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FINITE-ELEMENT STRUCTURAL ANALYSIS OF THE
10 m LONG DIPOLE PROTOTYPE MAGNET FOR THE LHC

M. Bona, D. Perini

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Finite-element structural analysis of the 10 m long dipole prototype magnet for the LHC

M. Bona, D. Perini

CERN, European Organization for Nuclear Research, Geneva, Switzerland

Abstract— In the framework of the R&D programme for the LHC superconducting magnets, a series of dipole prototypes, manufactured by European firms, will be placed in a test cell of the LHC machine to be tested in the second part of 1992.

The design of the cross-section of the full length dipole prototype is a natural evolution from the solutions adopted in the CERN models and prototypes developed and built so far.

A detailed finite-element stress and strain analysis was carried out and the results were used to complete the design of the prototype.

The optimization of the design was done by studying the magnet behaviour at room temperature, and at 1.8 K in both energized and non-energized conditions. The Lorentz body forces distribution, corresponding to a 10 tesla central field, was computed from a magnetic field analysis by means of the same finite-element package used in the stress-strain analysis.

The present paper reports on the main results obtained, and illustrates the guide-lines which were followed in optimizing the design of the cross section.

I. INTRODUCTION

Eight 10 tesla dipole magnets will be placed in the LHC test cell.

Finite-element analyses were carried out to investigate the magnetic and mechanical behaviour of these dipoles.

The first step of this work was the determination of the electromagnetic forces in the coils once the magnet is energized.

Then the computed Lorentz body forces were applied to the structural model of the magnet cross-section (Fig.1).

In this way it was possible to determine the value of several parameters influencing the behaviour of the dipole at room temperature and its performance at the operating temperature of about 2 K.

When current is flowing in the cables, the coils must be under compressive stress in order to limit as much as possible their deformation due to energization [1].

This calls for a very stiff structure which can be achieved by fulfilling the following two conditions [2]:

- a) the gap between the two iron yoke halves must be closed both before and after the energization
- b) a good contact must exist between the iron yoke and the collars for all design conditions.

Fig.1 The dipole cross-section.

Furthermore the compressive stresses in the coils at room temperature must be limited to avoid creep of the insulation.

The parameters to be optimized in order to satisfy the two conditions are the following:

- 1) thickness of the shims to be added to the coils to obtain the required prestress at room temperature ($sp1$, $sp2$, $sp3$, $sp4$).
- 2) size of the gap between the iron yoke halves (Sg , Ig).
- 3) interference between the iron yoke and the stainless steel shrinking cylinder (pre).

The latter parameter is a crucial one, since it defines the correct "driving" force required to close the iron yoke gap during cool down.

II. FINITE-ELEMENT MODELS AND OPTIMIZED PARAMETERS

Magnetic and structural analyses were carried out on finite-element models of the cross-section in order to achieve the detailed design of the dipole magnet [3]. The package used was ANSYS 4.4. Details on the adopted element types as well as on the analyses can be found in the ANSYS manuals [4].

A. Magnetic model

The scope of this analysis was to find the distribution of the Lorentz body forces in the coils at a current of 15060

amperes in the cables, corresponding to a central field of 10 tesla.

The computation was done considering the model at its nominal dimensions; the non linearity of the iron-yoke $B-H$ curve was taken into account by means of an iterative approach. The model consisted of about 8000 two-dimensional multi-field solid elements, named *STIF13* in the *ANSYS* library.

B. Structural model

Meshing only one quarter of the structure shown in fig.1 would not be sufficient to properly simulate the behaviour of the magnet cross-section.

It was actually necessary to mesh at least two layers of laminated collars to account for the fact that the aluminium collars consist of two parts, alternatively placed along the axial direction. A correct model must consider the axial continuity of the coils, the iron yoke and the shrinking cylinder and the alternate disposition of the collars laminations. This problem was solved by meshing a one millimeter thick layer for the external cylinder, iron yoke and coils and two 0.5 mm thick layers for the aluminium collars. The collars are connected one to the other in the axial direction by pins and by proper boundary conditions along the horizontal axis.

Around 5000 two-dimensional isoparametric solid elements (*STIF42*) and 250 three-dimensional gap-interface elements (*STIF52*) were used. Due to the bilinear behaviour of the interfaces [4] an iterative calculation routine was necessary.

The magnet behaviour was assessed in several load cases which represent different conditions for the magnet:

- Collars and coils assembly at room temperature.
- Assembled magnet with prestressed shrinking cylinder at room temperature.
- Whole magnet at low temperature (2 K).
- Energized magnet.

Friction between surfaces where relative displacements can occur was not considered. Analyses including friction are at present under way at *CERN*.

The calculations yielded the nominal dimensions of all the magnet components. Obtaining a set of values for the design parameters which satisfies the design assumptions does not necessarily mean that such a set is the optimal one to be used in the magnet fabrication drawings. An intense programme of calculations was carried out on different 'nominal' solutions, by varying some of the crucial parameters over the expected tolerance excursions. In such a way, the 'optimal' set of design parameters was defined, as that which provides the most stable condition when perturbed around its nominal values. The main variations were as follows:

- Variation of the iron-yoke gap.

By means of shims it is possible to fix the gap at the inner radius (I_g) to its nominal value. The outer radius gap (S_g) is varied inside the range ± 0.02 mm (run1,run2).

- Variation of the insert dimensions.

The vertical contact between the collars and the yoke, important to increase the stiffness of the structure, is mediated by the insert. The possible combinations of collars, insert and iron yoke dimensions are here simulated varying the insert size by ± 0.04 mm (run3,run4).

- Variation of the interference between the iron yoke and the external cylinder.

The interference is changed within the range of ± 0.05 mm (run5,run6).

III. OBTAINED RESULTS

Fig.2 represents a plot of the magnetic field inside the aperture.

The resulting forces in the right quadrant of the coils per millimeter of thickness are $F_x=2237$ N, $F_y=-1192$ N.

The optimized values of the critical parameters are listed in Tab.I while Tab.II gives the displacements and stresses in several important locations in the structure (Fig.3) in all working conditions.

As can be seen, stresses inside the coils at room temperature, both after collaring and at assembled magnet, are low enough to avoid creep of the insulation.

During the cool down between 293 K and 2 K the compressive prestresses in the coils increase reaching the maximum at 2 K. The gap between the iron halves is completely closed providing a large stiffness of the structure. The total mating force for the considered quarter of the structure reaches its maximum at 2 K with a value of 2229 N per millimeter of length.

At 2 K the contact force between iron and collars at the collar shoulder reaches its minimum value of 92 N per millimeter of length.

Once the magnet is energized the upper edge of each coil is still compressed (Fig.4), the mating force between the yoke halves is lowered to 1224 N per millimeter of length and the gap stays closed (the yoke behaves as a single piece). This guarantees that the structure will have a large stiffness.

Fig.2 Magnetic field [T] in the 50 mm diameter aperture.

TABLE I
OPTIMIZED MAIN PARAMETERS

Interference iron yoke - shrinking cylinder in the radius (pre) :				0.46 mm
Iron yoke gap:				
	inner (I _g) :	2x0.45 mm		
	outer (S _g) :	2x0.70 mm		
Shim thickness:				
	inner coil, inner position (sp1) :	0.28 mm		
	inner coil, outer position (sp2) :	0.30 mm		
	outer coil, inner position (sp3) :	0.30 mm		
	outer coil, outer position (sp4) :	0.32 mm		

The ovalization of the magnet due to the electromagnetic forces is therefore very low and the stress distribution in the external cylinder is practically the same before and after the energization.

Tab.III compares the 'optimal' nominal situation to those where some tolerance errors are introduced.

Limited changes of the gap size do not affect very much the general structural behaviour (run1,run2).

The sensitivity of the system to the variations of the collar\iron yoke vertical contact is more important. The iron yoke gap actually starts to open at its inner radius once the magnet is energized (run3,run4).

In the cases considered such an opening is not very dangerous since it remains low. However it could become dangerous if larger errors are present in the real magnet.

Fig.3 Significant points in the structure and adopted symbology.

As expected, increasing the cylinder prestress could induce in it too high stresses, while too low a prestress causes the gap to open (run5,run6).

TABLE II
STRESS AND DISPLACEMENTS IN SOME SIGNIFICANT POINTS OF THE STRUCTURE (FIG.3)

Nodes	Coll.+coils assembly		T=293 K		T=2 K I=0 A		T=2 K I=15060 A	
			Displ. [mm]		Displ. [mm]		Displ. [mm]	
	Ux	Uy	Ux	Uy	Ux	Uy	Ux	Uy
A			0.0000	0.1687	0.0000	-0.4926	0.0000	-0.5248
B			0.2166	0.0000	-0.5598	0.0000	-0.5110	0.0000
C			-0.5116	-0.2818	-0.7000	-0.8645	-0.7000	-0.8971
D			-0.3188	-0.2678	-0.4500	-0.5368	-0.4500	-0.5632
E			-0.2290	0.0000	-0.8110	0.0000	-0.7537	0.0000
F	0.0141	0.2281	-0.0652	0.2810	-0.3557	-0.1563	-0.3349	-0.2166
F'	0.0164	0.2216	-0.0585	0.2762	-0.3511	-0.1598	-0.3286	-0.2194
G	0.0082	0.2762	0.0203	0.2774	-0.2317	0.2726	-0.2863	0.2665
H	-0.0227	0.2759	-0.2012	0.2803	-0.5059	0.2731	-0.3824	0.2673
	Stress [N/mm2]		Stress [N/mm2]		Stress [N/mm2]		Stress [N/mm2]	
	radial	azimut.	radial	azimut.	radial	azimut.	radial	azimut.
1	-7.08	-37.11	-6.44	-40.39	-9.00	-51.95	-38.55	-99.77
2	-19.73	-33.54	-22.50	-39.49	-33.09	-47.07	-35.73	-20.82
3	-5.51	-32.85	-4.39	-39.61	-6.46	-52.92	-24.28	-78.52
4	-8.27	-34.22	-9.42	-43.67	-13.05	-55.75	-5.46	-17.82
5	-7.20	-38.43	-11.19	-52.40	-8.99	-59.29	-26.32	-79.82
6	-9.89	-39.27	-16.32	-54.97	-15.61	-61.50	-6.53	-19.38
7	-10.65	-45.87	-17.01	-49.28	-13.26	-59.80	-41.86	-100.60
8	-22.49	-41.97	-19.05	-49.53	-34.14	-54.75	-36.18	-22.14

TABLE III
STABILITY ANALYSIS. THE MEANING OF THE SYMBOLS IS GIVEN IN
TAB. I AND FIG. 3. N/20 IS THE FRACTION OF THE GAP
LENGTH WHICH IS CLOSED

Reference situation				run1			run2		
Sg [mm]	0.70			0.72			0.68		
Ig [mm]	0.45			0.47			0.43		
Gd [mm]	0.30			0.30			0.30		
pre [mm]	0.46			0.46			0.46		
Load step	b	c	d	b	c	d	b	c	d
Fg	--	2229	1244	--	2061	1056	--	2378	1432
n/20	--	20	20	--	20	18	--	20	20
Fi	431	192	373	431	197	382	431	189	364
Fs	1115	92	1075	1115	168	1171	1115	34	978
$\sigma_{cyl.}$	111	232	232	111	223	223	111	241	241
run3				run4			run5		
Sg [mm]	0.72			0.68			0.70		
Ig [mm]	0.43			0.47			0.45		
Gd [mm]	0.30			0.30			0.34		
pre [mm]	0.46			0.46			0.46		
Load step	b	c	d	b	c	d	b	c	d
Fg	--	2251	1294	--	2196	1209	--	2236	1262
n/20	--	19	20	--	20	10	--	20	18
Fi	431	363	542	431	23	217	347	33	203
Fs	1115	47	1001	1115	148	1136	1071	69	1040
$\sigma_{cyl.}$	111	230	229	111	235	234	107	231	230
run6				run7			run8		
Sg [mm]	0.70			0.70			0.70		
Ig [mm]	0.45			0.45			0.45		
Gd [mm]	0.26			0.30			0.30		
pre [mm]	0.46			0.51			0.41		
Load step	b	c	d	b	c	d	b	c	d
Fg	--	2220	1225	--	2589	1605	--	1868	883
n/20	--	20	19	--	20	20	--	20	19
Fi	516	358	547	517	224	407	344	159	339
Fs	1159	118	1110	1266	91	1073	965	94	1077
$\sigma_{cyl.}$	116	234	233	126	268	268	96	196	196

A = -137.298 D = -80.074 G = -22.851
B = -118.224 E = -61 H = -3.776
C = -99.149 F = -41.925 I = 15.289

Fig. 4 MICO, load step 3, the magnet at cold, 1.8K ENERGIZED

Fig. 4 T=2 K, I=15060 A, Azimuthal stress in the coils [N/mm²].

IV. CONCLUSIONS

The finite-element analyses discussed in this paper have led to a detailed design of the magnet cross section.

The obtained values for the design parameters satisfy all requirements of the kinematical and structural behaviour of the magnet, according to the design philosophy.

These calculations have also pointed out the great sensitivity to fabrication tolerances of the behaviour of the magnet. This calls for strict dimensional checking during the fabrication process.

Further analysis and experimental tests are necessary to investigate the influence of friction in the kinematics of the structure.

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