Mechanical Design of the 2D Cross-Section of the SSC Collider Dipole Magnet*

J. Strait, J. Kerby, R. Bossert and J. Carson
Fermi National Accelerator Laboratory, P.O. Box 500, Batavia, IL 60510

G. Spigo and J.R. Turner
SSC Laboratory, 2550 Beckleymeade Ave., Dallas, TX 75237

Abstract

The 50 mm aperture collider dipole magnet uses stainless steel collars to position the conductors at the locations specified by the magnetic design and to prestress the coil to prevent conductor motion under excitation. The collars are supported by the vertically-split yoke and cold mass skin to reduce their deflection under excitation. The collar interior is designed to give the coil its required shape at the operating temperature taking into account all deflections that occur from assembly and cooldown.

INTRODUCTION

This paper describes the mechanical design of the two dimensional cross-section (Fig. 1) of the base-line collider dipole magnet for the Superconducting Super Collider. The components described here are the collar and yoke laminations and the cold mass shell.

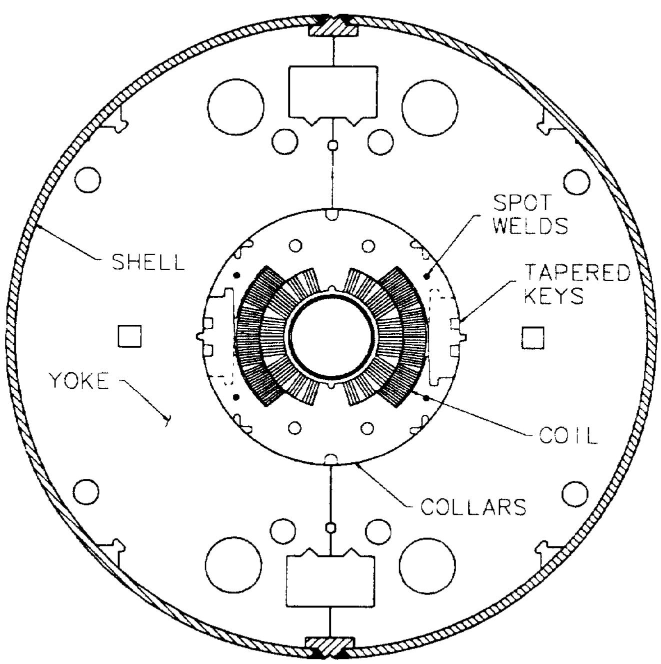

Figure 1. Cross-section of the SSC collider dipole magnet.

The collars, made from 21-6-9 stainless steel, are 17 mm wide and have an outer radius of 67.82 mm. They serve to position the conductors as specified by the magnetic design[1] and to provide restraint against conductor motion under excitation. As in the 40 mm dipoles[2] the upper and lower collars are locked together by tapered keys and left-right pairs of collars are spot welded to give greater horizontal stiffness. The collars are sufficiently stiff by themselves to limit deflections to <0.1-0.2 mm under excitation.

The yoke is used to provide additional support to the collars near the horizontal mid-plane to limit deflections under the Lorentz force. The collars are designed to have a small interference (0.08 mm) with the yoke near the horizontal mid-plane at the operating temperature of 4.35 K. The 4.95 mm thick, 340 mm O.D. 304N stainless steel shell is pretensioned to 200-250 MPa at 293 K to clamp the vertically split yoke around the collared coil. Due to the larger thermal contraction of the shell than the yoke the pretension grows to 350-400 MPa with cooldown and provides adequate clamping to restrain the Lorentz force up to fields well above the design operating point. With the collars supported by the yoke the coil deflections under excitation are $\lesssim$0.02 mm.

We describe in detail below the shape of the outer surface of the collars, which defines the yoke-collar interface, and the shape of the collar interior, which defines the conductor placement. The analysis that lead to this design and a tolerance sensitivity study are also presented. More details of the analysis and a discussion of other collar and yoke features may be found in Reference 3.

YOKE-COLLAR INTEFACE

A vertically split yoke design has been chosen to optimize the horizontal support of the collars by the yoke. The detailed reasoning for this choice has been previously presented[4]. Because the Lorentz force is mainly horizontal, collar deflections are minimized if the yoke supports the collars near the horizontal mid-plane. For assembly reasons the collared coil must be smaller than the yoke along the yoke split direction. During cooldown the collars shrink more than the yoke and may lose contact along the split direction. With a vertically split yoke an interference fit near the horizontal mid-plane can be guaranteed over the full temperature range. The horizontal deflection of the collars under prestress and hence the magnet-to-magnet variation in horizontal diameter are <20% as large as in the vertical direction[5]. This results in a more reproducible yoke-collar fit than in horizontally split yoke designs.

The undeflected collar has a 0.15 mm interference with the yoke within 30° of the horizontal axis and a 0.45 mm clearance between 30°

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and 90°. In both regions the collar surface has a radius 0.01 mm larger than the yoke inner radius; the horizontal interference and vertical clearance are generated by displacing the centers of curvature as shown in Fig. 2.

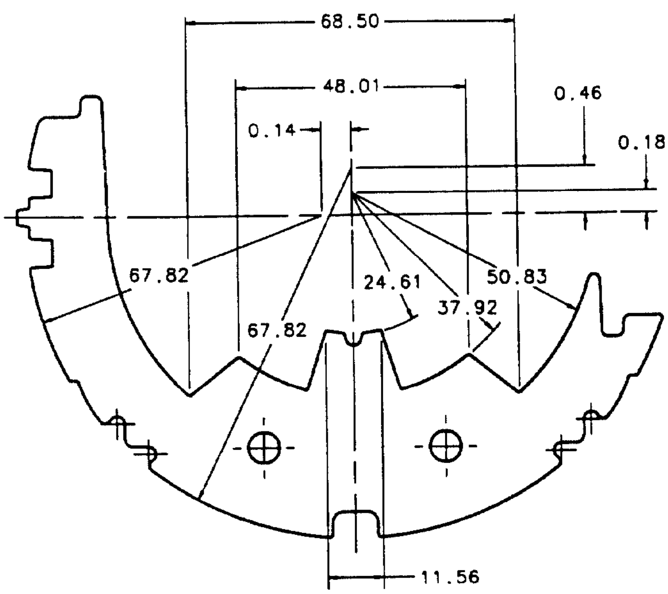

Figure 2. Collar for the SSC collider dipole magnet.

Finite element and analytical calculations were done to determine the required collar vertical and horizontal radii and the sensitivity to parts tolerances. The objectives of the design are: 1) The yoke should load the collars by $\gtrsim$100 N/mm at the horizontal mid-plane at 4 K and zero field. The close collar-yoke-shell fit both limits deflections in the transverse plane and causes the axial Lorentz force to be transferred the shell to limit the compressive loading of the coil end. 2) The yoke parting-plane gap should be closed at T = 4 K to B > 8 T (20% in field and 45% in force above the operating point). 3) The yoke gap should also be closed at 293 K; requirements 1 and 2 take precedence, however.

Two-dimensional ANSYS finite element calculations[6] were done to determine the forces at the yoke-collar and yoke-yoke interfaces and the deflections of the collars, yoke and shell under assembly, cooldown and excitation. The model assumed constant, isotropic and elastic material properties, frictionless contact surfaces, plane stress analysis, and Lorentz loads calculated using infinite permeability iron. The collar dimensions used in the finite element analysis differed in a few places by up to 0.09 mm from the final dimensions.

A simple spring model was used to calculate variations in the interaction among the collar, yoke, and shell with variations caused by parts tolerances and by uncertainties in the parameters of the model. Final collar dimensions were determined by these calculations. The collars were modeled as coupled vertical and horizontal springs for forces applied by coil prestress and shell tension. Because the collars are designed always to clear the yoke in the vertical direction, the model is insensitive to parameters relevant to the vertical radius. The parameters of the calculation are: 1) the vertical and horizontal collar radii at 293 K, 2) the coil prestress, 3) the rate of change of collar radii with prestress, 4) the ratio of vertical to horizontal collar radius change for forces applied by the yoke, 5) the integrated thermal contraction to 4 K of the collar and the yoke, 6) the prestress loss with cooldown, 7) the azimuthal shell stress at 293 K and 4 K, and 8) the rate of change of collared coil vertical and horizontal radii with shell azimuthal tension.

The collar vertical radius is chosen to be 0.45 mm smaller than the yoke inner radius so that there is always clearance. The collar horizontal radius is chosen to give a 0.15±0.05 mm interference fit with the yoke, where 0.05 mm is the combined tolerance band on the radii and horizontal offsets of the collar outer and yoke inner surfaces.

The coil prestress at 293 K was taken to be 70±20 MPa. The prestress loss with cooldown is modeled by cooling the collared coil and the yoke separately and assembling them cold. (This procedure is valid for a linear, elastic system.) The prestress loss was taken to be 19±4 MPa based on measurements[7] of early 40 mm magnets which were built with clearance between the yoke and collars.

The rates of collar radial deflection with prestress were taken to be $1.6\pm0.3\times10^{-3}$ (vertical) and $-0.2\pm0.2\times10^{-3}$ (horizontal) mm/MPa from the finite element calculations corrected by the difference between similar calculations[8] and measurements[9] for 40 mm dipoles. The rate of collar deflection with horizontal force applied by the shell tension is taken from finite element calculations to be $-1.0\pm0.2\times10^{-3}$ mm/MPa of shell stress. The difference in collar and yoke integrated thermal contraction was taken to be $-0.9\pm0.1\times10^{-3}$ based on measurements made at BNL[10] and data from standard tables[11].

The horizontal yoke-collar interference at 4 K is computed to be 0.08±0.05±0.005±0.02 mm, where the first error bar is from parts tolerances, the second is from prestress variations, and the third is from uncertainties in the calculation parameters. The vertical yoke-collar clearance at 293 K after assembly is 0.25±0.05±0.03±0.02 mm. The shell tension required to close the yoke-yoke gap at 293 K is 130±50±7±20 MPa. The 4 K horizontal yoke-collar clamping force is 770±500±50±200 N/mm.

Shell tension is applied at room temperature by weld shrinkage and has been measured in 40 mm dipoles[12] and quadrupoles[13] to be 175-200 MPa, which is near the yield strength of the 304 stainless steel used in these magnets. In the 50 mm dipoles 304N stainless with a minimum yield strength of 310 MPa will be used and a modest increase in shell tension is expected. Thus the shell tension is sufficient to close the yoke parting-plane gap at 293 K under all but the most pessimistic assumptions.

The shell tension has been measured[12,13] to increase by 150-200 MPa with cooldown due to the difference in contraction of the shell and the yoke. The 4 K shell tension, using 304N stainless steel, is expected to be $\gtrsim$350 MPa. Of this 80±50±5±20 MPa is required to close the yoke gap. The remaining force is balanced by the pressure at the yoke mating surface and is available to balance the Lorentz force. With a 4.95 mm thick shell, the yoke halves are clamped with a force ≳2700±500±50±200 N/mm. The lower bound is about 10% larger than the Lorentz force of 1780 N/mm at the operating field. However, finite element calculations indicate that only 45% of the force is transmitted to the yoke, so this design has a margin of >140% against yoke gap opening and the yoke gap should stay closed to at least 10 T.

COLLAR INNER SURFACE

The interior shape of the collar determines the conductor placement which in turn determines the field shape. The combination of collar deflections due to coil prestress, assembly into the yoke and cooldown must result in a coil of the design shape: round, at the correct radius and with the correct pole angles. The target shape for the coils is that specified by the magnetic design shrunk according to the thermal contraction of stainless steel. This would be the shape if the collars were infinitely rigid. Prestress causes the vertical radius to increase, yoke assembly causes the horizontal radius to decrease and the vertical radius to increase and cooldown causes both to decrease with the vertical decreasing somewhat more. The undeflected collar at 293 K must therefore have a horizontal radius larger and a vertical radius smaller than nominal to arrive at the correct shape cold.

Finite element calculations[6] determined the net deflection of the collar due to assembly and cooldown, including both mechanical deflections and thermal contraction effects. The differences between the final dimensions from the finite element calculation and those of the undeflected collar at 4 K (the target shape) are shown in Table I for the case in which the coil prestress at 293 K is 70 MPa. These differences were subtracted from the appropriate dimensions of the nominal collar design to give a shape which, after all deflections have occurred, is correct at 4 K.

Table I
Deflections Relative to a Free Collar at 4 K

Location	δx (mm)	δy (mm)
Outer mid-plane	-0.09	0.00
Outer pole outer corner	-0.02	0.08
Outer pole inner corner	-0.01	0.11
Inner pole outer corner	0.00	0.11
Inner pole inner corner	0.00	0.11

The rest of the collar surfaces are made circular with radii and centers chosen to pass through these five points. Because the cable width is preserved by the deflections, the radii must all be changed by the same amount and they must be concentric. The radii are increased by 0.09 mm to generate the required increase in x at the horizontal mid-plane. The centers of curvature are offset vertically by -0.20 mm relative to the horizontal center line to give the best fit to the design values of y-δy at the values of x-δx in Table I.

The coordinates that were actually used to make the collars differ from the specified values by a small amount. (Numerous design activities were being carried on simultaneously and not all of them were perfectly coordinated.) The largest discrepancy results in a displacement of the outer coil pole surface azimuthally towards the mid-plane by 0.06 mm. All other discrepancies are <0.02 mm.

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