A Model Dipole for FAIR SIS300: 3D Design of the Mechanical Structure

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Abstract—The FAIR facility, under development at GSI, includes the synchrotron SIS300 (300 Tm rigidity). In order to reach the required high intensities of proton and heavy ion beams, the bending dipole magnets have to be pulsed from the injection magnetic field of 1.5 T up to 4.5 T maximum field at the rate of 1 T/s. These 7.8 m long magnets have cos-theta shaped coils with a 100 mm bore with the particular characteristic to be geometrically curved (the sagitta is 114 mm). The 2D mechanical cross section has been designed in detail: it consists of 3 mm thick laminated stainless steel collars, assembled through keys, and 1 mm thick iron yoke laminations, assembled through large stainless steel C-shaped clamps. This paper analyses some aspects that are intrinsically 3-dimensional: the mechanical behavior of the ends, the effect of the longitudinal pre-stress, the design of the external flange and outer shell.

Index Terms—Pulsed dipole, SIS300, 3D mechanical analysis.

I. INTRODUCTION

T HE SIS300 synchrotron of the new FAIR facility [1] at GSI (Darmstadt, Germany) calls for the development of fast cycled superconducting dipoles, to be operated at 1 T/s between 1.5 and 4.5 T in a 100 mm bore. They are 7.8 m long and have a curvature radius of 66.67 m, corresponding to a sagitta of 114 mm. In collaboration with GSI, INFN (Genoa, Milan and Frascati laboratory) has completed the design and the construction of a 3.8-m long model of one lattice dipole, which took place at the ASG-Superconductors premises.

The main coil characteristics can be found in details in [2]–[5]. Complete 2D and 3D electromagnetic analyses ([6], [7]) have been carried out to obtain a cos-theta configuration with good field quality and maximum temperature margin. The cross-section has been finalized through very detailed 2D finite element (FE) analyses [8] of the dipole mechanical structure. It is based on 3 mm thick laminated stainless steel collars, assembled through keys, and 1 mm thick iron yoke laminations, assembled through large stainless steel C-shaped clamps. The main mechanical function of the collars is to give axial pre-stress to the winding. During energization up to 4.5 T, the winding experiences very large Lorentz forces, 0.68 MN/m

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Fig. 1. 2D FE model in (left) open and (right) assembled configuration. Windings, coil wedges, collars, and iron yoke are clearly observable.

radially outward and 0.30 MN/m axially inward. The radial component of the Lorentz force is mechanically sustained by collars, the axial component has to be compensated by an adequate axial pre-stress in the winding (around 70 MPa at the end of the assembly operations). A self-sustained collared coil design has been abandoned in favor of giving the iron yoke a limited mechanical role: it has been designed to give no further pre-stress to the winding but to limit the deformations arising during the magnetic energization.

II. 3D ANALYSIS OF THE DIPOLE MECHANICAL STRUCTURE

Due to the high grade of complexity of the system, the 3D model cannot be as detailed as the 2D one. In particular, the laminations, the collaring procedure and the assembly of the iron yoke have to be in some way approximated. In facts, the 2D model describes very precisely the assembly operations: the laminations are represented performing the calculation in plane stress with the correct thickness assigned to each component, the collar keys and the C-clamps are modeled as designed, contact elements describe the sliding of each component with respect to the others. The 2D FE model in open and assembled configuration is shown in Fig. 1. The 3D model has to be much simpler. The lamination effects have been replaced by orthotropic Young moduli, having equivalent values keeping into account the lamination in the longitudinal direction. Collar keys and C-clamps are not modeled. We took advantage of the results coming from the 2D analysis and simulated the effect of collar keys and C-clamps by imposing to the 3D model (see Fig. 2) the previously calculated axial distances between collars and yokes listed in Table I.

The material properties used in the FE analyses are described in detail in [8].

The main results of this first 3D analysis were the finding of a huge peak stress located at the beginning of the coil end on

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Fig. 2. 1/8 symmetry 3D mechanical model of the SIS300 dipole. Windings, coil wedges, end spacers, collars, and iron yoke are clearly observable.

TABLE I AXIAL DISTANCE BETWEEN COLLARS AND YOKES AS FUNCTION OF THE APPLIED LOADSTEPS

loadsteps	axial distance between collars (µm)	axial distance between yokes (µm)
collars under pressure	0	-
collar pressure release	80	-
iron yoke under pressure	40	0
yoke pressure release	70	200
cool-down	70	55
energization	70	40

the turn nearest to the pole, as shown in Fig. 3, and a substantial absence of pre-stress in the coil end. Both these problems can be addressed with an adequate usage of shimming. In particular, the insertion of a tapered shimming nearly 200 mm long between the collar pole and the winding last turn actually removes the stress peak. One side of this shimming, opposite to the coil end, is 0.2 mm thick and it grows gradually as thin as possible towards the region where the peak is located. It is possible to profit of the same mechanism to give some azimuthal pre-stress to the coil ends. In particular, a shimming tapered from 0.1 mm to around 1 mm inserted on the midplane of two opposite coil ends can give a reasonable azimuthal pre-stress distribution (around 20 MPa).

The effects of the curvature have been neglected. As confirmed during the construction, the curvature does not affect the general mechanics of the system, if we except the actions taken to keep the curvature itself within the designed tolerance.

The goodness of this model can be assessed by comparing the resulting stresses of the 2D and the 3D model far from the coil end, on the dipole longitudinal symmetry plane. The results as function of the loadsteps are summarized in Table II and show a good agreement between the two models.

Last information, important for the next analysis, concerns the total lengthening of the winding at the end of the assembly operations. In Table III the longitudinal lengthening of 1-m long winding for two values of the friction coefficient are shown. There is a non-negligible influence of the friction coefficient on those results, which will make quite uncertain the *a priori*



Fig. 3. Von Mises stress [MPa] in the windings with collars under pressure.

TABLE II Resulting 2D and 3D Average Von Mises and Azimuthal Stress on the Dipole Longitudinal Symmetry Plane

loadsteps	azimuthal stress (MPa)		Von Mises stress (MPa)	
	2D	3D	2D	3D
collars under pressure	-78	-78	73	62
collar pressure release	-64	-72	61	58
iron yoke under pressure	-78	-86	73	69
yoke pressure release	-68	-77	64	62
cool-down	-38	-41	39	41
energization	-43	-41	42	39

TABLE III Longitudinal Lengthening of the Winding (Undeformed Half Length 500 mm)

landstons	$\Delta \ell \ (mm)$	Δℓ (mm)
loausteps	$\mu_r = 0.5$	$\mu_r = 0.1$
collars under pressure	+0.19	+0.28
collar pressure release	+0.18	+0.26
iron yoke under pressure	+0.27	+0.38
yoke pressure release	+0.24	+0.32
cool-down	-1.13	-1.21
energization	-1.07	-1.02

determination of the longitudinal pre-stress, as it will be described in the next paragraph. Moreover, even if the cumulative effect, at full energization, seems not to be so important, there is a great variation of the elongation due to Lorentz forces between cool-down and energization: almost negligible (0.06 mm) if the friction coefficient corresponds to 0.5, quite significant (0.19 mm) if the friction coefficient corresponds to 0.1.

III. 3D MECHANICAL ANALYSIS OF THE EXTERNAL FLANGE AND OUTER SHELL

A. Finite Element Model

The FE model built to perform the 3D analysis of the complete mechanical structure of the dipole is shown in Fig. 4. The basic principle is that the thick end flange, welded to the outer shell, is mechanically connected to the winding through four



Fig. 4. 3D FE model of the complete dipole mechanical structure.

screwed bolts per side. The longitudinal pre-stress is determined by the length of engagement of the bolts themselves, which, in the model, is obtained through a fictitious thermal expansion. In order to avoid the coil ends being in tension, we need to apply a longitudinal pre-stress able to counter balance the longitudinal Lorentz force, once the magnet is energized. The value of this force is 100 kN [8] and tends to stretch out the coil ends one with respect to the other. Due to the complexity of the system, the winding cannot be represented with the same level of details as the previous 3D mechanical analysis, but we need to introduce some approximations. In particular, winding, collars and iron yoke are represented through concentric cylindrical shells. Even if in this way there cannot be any axially or radially transferred pressure, this model can account for the longitudinal behavior of the whole system.

B. Longitudinal Pre-Stress Definition

In order to understand the results coming from the FE analysis, it is important to give a correct description of the mechanical system we are modeling. It can be considered as composed by two elastic constants, k_B representing winding, collars and iron yoke, and k_S , representing outer shell and thick end flange, mechanically connected in series through the bolts, whose elongation $x = x_B + x_s$ is a constant defining the applied pre-stress. The longitudinal pre-stress P is determined, before the energization of the magnet, following:

$$\begin{cases} P = k_B x_B = k_S x_S \\ x = x_B + x_S \end{cases} \Rightarrow P = \frac{k_B k_S}{k_B + k_S} x \tag{1}$$

When the magnet is energized, and thus the Lorentz force becomes active, the new equilibrium equations are:

$$\begin{cases} T - F_L = k_B x'_B \\ T = k_S x'_S \\ x = x'_B + x'_S \end{cases} \Rightarrow T = \frac{k_S}{k_B + k_S} F_L + P \qquad (2)$$

where T is the reaction force transferred through the bolts. The bolt elongation variation is negligible due to their small length. The main result is that only a fraction of the Lorentz force contributes to the total reaction acting on the flanges. It is impor-

tant to underline that this effect, which is often erroneously attributed to friction, is instead only a result of the balance of elastic forces.

Moreover, (2) is valid only for a concentrated force. In fact, a sample of length L and elastic constant K subjected to a force F perfectly distributed along its length will experience a total contraction Δl given by:

$$\Delta l = \int_{0}^{L} \left(\frac{1}{LK} \int_{y}^{L} \frac{F}{L} dy' \right) dy = \frac{F}{2K}$$
(3)

that is, a perfectly distributed force acts as a concentrated force of half value, or better the constraining reaction due to the application of a perfectly distributed force is half of the applied force itself. This point is important because in our case the coil end is represented as a cylindrical shell, so that the longitudinal Lorentz force in the ends cannot be correctly distributed. Nevertheless, the model described in the previous paragraph can give us the missing information: the ratio between the total reaction in the coil end and the applied force, precisely distributed along the modeled winding, is around 40%, so that a perfect distribution of the Lorentz force in the cylindrical shell representing the coil end is a conservative way out.

Another important aspect of this 3D model is the way winding, collars and iron yoke are connected together. In facts, by definition concentric cylindrical shells cannot generate a reaction force at the interfaces, so that it is totally useless the definition of friction coefficients, the resulting friction force being null, as proportional exactly to that normal reaction. Therefore, only two configurations are possible: either perfect gluing or sliding without friction. The part of the winding at constant cross-section is axially pre-stressed at 70 MPa by collars, with a limited contribution of the iron yoke, both laminated, so it appears implausible a major relative slipping. For that reason, they can be represented as perfectly glued. On the other side, the coil ends are not considerably pre-stressed, only the presence of the tapered shim loads the ends azimuthally up to around 20 MPa. Then, in these regions there can be significant sliding. The best representation for the coil ends is therefore sliding without friction inside the collars (this representation is conservative, resulting in the maximum reaction due to the Lorentz force). The results is that the fraction of the Lorentz force converting into longitudinal reaction is around 30 kN, out of the total longitudinal Lorentz force in each end being 100 kN, outward. Since this number has been determined through conservative assumptions, we only need the longitudinal pre-stress to be $P \gtrsim 30$ kN. A meaningful value is 50 kN, which corresponds to a length of engagement of the bolts of 0.5 mm.

C. Evaluation of the Effects of Thermal Contractions

The effect of the differential longitudinal thermal contractions on the complete mechanical system is important, as the outer shell and the iron yoke inside of it are fully mechanically decoupled. Whilst the thermal contraction of the system composed by the outer shell and the thick end flange is quite easy to be determined, the behavior of the assembly made by collared winding and iron yoke is more uncertain and depends on unknown parameters. In particular, it is the friction at the inter-



Fig. 5. Schematization of the mechanical system. Winding, collars, and iron yoke are represented by the elastic constant k_B and outer shell and thick end flange by the elastic constant k_S . *P* is the applied longitudinal pre-stress, and F_L is the Lorentz force.



Fig. 6. Von Mises stress [MPa] after energization in the thick end flange if the equivalent thermal coefficient of the system made by collared coils and iron yoke were $\alpha_B = 2.1 \cdot 10^{-3}$. Deformation are 100 times amplified.

faces winding/collars and collars/iron yoke which, influencing the shortening of the winding between assembly and cool-down as shown in Table III, most determines the thermal behavior of that set. Nevertheless, we can identify the two extreme conditions and find a way to tune the available parameters to be equidistant from them. We can restrict the unknowns to the sole interface between yoke and collars; in facts, due to the quite high axial pre-stress given by the collars, the winding/collars interface can reasonably be modeled as glued. The system can be again schematized as shown in Fig. 5, but in this case the pre-stress is given by the integral differential thermal contractions of the two springs: $\alpha_S = 2.8 \cdot 10^{-3}$ and α_B , comprises between the collar coefficient $\alpha_C = 2.4 \cdot 10^{-3}$ (if the friction between collars and yoke is negligible) and the iron coefficient $\alpha_Y = 1.8 \cdot 10^{-3}$ (if the friction between collars and yoke is dominating). Calculations shown in Section II can help us in determining more precisely the equivalent thermal contraction of the system made by collared coils and iron yoke in these two extreme cases, finding $2.1 \cdot 10^{-3} < \alpha_B < 2.4 \cdot 10^{-3}$. Very roughly, it follows that the total length of engagement of the bolts is comprises between $(2.8 \cdot 10^{-3} - 2.4 \cdot 10^{-3}) \cdot 2 \text{ m} = 0.8 \text{ mm}$ and $(2.8 \cdot 10^{-3} - 2.1 \cdot 10^{-3}) \cdot 2 \text{ m} = 1.4 \text{ mm}$ (2 m is nearly the half length of the magnet). It is evident that in both cases the length of engagement is larger than the desired value we found in the previous paragraph (0.5 mm), leading in the worst case to a huge pre-stress, as high as 130 kN at cold and 130 + 30 = 160 kN after energization, and then to a potentially dangerous stress on the end flange, as shown in Fig. 6. In facts, given the rough

approximations we made in this modelization, we conservatively decided to work well below the usual 2/3 of the yield strength, which for the end flange steel at cold is 600 MPa. Since we do not have any argument to decide what the case of our system is, the best solution is to impose intermediate conditions. A gap between the bolts and the winding end flange at room temperature of about 0.4 mm, subtracting from the total length of engagement at cold, seems to be a suitable solution to this problem. In this way, the two extreme cases are both acceptable: if $\alpha_B = 2.4 \cdot 10^{-3}$, the total length of engagement becomes 0.4 mm, leading to a longitudinal pre-stress at low temperature of 37 kN, which is not the optimal 50 kN, but it is still larger than the Lorentz force, as required. If $\alpha_B = 2.1 \cdot 10^{-3}$, the total length of engagement becomes 1 mm, leading to a longitudinal pre-stress of 92 kN, and a peak Von Mises stress on the thick end flange after energization of 260 MPa, still very high but acceptable.

IV. CONCLUSION

Complete 3D finite element analyses of the SIS300 dipole mechanical structure have been carried out and the issues not covered by the previous 2D analyses have been finalized. In Particular, the longitudinal effects of pre-stress, cool-down and energization have been clarified, and even in absence of all the necessary information regarding the material behavior at the interfaces, the FE analysis allowed us to adopt acceptable solutions.

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