. 2 and 3; the structural case is detailed in Fig. 5.

Fig. 5. Coil EF2 case cross section.

The allowable stresses are tabulated below and are based on Refs. 2 (metallic structures), 3 (insulating materials), and 4 (conductor). The allowable stresses reflect the intended cyclic operation of FED, which results in 5.5×10^5 full load cycles of ring coil operation (see Table 1).

Table 1. Allowable design stresses for ring coil components

Component	Loading	Allowable stress MPa (ksi)
Conductor	Radial bearing	34 (5)
G-10 support structure	Flexure in plates	180 (27)
	Bearing in rings	152 (23)
Stainless steel case	Flexure	138 (20)
	Shear	69 (10)

The magnetic running loads and sinusoidal approximations are plotted in Fig. 6. Structural analysis of the sample coil addresses the conductor, the insulating support structure, and the structural case.

Conductor

The conductor chosen for the FED baseline (Fig. 2) is the 50-kA conductor developed by Los Alamos National Laboratory (LANL) for the 50-MJ pulsed coil program. It is designed to operate at 8 T, which is also the peak field in the FED application. The axial running load will, therefore, not overstress the conductor. Radial running loads, on the other hand, accumulate, and the design qualification for the 50-MJ pulsed coil program does not qualify the conductor for FED application in this regard. The maximum amplitude of the radial load per pancake

Fig. 6. Magnetic running loads on coil EF2.

will be (see Fig. 6) $6.4875/4 \text{ MN/m} = 1.7993 \text{ MN/m}$; this results in a maximum radial pressure of 14.4 MPa, which is less than the allowable 34 MPa shown in Table 1.

Insulating Support Structure

Axial load is transmitted in bending of the axial support plates, which are treated as being simply supported by the support blocks. Although the bending moment is maximum at the midspan, the sloping helium channels cause the maximum bending stress to be located at points approximately 3/16 of the unsupported span from midspan toward the support blocks on each side. The maximum value of the axial running load is 3.5984 MN/m, resulting in a maximum bending stress of 191 MPa. This represents a slight overstress compared to the allowable 186 MPa in Table 1, but for conceptual design, this was considered acceptable. The maximum bearing stress in the support blocks is only 4.3 MPa (compared to the 158 MPa allowable).

Structural Case

The axial running load is transmitted by shear into the case top or bottom walls, and the radial running load must be borne in bending of the sidewall. At the location of maximum axial running load (3.9622 MN/m), the shear stress in the case is 26 MPa; by comparison with the limits described in Table 1, this is acceptable. The maximum radial running load is 6.5639 MN/m. The sidewalls on the inside and outside of the winding are initially sized to be strong enough to carry the maximum running load in bending, when evaluated as simply supported. To remain within the limit of 138 MPa (Table 1) for flexure in the case would require a sidewall thickness of nearly 17 cm, which is deemed to be excessive for ease of fabrication. Consequently, the configuration in Fig. 5 has been adopted for the coil case with box beam extensions at the inside and outside of the winding; 2.5-cm gussets spaced at 10 cm center-to-center tie the sides of the box extensions together. With the boxlike extensions, it is found that plate thickness of 2.5 cm is adequate with gussets 12.5 cm deep in order to equilibrate the radial running load in sidewall bending. In practice, the cover plates are 3.5 cm thick to provide sufficient section in the coil case cross section for bending of the coil as a whole.

For the dimensions of Fig. 5,

$$A = 0.2698 \text{ m}^2 \text{ (includes steel straps in conductor)}$$

$$I_1 = 4.281 \times 10^{-2} \text{ m}^4 \quad I_2 = 1.945 \times 10^{-2} \text{ m}^4$$

$$C_1 = 0.5790 \text{ m} \quad C_2 = 0.4291 \text{ m}$$

Then, with $F_{r0} = 2.7459 \text{ MN/m}$, $F_r^* = -3.8180 \text{ MN/m}$, and $F_z^* = 3.3963 \text{ MN/m}$, the application of Eq. (7) provides $\sigma_{\theta, \max} = 124 \text{ MPa}$, which is less than the allowable 138 MPa.

SUMMARY

Because of the symmetry existing in a tokamak, the radial and axial magnetic running loads are odd, periodic positions of azimuthal angle θ , which may be approximated as the superposition of a uniform component and a sinusoidally varying component, Eqs. (4). Coils are supported so that the uniform component of radial (in-plane) force is carried by the case and the steel in the conductors by hoop membrane action, whereas the sinusoidal components of running load are equilibrated by case bending. Superposition of the stresses for these three loadings produces the Eq. (7) expression for maximum hoop stress, which may be used to size the coil cases. In addition to ensuring that

the hoop stress is less than allowable, the design must be checked to verify that the coil case sidewalls can carry the maximum radial running load in plate bending and that the top and bottom walls of the case can carry the axial load in shear. As part of the structural evaluation of a particular coil, it must also be verified that there is a load path through the winding capable of transmitting loads on individual turns to the case. For the FED baseline design concept, radial loads accumulate from turn to turn and are finally borne by the outer wall of the case (or inner wall if the radial force is directed inwards). Axial loads between each pancake.

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