FEM Computations Concerning the Effect of Friction in Two LHC Main Dipole Structures

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Abstract — **The mechanical behaviour of a dipole structure is considered when also friction is taken into account, studying its effect on different components and in different conditions. In particular the difference in behavior between a structure with aluminium collars and one with austenitic steel ones was studied.**

I. Introduction

The Large Hadron Collider [1] requires 1232 superconducting dipole magnets with a magnetic length of 14.3 m and a nominal dipole field of 8.3 T.

The LHC superconducting main dipole is the result of the precise assembly of high precision components. All forces between the components are exchanged along the contact surfaces and change their strength during cool down and magnet operation at nominal field.

Up to May 98 all the FEM computations performed on the structure of the main dipole did not take into account the friction between the different components. The main reason was that the available computing power was not enough to develop and solve an adequate model. From May 1998 a new cluster of computers has been installed at CERN making possible this study that was performed in August 1998.

The influence of friction on the structure is studied in this article, trying to highlight the different behaviour of a structure with austenitic steel collars and of another one with aluminium collars.

II. The Dipole Structure

The dipole cross section is defined by four main components:

- the coils, assembled with prestress inside the collar cavity;
- the collars, clamped around the coils to maintain the coil prestress;
- the iron yoke, split in two parts by a vertical gap;
- the shrinking cylinder, that provides the driving force to keep the two iron yoke halves in contact.

The behaviour of the structure during assembly, cool down and magnet operation and the characteristics of the main components are discussed in detail in [2]. The collars are 3 mm thick. Two possible configurations are here compared:

Design A: collars made of aluminium. Parameters of the structure optimised for that condition.

Design B: made of austenitic steel. Parameters of the structure optimised for that condition.

Fig. 1 shows a quarter of the structure and the main contact surfaces.

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III. Finite Element Models

Two, 2-D finite element (FE) models were meshed and the plane-stress option was used for the elements. It is possible to simulate the collar structures by creating two layers with a 0.5 mm thick mesh. For the coils, iron yoke, cylinder, insert and locking rods a 1-mm thick layer was meshed. The prestress of the coils is imposed by giving an interference at the interfaces between the collars and the coil.

The geometry is modelled at room temperature and in non-deformed conditions. That is to say the dimensions used in the model are for parts at their nominal size. The areas were meshed using *two-dimensional linear* elements (plane 42) and contact surfaces with *three-dimensional contact* elements (contac 52). Further information concerning these elements is available from the ANSYS manuals [3]. Magnet operation is simulated by loading the coils with the electro-magnetic forces computed with the same code. Iron saturation was taken into account.

TABLE I
Material Properties Used for the Finite Element Models

	Temp. [K]	Young Ex [MPa]	Mod. Ey [MPa]	Therm. Exp. Coeff. 10^{-3}[m/K] * αx	αy
Inner layer Insulated cables	1.8	20 700	17 250	1.447	1.95
	293	13 800	11 500		
Outer layer Insulated cables	1.8	18 900	15 750	1.52	2.15
	293	12 600	10 500		
Collaring Rods (St. 316 LN)	1.8	209 000		1.0206	
	293	195 000			
Copper wedges	1.8	150 000		1.128	
	293	120 000			

$$* \ \frac{\int_{1.8}^{293} \alpha(T)\, dT}{293 - 1.8}$$

The mesh considers a one-quarter structure, with appropriate boundary conditions for the simulation of two layers of collars.

Due to the history dependency of systems where friction is present, it has been necessary to reproduce the story of the magnet from assembly to operation. The three main stages, which have to be described, are assembly, cooling and operation at nominal field.

1) Assembly: during assembly the two shrinking cylinder half-shells are pressed one towards the other to apply the correct pre-stress after welding. In the FE model, to simulate this process, the extremities of the half shells are displaced till the desired tension is achieved. When this condition is reached, the edges are blocked in position to simulate the welding. This position is then kept fixed during the following steps.

2) Cool down: in order to get the correct displacements during cool down it has been necessary to introduce, for each material, the curves of thermal contraction and of the Young modulus as function of temperature. This has been done for the following materials: steel or aluminium for the collars, steel for the shrinking cylinder, iron for the insert and the yoke. For the Poisson coefficient a value of 0.3 has been assumed constant with the temperature and equal for all the materials. For the other materials the properties are reported in Tab. 1.

3) Ramp up in field: the current is injected into the cables and the field rises. Because of the Lorentz forces the force exchanged along the vertical gap decreases while those between collars and yoke increase.

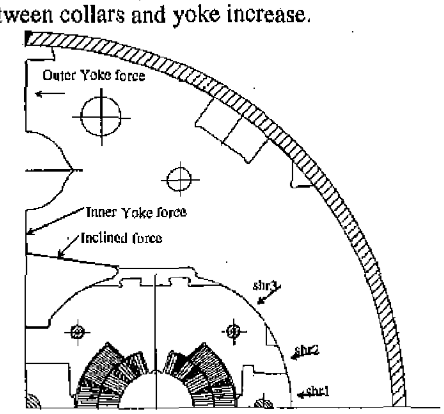

Fig. 1. Cross section and main contact surfaces of the LHC main dipole.

Apart from the effect of friction, from the following results it appears that the two structures behave quite differently. The difference is determined by the presence of an open yoke gap in design A (gap size 1 mm before assembly and 0.6 mm after assembly) while in design B (gap size 0.3 mm before assembly) the yoke gap is already closed at room temperature. In particular for design A the closure of the inner gap corresponds to a change in slope at 90 K for the curve of the mating force along the inclined gap. Because of the differential thermal contraction between collars and yoke, design A shows higher contact forces between the collars and the yoke. Also in the evolution of the shrinking cylinder stress, the difference between an open gap and a closed one is clear. In the first one, in fact, part of the energy is lost closing the gap, while in the second, the force is increasing throughout the cool down.

IV. Computation Along a Cooling, Warming, Cooling Cycle

Friction was applied at the interface yoke/cylinder and along the inclined contact. The value of the friction coefficient (μ) was taken 0.1 for both the contacts [3,4].

This value of μ was used in the following computations in which a complete cool down (293K$\rightarrow$1.8K), warm up (1.8K$\rightarrow$293K), and partial cool down (293K$\rightarrow$140K) have been simulated to study possible memory effect of the structures. It can be generally assessed that no differences are present between the first and the second cool down cycles. The curves of the cool down are partially overlapped in Fig 2 to Fig 4.

1) Effect of friction on the yoke mating forces: the effect of friction along the cylinder-yoke contact and along the inclined contact is a reduction of the forces at cold on the outer and inner gap. For design A (Fig 2) the reduction is about 120 N/mm (-29%) for the outer gap force and about 50 N/mm (-7%) for the inner gap mating force. In the design B (Fig 3) the reduction is respectively 80 N/mm (-5%) and 10 N/mm (-2%). The amplitude of the hysteresis cycle and the reduction of the transmitted forces is bigger for the design A. Since larger matting forces mean more stable mechanical structure, the computations with friction are therefore a good approximation in case of steel collars but too optimistic in case of aluminium collars.

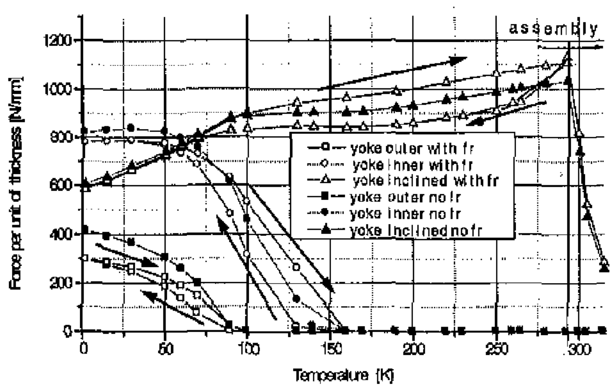

Fig. 2. Design A. Behavior of the mating forces along the gap. The arrows show the cool down and warm up. Large differences are present between the situation with friction and without friction

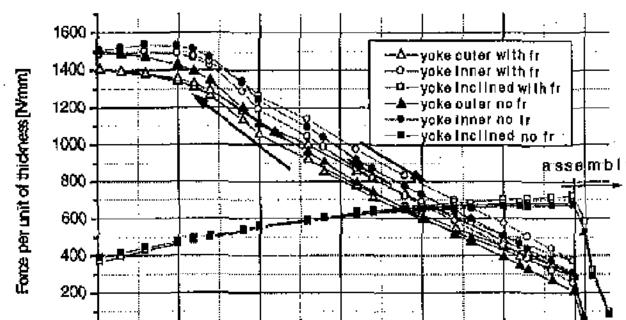

Fig. 3 Design B. Behavior of the mating forces along the gap. The arrows show the cool down and warm up. As it can be see the difference between the two situations (friction /no friction) is so small that the curves are practically overlapped

2) Effect of friction on the shoulder forces: apart from a remarkable hysteresis cycle for design A, there is no big influence of friction for both the designs.

3) Effect of friction on the cylinder tensile stress: in Figs. 4 and 5 the evolution of the shrinking cylinder stresses are shown in two different sections: along the mid-plane and at 80° with respect to the mid-plane. For design B, it is possible to observe that the amplitude of the hysteresis loop is small (5 MPa) and that the higher stress is at the mid-plane with 40-50 MPa more than at 80°. For design A, the difference between the highest and lowest stress is almost the same, but the amplitude of the hysteresis cycle is much wider (40MPa). A higher pre-stress in the cylinder increases the mechanical stability of the structure. Again the situation for the design B is very similar in the two cases: with and without friction. The computations without friction give a bad estimation of reality for design A.

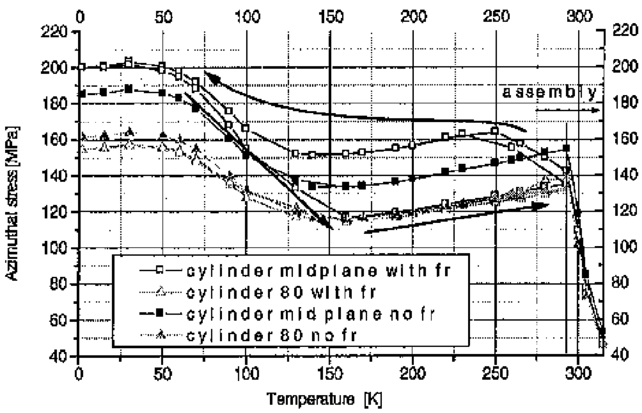

Fig. 4. Design A. Behavior of the shrinking cylinder' s tensile stress during cool down, warm up and partial cool down. Values at 0°, 80° with respect to the magnet mid plane.

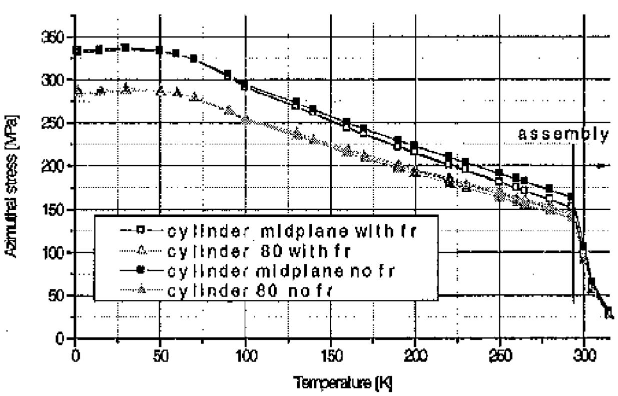

Fig. 5. Design B. Behavior of the shrinking cylinder' s tensile stress during cool down, warm up and partial cool down. Values at 0°, 80° with respect to the magnet mid plane.

V. Computation with Electromagnetic Forces

The computations were re-done adding also the effect of electromagnetic forces up to 8.5 T in 5 steps.

1) Effect of friction on the yoke mating force: the effect of friction during operation is quite small for design B (inner gap + 3%, outer gap – 8%). For design A the outer gap mating force shows the same reduction already present after cool down. For the inner gap the mating force with friction is bigger than the one without friction (+ 54%, + 60 N/mm). This means that although the gap is still closed, the structure is moving at least partially. In this particular case the friction acts like a break helping to keep the gap closed. See Figs. 6 and 7.

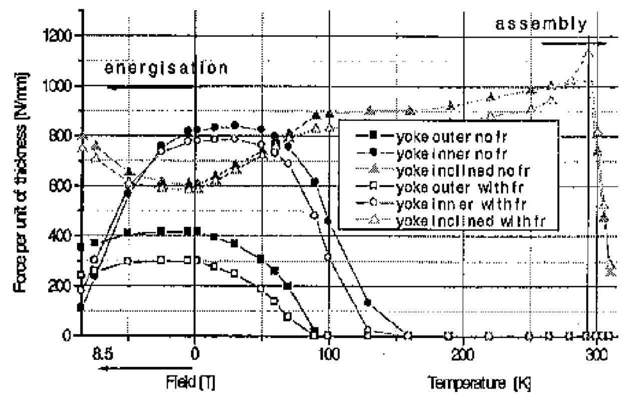

Fig. 6. Design A. Behavior of the gap mating forces during assembly, cool down and magnet operation.

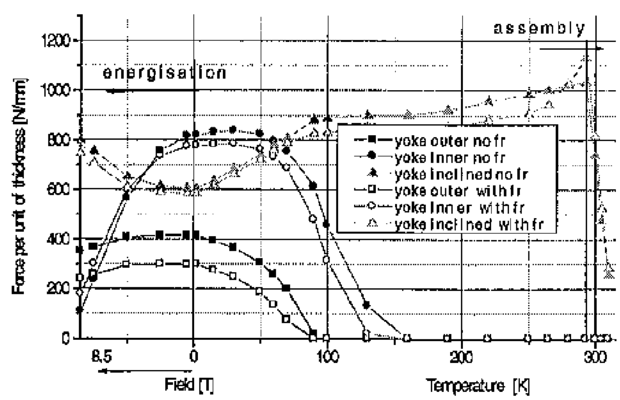

Fig. 7. Design B. Behavior of the gap mating forces during assembly, cool down and magnet operation.

2) Effect of friction on the shoulders' forces: under the effect of the electromagnetic forces the collars are pushed against the yoke. No effect of friction on the strength of forces transmitted to the yoke by the shoulders is observed.

3) Effect of friction on the cylinder tensile stress: when the gap is still closed there is no important variation of the cylinder stress. The effect of the friction is the one left after cool down. See Figs 8 and 9.

In [5] the probability that a magnet meets the design requirements taking in account the effect of the components tolerances has been computed. The same computations performed again considering friction would give a slightly different picture for design A. The probability to have the inner gap closed would slightly increase while the one concerning the outer gap would show a reduction. For design B no evident change would be remarked.

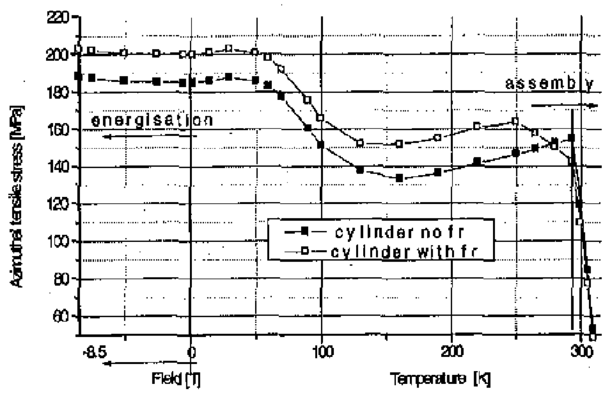

Fig. 8. Design A. Behavior of the cylinder' s tensile stress during assembly, cool down and magnet operation.

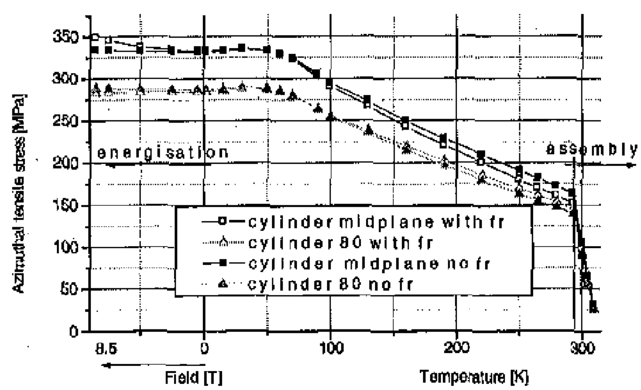

Fig. 9. Design B. Behavior of the cylinder's tensile stress during assembly, cool down and magnet operation.

VI. Experimental Check

On the graph of Fig. 10 the measured curve of the shrinking cylinder's stress during the cool down of the 10-m model MBL1AJ2 on 2nd July 1997 is reported. The magnet had aluminium collars. The starting points of measurements and computations are brought to match. The curves are slightly different because the measured curve describes a structure with racetrack collars with insert inside the collar and no inclined contact. The difference is evident after some points along the curves where the decrease of stress in the structure with racetrack collars is larger.

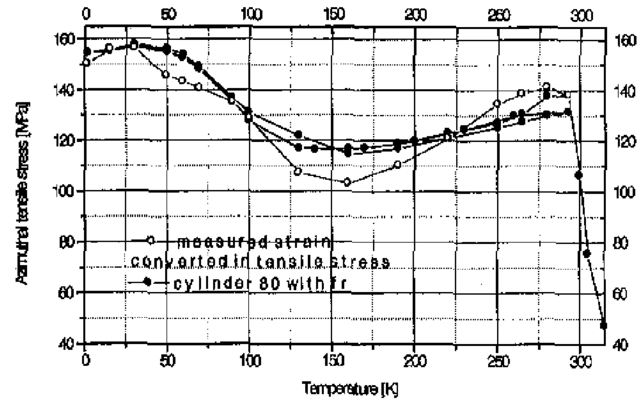

Fig. 10. Comparison between the measured cylinder's tensile stress of the MBL1AJ2 and the computed one. The computations show the cool-down, warm-up, cool-down cycle.

After that the curves present good matching. This proves that the choice of a friction coefficient of 0.1 should not be too far from the reality.

VII. General Remarks and Conclusions

It can be generally assessed that design A, which requires an open yoke gap (0.6 mm) before cool down, presents a higher sensitivity to friction than design A (closed gap). The effects of friction along the cylinder and on the inclined contact between the yoke and insert are relevant for both cases and could affect the stability of design A. Further studies, not reported in this article, show that the friction along the shoulders does not seem to affect the structures. From the literature, the computations and the measurements it seems that a coefficient of friction of 0.1 is a good approximation of the reality. These results indicate that for the purpose of computation, friction should be included in design A, but can be ignored in design B.

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References

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