BABAR Note 233

Design Study For The BABAR Superconducting Solenoid

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INTRODUCTION

This report describes the engineering study of the superconducting solenoid for BABAR magnet. The actual design includes the cold mass, the cryostat with the chimney for hydraulic and electrical feedings, the current leads cryostat and the ancillary equipment. Special problems as the supporting system of the coil to the Iron Flux Return are extensively discussed. Information about the magnetic forces at the IFR are also given.

The superconducting solenoid design is based on the criteria developed in the last 15 years for the aluminum stabilized thin solenoids. The first magnet of this class can be considered CELLO, built at Saclay for Petra Collider at DESY. The common feature of these magnets consists in the use of aluminum stabilized conductors indirectly cooled. The cooling pipes are connected to the supporting structure, made by aluminum alloy. The technique developed for CELLO was subsequently improved on building several thin solenoids like CDF, TOPAZ, VENUS, AMY, ALEPH, DELPHI, CLEO-II, H1 and ZEUS, with bore up to 5 m.

Table I   Main characteristics of some thin solenoids

	CDF	ZEUS	CLEO II	ALEPH	BABAR
Location	FNL	DESY	CORNEL	CERN	SLAC
Manufacturer and year of completion	Hitachi 1984	Ansaldo 1988	Oxford 1987	SACLAY 1986	? 1997
Central Field (T)	1.5	1.8	1.5	1.5	1.5
Inner Bore (m)	2.86	1.85	2.88	4.96	3.01
Length (m)	5	2.5	3.48	7	3.47
Stored Energy (MJ)	30	12.5	25	137	27
Current (A)	5000	5000	3300	5000	6830
Total weight (t)	11	2.5	7.0	60	7.0
Radiation Length	0.85	0.9	-	1.6	1.4 max
Conductor overall dimensions (mm)	3.89 x 20	4.3 x 15 5.56 x 15	5 x 16	3.6 x 35	3.2 x 32 5.8 x 32
Overall Current density A/mm^2	64	78 60	42	40	67 37

Table I shows the main characteristics of some of these solenoids compared with the ones of BABAR solenoid. At the present time the huge superconducting magnets for LHC detectors, ATLAS and CMS are being developed starting from the same technology we are using for the design of the BABAR solenoid.

A special remark must be given to the conductors used for these magnets, made by a flat Rutherford cable of NbTi/Cu immersed in a pure aluminum matrix. The coupling of the Rutherford to the matrix is usually obtained through a co-extrusion process. The large Al matrix allows both an adequate protection in case of quenching and a good stability margin with respect thermal disturbances.

CHAPTER 1

1. OVERVIEW OF THE SOLENOID

1.1 REQUIREMENTS

The present design is related to a superconducting solenoid to be used in the BABAR detector. The required central magnetic field is 1.5 T. The field uniformity in the tracking chamber must be maximum ±2 %. The chamber extends axially up to 1483 mm and radially up to 800 mm. The solenoid has to be transparent to radiation; the nuclear interaction length should be limited to 0.4 λ_{int}. The allowable space for the cryostat is limited from a radius of 1380 mm to 1730 mm. The cryostat maximum length is 3850 mm. In designing the solenoid, the segmented flux return and the end doors shields must be considered for their effects on field, field uniformity and offset forces on the solenoid.

1.2 COLD MASS

The cold mass at the operating temperature of 4.5 K is composed of the winding supported by an outer Al alloy cylinder and the supporting system to the vacuum chamber. The winding is made by a flat superconducting cable composed of 20 multifilamentary NbTi/Cu wires. The cable is stabilized by cladding it in a pure aluminum (99.998) matrix. The conductor is insulated by a fiber-glass tape. The winding is directly wound (740 turns) inside the supporting cylinder and impregnated using two components epoxy-resin under vacuum. In order to obtain the required field uniformity, the current density at the solenoid end is designed to be higher than in the central portion. This is made by using two different conductors: thinner at the sides (3.6 mm) and thicker in the center (6.2 mm). The operating current is 6830 A. The peak field at the winding is 2.5 T.

1.3 CRYOSTAT

The function of the cryostat is to maintain the environment for the cold mass. It consist of a tubular vacuum vessel containing the cold mass at 4.5 K and a set of radiation shields kept at 80K by circulating coolant through pipes connected to the shields. The cryostat is supported to the IFR using support brackets, which take vertical, radial and axial loads

1.4 COOLING

The cold mass will be indirectly cooled by circulation of two-phases helium in circuits attached to the cold mass support cylinder. A thermosyphon process is proposed as the coolant driver.

1.5 SUMMARY OF MAJOR DESIGN PARAMETERS

Coil mean conductor radius	Warm Cold	1505mm 1498.5mm
Coil length (conductors only)	Warm Cold	3470mm 3455mm
Field	Central Peak	1.5T 2.5T
Number of turns	Central region Side regions (each) Total	310 215 740
Design current		6833 A
Conductor size	Central portion Ends	32 x 5.8 mm^2 (bare) 32.4 x 6.2 mm^2 (insulated) 32 x 3.2 mm^2 (bare) 32.4 x 3.6 mm^2 (insulated)
Cryostat limiting dimensions (including maximum tolerances)	Internal radius External radius length excluding support brackets	1380mm 1730mm 3850mm
Cryostat nominal dimensions	Internal radius External radius length excluding support brackets	1390mm 1720mm 3848mm
Mass	Solenoid Radiation shields Vacuum vessel Total inc. misc items	7300kg 1000kg 5200kg 13500kg
Earthquake design loads	Vertical Horizontal	2g 1.2g
Magnetic forces on solenoid	Geometric offset Alignment errors	10t axial 20t (2cm errors)
Supports - external	Vertical Radial Axial	Four positions on horizontal center plane, shared with inner detectors Ditto Four positions each end, each acts in one direction only. Inner detectors supported from coil end flange
Supports - internal	Radial+vertical Axial	8 tie-rods, four each end 6 tie-rods one end

CHAPTER 2

2. MAGNETIC DESIGN

2.1 THE MODEL

The magnetic analysis is based on the 2D model shown in Fig. 2.1 The model includes:

1. The solenoid;
2. The laminated barrel and end caps flux return , composed each by 20 steel plates of different thickness;
3. The Q2 shield in the forward end door (see Fig.2.2);
4. An iron shield in the backward end door, (see Fig.2.3);

The model also includes a gap of 150mm between barrel and end caps.
The backward shield is designed to accommodate a DIRC detector, which is supported by two steel rings, which are also included in the model. The main role of the backward shield is to symmetrize the magnetic field , to balance the magnetic force on the solenoid due to the Q2 shield and to improve the field uniformity in the backward region of the tracking chamber. With respect to the real magnet, having hexagonal structure, the main magnetic analysis was carried out for the plane intersecting the hexagon at the center of two opposite sides. A 3 D analysis for a simplified model was also carried out.
The computations were carried out using a 2D magnetic element of ANSYS code (version ANSYS 50 a) , implemented on a Digital ALPHA-VAX and on HP730 stations.
The magnetic steel properties used for computation are in agreement with HOT ROLLED CARBON STEEL having the magnetic properties shown in Fig. 2.4, also called in the present design *US Mild Steel* . In order to study the effect due to the use of not homogeneous steel, some element of the IFR was also considered to be made by *Russian steel* (See fig.2.4 for the BH curve) , as explained later.
The mesh for the magnetic analysis, shown in Fig.2.5, was suited in order not to exceed 15000 plane elements, so that acceptable CPU time was required for calculations. This approach allowed the study of several magnetic configurations leading to an optimization of the field homogeneity.
The coil is meshed into 60 elements of rectangular shape. The radial thickness is 1 cm, in order to reproduce the thickness of the Rutherford cable. The coil axial length is the cold length, i.e. the length

at 4.5 K. The solenoid radius also is considered at 4.5 K. The axial dimensions of the mesh elements are 6.38 cm for the central region and 5.13 cm for the two end regions.

2.2 CENTRAL FIELD AND UNIFORMITY

The aim of the magnetic design was to obtain a magnetic field of 1.5T with uniformity ±2% in the tracking region. This region approximately covers the axial range from -1483 mm backward to +1287 mm forward, with respect the magnetic center; the radial limit is 800mm. Fig. 2.6 shows the details of this region. The uniformity is obtained by grading the current density of the solenoid in three regions. A central region covering ±961 mm including 310 turns and two end regions of length 774mm including 215 turns each. The current density in the end regions is 1.7 times that one of the central part. The magnetic field of 1.5T is obtained by powering the solenoid with a current of 6833 A, the total ampere-turns are 5.0564 10^6. The three sections of the magnet are connected electrically in series. Table 2.1 summarizes the main characteristic of the solenoid.

Fig. 2.7 shows the graph of the field lines over the full detector region. Fig 2.8. shows the field uniformity in the central region defined by 0.225 m<r <0.80 m and -1483mm < z <1275 mm.. The target uniformity of ±2% is obtained in the whole region of interest except at the backward edge, where the uniformity decreases to a minimum of - 3%. In the forward direction a better field uniformity is obtained due to the symmetry.

Generally speaking a better field uniformity could be obtained by reducing the axial length of the two end regions. Nevertheless this causes an increase of the current to generate the same field with a consequent reduction of the stability against thermal disturbance. For the initial design we assumed, as maximum current density in the conductor, the maximum value used up to now for the magnets of this kind, i.e. ≈ 80 A/mm^2 (ZEUS magnet). Using a conductor of cross section ≈ 90 mm^2 , the maximum current results to be ≈ 7000 A. The present design can be considered a compromise between the opposite requirements coming from field uniformity and stability. The chosen operating current of 6833 A gives a current density in the conductor of 66 A/mm^2, which can be considered an upper limit. An increase of the current would cause the coil to be spliced into two layers .

Table 2.1 Overall coil parameters

Central Induction	1.5T
Conductor peak field	2.5T
Uniformity in the tracking region (r< 800mm -1483 mm < z < 1287 mm)	± 2% (3% at the backward edge)
Winding Length	3470 mm *warm* 3455 mm *cold*
Winding mean radius	1505 mm *warm* 1498.5 mm *cold*
Amp turns	$5.0564\ 10^6$
Operating current	6833 A
Inductance	1.15 H
Stored Energy	27 MJoule
The coil is made by two conductors forming 3 regions with different current density: 1 Central region: length Number of turns 2 Side regions: length Number of turns	 1922 mm *warm* 1913.63 mm *cold* 310 770.56mm *cold* 774 mm *warm* 215
Total turns	740
Total length of conductor	6998 m

An interesting feature is that, in spite of the iron a-symmetry, the magnetic field is very close to being symmetric so that a residual net force of only 90 kN is applied directed backward. This fact must be considered in trying adjustment of the geometry leading to an improved field uniformity. As an example, Fig. 2.9 shows the field uniformity obtained by moving the end plug of 40 mm inward. The better uniformity is compensated by the a growth of the net forces on the solenoid from 90 to 220 kN.

2.3 PEAK FIELD AT THE SOLENOID

As described in section 2.1 the solenoid was modeled in form of a thin cylinder of radial thickness 1 cm. The real situation is quite different because the current is shared by 740 turns and flows in the Rutherford conductors, which have a radial thickness of 11 mm and axial thickness of 1.42 mm. In order to evaluate the peak field at the solenoid the real current distribution must be taken into account. The peak fields occur at the two opposite axial ends of the solenoid in the higher current density regions. We have replaced a small part of one of this region, for an axial length of 36 mm, with 10 smaller zones having the dimensions of the Rutherford cable. Fig. 2.10 shows the variation from the usual to the improved solenoid model. This allows to take into account, for the conductors at the solenoid end, both the field and the self field. The peak field is just applied at the last conductor (at the solenoid end) and has a value of 2.35 T. In the present design we will consider , for safety reasons, a peak field slightly higher, i.e. 2.50 T.

2.4 MAGNETIC FORCES AT THE SOLENOID

Axial and radial magnetic forces are applied to the solenoid as shown in fig. 2.11, where the forces at each element of mesh are displayed as arrows.. The general characteristics of these forces are:
1- The radial forces are higher at the end regions than at the central region
2- The axial forces are inward directed for the end regions and outward directed for the central region
The radial pressure as function of the axial position is shown in fig. 2.12. The pressure applied to the central region, with lower current density, is quite independent on position and equal to 0.85 MPa. The pressure at the end regions is much higher, ranging from 1.15 MPa to 1.52 MPa. There is a strong gradient of the radial pressure at the interface between higher and lower current density regions. The pressure falls from 1.52 MPa to 0.86 MPa in few millimeters. In the stress analysis these two zones will be studied more carefully.
Fig. 2.13 shows the axial forces at each element of the mesh as function of the position of the elements. The force is substantially compressive for the end regions as also displayed by fig. 2.11. An important parameter is the maximum axial force applied at a single element. From Fig. 2.13 this force is 1.3 MN, applied to the end elements of the end regions. In performing shear stress calculation , this force must

be taken into consideration. The integral force as function of the position is shown in fig. 2.14. The total compressive force at each end region is 4.8 MN, while the outward directed force at half central region is 1.6 MN, so that the coil is compressed with a total force of 3.2 MN.

2.5 MISALIGNMENT IN THE IFR

As already mentioned, due to the IFR a-symmetry, the total axial force at the solenoid is not zero but there is a net force of 90 kN backward directed. Due to the non-linearity of the B-H curve, the net force strength and direction depends on the field: at half the field (B=0.75 T) the net force on the solenoid is 35 kN forward directed. We have studied what are the consequences on the offset force at the nominal field due to misalignment of the coil in the IFR or to the variation of position of some elements of the IFR. We performed several exercises as following:

1) The first exercise was made moving the solenoid with respect to the IFR of 20 mm in the backward direction. The net axial force changed from 90 to 300 kN, so that we have the information that the axial misalignment causes a force of 10.5 kN/mm.
2) The second exercise consisted in removing the inner plate of the backward end cap. The calculated axial force at the solenoid changed of 200 kN, (From 90 backward directed to 190 forward directed). The information coming from this result is that changes in the IFR as big as expected (plate mispositioning) do not cause the solenoid to suffer for high loads (being of course adequately supported).
3) The third exercise was to move axially the backward end plug. Apart the effect on the field uniformity, the result on the axial force is 3.3 kN per mm of the end plug displacement. The force at the solenoid is backward directed moving inward or forward directed moving outward. The consequence of this calculation is that a movement of the backward plug outward of 28 mm reduces to zero the net force at the solenoid. Generally speaking the backward plug can be used to trim the magnetic force due to misalignment of the coil or to IFR a-symmetry caused by plate mispositioning.
4) We studied the effect of changing the magnetic properties of the iron. A big change in the net axial force was expected by using two different steels for the end caps. We performed a calculation using *US mild steel* (the usual steel used in the present design) for the forward end cap (Q2 shield included) and *Russian steel* (see Fig. 2.4) for the backward end cap. The observed change in the axial force was few kN. In order to take into account the non-linear effect of the iron, this computation was also made at half the nominal current (i.e. at 3400 A). We found a force of 40 kN

forward directed, i.e. a change of 130 KN with respect the force at the nominal current. A result very similar to the one obtained when using only one steel for both the doors . A little worst field uniformity was also observed. Though this result would not encourage the use of whatever steel for the IFR, we have the important information that IFR elements of not homogeneous steel, do not cause *catastrophic* loads at the solenoid.

5) As last exercise we changed the gap distance between the plates of the backward end caps from 30 to 33 mm (Fig. 2.15). This was made in order to give a tolerance on the space reserved for the RPC detectors. The offset force on the solenoid changed from 90 to 30 kN, with no effect on the field uniformity at the tracking chamber.

After analyzing the net axial force, we studied the effect of radial displacement of the solenoid with respect to the IFR. A correct study of this effect would require the use of a 3D code. Nevertheless important information can be drawn from a 2D analysis. We calculated the total radial force applied at a solenoid with the same axial dimension and Ampere-turns, but with a mean radius of 20 mm higher than the real solenoid. Gluing together half solenoid of the real case and the solenoid with higher radius, we obtain a fictitious solenoid radially moved 10 mm with respect the original one. Looking at the total radial force of this fictitious solenoid we obtained 100 kN outward directed so that we have a force per unit displacement of 10 kN/mm.

These exercises were used to determine the maximum mechanical loads, which can be applied at the solenoid, in relation to the tolerances of the coil and IFR positioning.

2.6 FORCES AT THE IFR

The magnetic analysis, required for the solenoid design, was made using the IFR geometry under design, as already pointed out in section 2.1. As consequence of this approach, the complete force configuration at the IFR was available as shown in fig. 2.16 for the whole IFR and fig. 2.17 and 2.18 for the end caps.

Table 2.1 Magnetic Forces at the backward end cap

Element	Axial force (kN)	Radial force (kN)
End Plug	- 1568	+512
DIRC support inner ring	- 274	+245
DIRC support outer ring	-284	+412
I plate (the inner)	-519	-118
II	-352	-29
III	-264	-29
IV	-206	-29
V	-157	-39
VI	-108	-39
VII	-69	-49
VIII	-39	-59
IX	-10	-59
X	+10	-59
XI	+20	-59
XII	+39	-108
XIII	+39	-98
XIV	+29	-78
XV	+29	-69
XVI	+20	-49
XVII	+20	-39
XVIII	+20	-69
IX	+20	-59
XX	+10	-49

The forces at each elements of the IFR are also listed in Table 2.1, 2.2 and 2.3 , where the values, are given over 2 p. The sign minus (-) means that the force (axial or radial) is directed from outer the magnet toward the inner.

Table 2.2 Magnetic forces at the Forward end cap

Element	Axial force (kN)	Radial force (kN)
Q2 shield I shield II shield III shield	 - 1372 -29 -10	 +? (*) + 88 +$
I	-617	?
II	-392	?
III	-304	?
IV	-235	?
V	-176	?
VI	-127	?
VII	-88	?
VIII	-69	?
IX	-49	?
X	-29	?
XI	-20	?
XII	-20	?
XIII	-20	?
XIV	-10	?
XV	-10	?
XVI	-5	?
XVII	-6	?
XVIII	-5	?
IX	-3	?
XX	-4	?

(*) The radial force between the first Q2 shield and the forward plates can not be known because there is no air gap between them. For the model used, the first Q2 shield and forward plates constitute a single element. However an indication of the force can be obtained looking at the force between backward plates and backward end plug.

Table 2.3 Magnetic forces at the barrel

Element	Axial force (kN)	Radial force (kN)
I	-3	-402
II	-2	-314
III	-2	-235
IV	-2	-176
V	-2	-137
VI	-2	-98
VII	-2	-69
VIII	-2	-49
IX	-2	-39
X	-2	-20
XI	-1	-5
XII	-1	-29
XIII	-1	-4
XIV	-2	-4
XV	-2	+5
XVI	-2	+6
XVII	-2	+7
XVIII	-2	+8
IX	-3	+29
XX	-2	+245

2.7 COMPARISON WITH OTHER CODES

In order to have a confirmation of the results obtained by using ANSYS code , we carried out a magnetic analysis on a simplified 2D model of the BABAR magnet. The model is shown in fig. 2.19 . The same model was used to perform magnetic field calculation using PE2D at LLNL. The results obtained by ANSYS and PE2D are shown respectively in fig. 2.20.a and 2.20.b for the field in the drift chamber. The absolute field at the center of the solenoid is 1.5274 as given by PE2D and 1.5209 by ANSYS. The field uniformity in the drift chamber region is quite the same. As conclusion we can say that the results obtained by ANSYS code, and discussed in the present technical design, are confirmed by this comparison test.

2.8 3 D MAGNETIC ANALYSIS

The IFR has hexagonal structure. A more realistic magnetic analysis should be carried out using a complete 3D model. Nevertheless a 3D model containing all the elements of the IFR (60 plates and the shields) would be a very hard task. We tried a different approach by using a simplified 3D model, which is shown in fig.2.21. The used symmetry is 1/2. The main difference with respect the 2D model is that the barrel and the end caps are modeled as 3 single pieces with averaged and anisotropic magnetic properties. This simple model allows to evaluate the effect on field uniformity due to the hexagonal shape and the forces on the solenoid due to non-symmetric iron distribution (as the lack in the backward end cap for leaving the space to the chimney).

When calculating the field uniformity in the plane normal to Z axis at Z=-1400 mm, i.e. at the backward border of the drift chamber., where we have more effect of the IFR geometry, we found that the field variation due to the hexagon shape is less than 0.15 %, so that we can conclude that the 2D analysis is substantially correct.

Presently we have not still ready the calculation of the effect of the lack of iron due to the chimney. Nevertheless a preliminary evaluation seems to show that an axial magnetic load of 90 KN, applied to the solenoid in the forward direction, and a vertical load of 30 KN, downward directed, take place.

CHAPTER 3

3. CONDUCTOR

3.1 Al-Stabilized Conductor

The conductor is composed of a superconducting Rutherford cable embedded in a very pure aluminum matrix through a co-extrusion process, which ensures a good bonding between aluminum and superconductor. Fig. 3.1 shows the cross section of the two conductors used for the solenoid. The overall dimensions of the bare conductor are 3.2mm x 32mm, for the higher current density regions and 5.8mm x 32mm for the central region as reported in Table 3.1 showing the main parameters of the conductor.

The Rutherford cable was designed starting from multifilamentary NbTi strands of diameter 0.84 mm, this strand is in the range of the standard products by companies making superconducting wires.

To determine the electrical transport properties of the strands we took advantage from our direct experience in developing the conductors for LHC experiment magnet at CERN. Fig. 3.2 shows the critical current density vs. the field at T=4.5 T, as expected for the virgin strands on the basis of our measurements on similar strands. The cabling process causes a degradation of the critical current , measured to be 8 - 9%, whilst further 3 % degradation is due to the co-extrusion process. For safety reasons we assumed a total degradation of 20%. As result, the total S/C cross section of 3.96 mm^2, shared by 20 strands of 159 filaments 40 µm diameter , should be able to carry 15200 A at B=2.5 T and Temperature 4.5 K.

Fig. 3.3 shows the load line intercepting the critical curve. The operating current is 45% of the critical current at the peak field giving a large safety margin. In the case of local heating up to 5.2K, there is still significant margin on the critical current (I =60% Ic).

The winding require a conductor length of 3 km for the central region and 2.0 km for each end region. The conductors could be produced in minimum four lengths (2 of 1.5 km and 2 of 2.0 km) or maximum 6 lengths depending on several conditions (quality over the length, costs, required tooling for the winding, ...). The use of different conductor lengths requires 3 or 5 electrical joints to be made inside the winding. The joints made by either a welding of the aluminum matrix or a soft soldering, after a copper covering (electro-deposition) of the aluminum, should have a resistance less than 5×10^{-10} Ω each, in order to limit the power dissipation to few mW.

Table 3.1 Conductor characteristics

Conductor type	Pure Al-stabilized co-extruded	
Aluminum RRR	1000	
Conductor unit length	Central region	3.Km
	Side regions	2.0 Km each
Number of lengths	Central region	2
	Side regions	2
Dimensions at 300 K:		
Bare	3.2 and 5.8 x 32.0 mm2	
Insulated	3.6 and 6.2 x 32.4 mm2	
Conductor Overall current density		
Central region	37 A/mm^2	
Side regions	67 A/mm^2	
Superconducting cable	Rutherford	
Dimensions	8 x 1.42 mm^2	
Strands diameter	0.84mm	
Number of strands	20	
Cu/Sc	1.8	
Filament diameter	40 µm	
Ic(B=2.5, T=4.5)	15 kA	
Insulation type	fiber-glass tape	
thickness	0.2mm	
Material content:		
NbTi	3. 96 mm^2 (3.3 and 2 %)	
Cu	7.13 mm^2 (6.9 and 3.5 %)	
Insulation	14.1 mm^2 (12 %) end reg. 15.2 mm^2 (7.5 %) center reg.	
Al pure (99.998)	91 mm^2 (77.7 %) end reg. 174 mm^2 (87 %) center. reg.	

3.2 Electrical insulation

Electrical insulation is an extremely important aspect of solenoid design and manufacture. Two categories of insulation are required:

(i) ground plane insulation between the coil and support cylinder. The ground plane insulation must operate at relatively high voltages during quench conditions and will be subjected to strict QA controls. The design of quench protection systems is based on a maximum voltage to ground of 250V. The ground plane insulation will be made by a 1mm layer of glass fiber epoxy laminate which is bonded to the support cylinder before winding. The insulation will be fully tested at 2kV before winding.

(ii) turn to turn insulation: Conductors will be insulated with a double wrap of ~0.1mm glass tape during winding to give an insulation thickness of 0.2mm. The turn to turn insulation thickness will be 0.4mm and will be fully impregnated in the bonding process. The conductor must be insulated during the winding process.

Electrical tests will be carried out during winding to detect any failure of insulation. The tests will include continuous testing for turn to turn and turn to ground insulation.

3.3 Conductor stability

The BABAR Solenoid coil will be indirectly cooled using the technology established for detector magnets such as DELPHI, ALEPH, CDF etc. The reliable operation of these existing magnets has demonstrated that safe stability margins can be achieved using high purity, aluminum clad superconductors in a fully bonded, indirectly cooled coil structure.

3.3.1 Stability Modeling

Modeling Codes

Conductor stability has been estimated using modeling codes developed at RAL and INFN Genoa for the study of LHC Detector magnets (Ref. (1) ASC Paper). The concept of the stability model is shown in figure 3.4 which also outlines the terminology. The model is set up to represent a defined length of coil matrix (typically 10m) in the longitudinal direction. The model is divided into longitudinal elements and an initial heat pulse is applied to a specified conductor length for a specified duration.

The response of the conductor is modeled by a simple stepwise time integration taking into account the following effects in a 'lumped' conductor model:-

i. longitudinal thermal conduction
ii. ohmic heat generation when the temperature exceeds the current sharing temperature of the superconductor - note current diffusion can be included in the model
iii. heat transfer to the support shell through the ground plane insulation
iv. transverse heat transfer to adjacent conductors by conduction through the conductor insulation
v. transverse heat transfer by conduction in the support shell which is also taken as a lumped model for simplicity. The longitudinal thermal conductivity of the support shell is not modeled because it is small compared to the longitudinal conductivity of the conductor high purity aluminum
vi. the resistivity of the aluminum substrate is assumed to be invariant with temperature - a reasonable assumption for temperature <20K
vii. the thermal conductivity of the aluminum substrate is assumed to obey the WF Law
viii. the thermal conductivity of the epoxy glass insulation is modeled as a temperature dependent curve given by the algorithm
ix. the non-linear thermal capacity of the elements is modeled with algorithms evaluated to fit measured material properties.

The stability models have been compared with experimental results obtained on a DELPHI test coil and have been shown to agree with experimental results.

3.3.2 BABAR Conductor Stability

The parameters of the stability model for the BABAR Solenoid are given in table 3.2 for the conductor in the high current density section of the coil. Stability estimates are given in Table 3.3 for three different conductors with aluminum working (at field) RRR 250, 500 and 750 respectively. Table 3.3 also includes quench propagation velocities computed using the stability model. Quench propagation in the longitudinal and transverse directions for the conductor with RRR 500 are shown in figures 3.5, 3.6.

In all cases the stability is computed for an input heat pulse duration of 100msec. Figure 3.7 shows typical temperature profiles in the conductor at 100msec and 200msec.

Table 3.2 - BABAR Solenoid Stability Parameters

Model Parameters	
Conductor width	32mm
Conductor thickness	3.2mm
Insulation (turn/turn)	0.4mm
Shell thickness	30mm
Ground plane insulation	1mm
RRR Al	500
Peak field	2.5 Tesla
Design Current Ic(2.5T)	16kA
Operating Current	6.83kA
Current Sharing Temp	6.5K
Critical Temp	8.2K
Pulse Time	100 msec

The computed stability shows a strong dependence on the properties of the aluminum substrate. In the BABAR Solenoid this is one of the few parameters which can be adjusted with minimum impact on coil geometry and magnet performance. The only impact of changing RRR will be on the magnet cost although the effect of changing RRR 500 to 750 is expected to be small ~2-3% for the solenoid.

Table 3.3 - BABAR Solenoid Stability

Working RRR B=2.5T		250	500	750
Operating Current	Amps	6833	6833	6833
Width	mm	32	32	32
Thickness	mm	3.2	3.2	3.2
Quench Energy	J	0.6	1.6	2.8
MPZ length	m	0.3	0.6	0.9
MPZ characteristic time	sec	0.17	0.18	0.18
Propagation Velocity				
Longitudinal	msec	4.5	4.4	4.2
Transverse	msec	0.1	0.08	0.06

In order to understand whether the calculated stability margins are acceptable, we can consider that in the case a complete turn, placed at the solenoid end, losses contact with the Al cylinder (due to an epoxy failure) and moves under the action of the magnetic load, a heat dissipation of 0.4 J occurs. The quench energy 1.6J computed for the RRR = 500 conductor is expected to be adequate for a coil such as BABAR which is designed to work at low stress levels.

CHAPTER 4

4. COLD MASS

4.1 Design concept

The cold mass is designed as a simple solenoid, with the windings supported by an external cylinder of 30mm thick Aluminum alloy 5083 (see figure 4.1). Cooling is provided by externally wound liquid Helium pipes, which cool the conductor indirectly. At the ends, the cylinder is thickened to provide a substantial anchoring point for the supports. Flanges are provided to help to keep the windings in place axially. These features will be discussed in detail in the following sections.

4.2 Supporting Cylinder

In chapter 2 we showed how the magnetic field causes a radial pressure of 1.5 MPa at the solenoid ends and 0.8 MPa at the central region. The pure aluminum of the conductor is not able to support such pressures due to its low yield strength (less than 15 MPa), so that it is necessary to include a mechanical structure providing the hoop strength. This problem is common for all the thin Al-stabilized detector magnets. The usual solution consists in supporting the winding by an outer aluminum alloy (5083 or 6061) cylinder. In designing such structure we followed some criteria:

i) The mechanical stresses in the winding and supporting structure are due to several effects like residual stresses after the construction, differential thermal contractions, magnetic loads and thermal and magnetic cycles. We have to look at the stresses generated at each stage (construction, cooldown, and charge / discharge cycles) to determine the thickness of the supporting cylinder.

ii) The cylinder thickness should be designed to limit the strains of the Al-stabilizer in the elastic region during charge and discharge. This criterion, which does not always apply, help in minimizing the thermal disturbances coming from the conversion of elastic energy into heat.

iii) The bonding between cylinder and winding can be critical for the axial shear stress. In fact due to the differential thermal contraction and to the axial magnetic loads, different loads are axially applied to winding and cylinder. In the present design the mechanical coupling between winding and cylinder is supposed to be given by epoxy resin impregnation. The mechanical design should minimize the shear stress at the cylinder-winding boundary

The thickness of the cylinder was designed on the basis of a simple 1D stress analysis, which approximates the solenoid to a series of concentric shells of different materials. This analysis was made looking at each step of the solenoid life as explained in the previous item (i). A 1D analysis was also used to study the effect of the axial forces. After designed the cylinder and the end flanges the stresses due to magnetic loads were verified using ANSYS code. Different models for the stress analysis were used as better explained later.

In order to have a first indication of the cylinder thickness, we can considered that the minimum thickness is given by:

$$\Delta R = \frac{PR}{\sigma_{max}}$$

where P= 1.53 MPa is the magnetic pressure, R is the inner radius and σ_{max} the maximum allowable circumferential stress, which for Al alloy is 279 MPa (see APPENDIX B). Considering that the cylinder could be obtained by welding rolled plates, we have assumed as limit stress 160 MPa. ΔR_{min} results to be 15 mm. We have assumed a thickness of 30 mm, which both gives a high safety factors and reduces the elastic deformation of the pure aluminum as explained in the next section. The ends of the cylinder were further thickened for anchoring the supports.

4.3 Stress Calculations

This section is devoted to the stress analysis made in order to design the cold mass. The stress analysis was carried out through several steps:

I. The first part is related to a simple 1-D analysis to determine the circumferential stress at the winding due to the cooldown and to the magnetic loads.

II. A Finite Element analysis, with ANSYS code, was then performed to verify the stress due to the magnetic loads obtained by the 1-D analysis. A local model was used for this analysis, simulating very closely some parts of the winding.

III. A further 1 D analysis for thermal and magnetic axial loads was carried out.

IV. A Finite Element analysis for both axial and radial loads was carried out for the full winding, using two different models.

V. As final step the stress coming from the mechanical connection between cold mass and cryostat were calculated.

4.3.1 1-D Analysis For Circumferential Stress

This stress analysis is performed using a 1D shell model. The winding is considered as a set of concentric shells, each one being defined by an inner and an outer radius , a Young modulus depending on temperature E(T) and a thermal contraction coefficient a(T). The used mechanical properties are listed in APPENDIX B. Two neighboring shells are mechanically coupled. In our case the winding was subdivided into 6 shells; a detail of the winding model is shown in fig.4.2. Though analysis is carried out in the pure elastic limit, it is possible to limit the maximum stress of pure aluminum to 14 MPa simulating the transition from elastic to plastic regime. We have analyses 4 different situations : Construction, Cooldown, First Enerigization, First discharge . The results are summarized in Table 4.1 for the higher current region and in table 4.2 for the central region.

Construction - We have previously remarked that the construction can produce some residual stress at the winding. This assumption is mainly valid for winding on removable mandrels; the inner winding method should produce no significant residual stress. Only the resin impregnation could give some stress at room temperature. In fact the epoxy-resin polymerizes at 120 ^{o}C; on cooling down to room temperature the differential thermal contraction between winding and supporting cylinder can act in different way with respect the previous warm up. As a result we could have a light circumferential tension of the cylinder and a low compression of the winding. A preliminary calculation with our shell model seem to indicate very low values for the stresses, so that we assume no stresses at room temperature.

Cooldown - On cooling down to 4.5 K, the winding contracts less than the pure aluminum due to both the Rutherford and the insulation. According to the shell model the pure aluminum results to be in tension of 3 to 9 MPa depending on the region (ends or central) , the outer cylinder takes more stress

Table 4.1 Circumferential Stress for the end regions (MPa)

Layer	Constr.	Cool down	I Charge Al Elastic	-Al Plastic	Discharge (Al Plastic)
Outer Cylinder	-	20	57	70	30
Insulation	-	-69	-49	-37	-67
Winding Pure Al	-	9.6	39	14	-14
Rutherford	-	-140	-76	-43	-140
Winding Pure Al	-	9.7	40	14	-14
Insulation	-	-69	-49	-36	-68

Table 4.2 Circumferential Stress for the central region (MPa)

Layer	Constr.	Cool down	I Charge Al Elastic	-Al Plastic	Discharge (Al Plastic)
Outer Cylinder	-	17	36	45	21
Insulation	-	-71	-59	-37	-70
Winding Pure Al	-	6.7	25	14	-14
Rutherford	-	-147	-107	-64	-150
Winding Pure Al	-	6.7	25	14	-14
Insulation	-	-71	-59	-37	-70

(17 - 20 MPa) . Due both to the high differential thermal contraction between the Rutherford cable and the aluminum and to the low cross section of the Rutherford, the Rutherford cables take a lot of compression (150 MPa).

In the real situation it is possible that the Rutherford cable moves in the pure aluminum like in a viscous medium so that there is not a significant compression of the Rutherford nor a tension in the aluminum parts. Nevertheless we will make the complete analysis based on the original assumption , reserving this not elastic response of the conductor for final considerations.

Solenoid charge and discharge - We studied two situations: Elastic and plastic response of pure aluminum. If the pure aluminum were mechanically elastic, on charging the solenoid, it would be put in tension up to 39 MPa (starting from 9.6 MPa due thermal stress) at the end regions, whilst the cylinder comes in tension for 57 MPa (only 37 MPa are due to magnetic load) as shown in tables 4.1 and 4.2. If , as expected, the maximum stress of Al pure is limited to 14 MPa the outer cylinder takes further tension up to 74 MPa at the end regions and 56 MPa at the central region. These values of the stress at the supporting cylinder are well inside the safety region (Assumed maximum allowable strength of 5083 Alloy is 160 MPa). The only problem remains for the pure aluminum stabilizer, which at the first energization is stressed well beyond the elastic limit. Nevertheless we have the information that in case of elastic response the stress due to the magnetic load is 30 MPa. This means that on discharging the magnet, firstly the tension of pure aluminum is progressively reduced to zero; on continuing to discharge, the pure aluminum is elasticity compressed down to 14 MPa. When the current in the solenoid approaches to zero the pure Aluminum is lightly stressed in compression (see fig. 4.3) . From a theoretical point of view, if the yield strength of pure Al would be a little higher than 140 MPa, it would be expected that after the first charge the stabilizer never overcame the elastic limits, working between elastic compression (no field) and tension (applying the field). However the problem of Al plastic deformation is confined in two small parts of the high current density regions. In fact looking at the cold mass configuration , both the central region (where the magnetic load stresses the Al stabilizer for a maximum of 25 MPa) and the side regions, where the cylinder is thickened (the magnetic load causes a stress of pure Al of 23 MPa) would have to work according to the scheme described.

4.3.2 Finite Element Analysis For Circumferential Stress

In order to verify the results obtained by the 1-D analysis using the shell model, we have performed a finite element analysis, modeling a small part of each central and end regions as shown in fig.4.4. Only the magnetic loads were applied . The analysis was carried out considering the stress -strain curve of pure Aluminum as in Fig.4.3. We obtain for the stress at the outer cylinder 44 MPa at the end regions and 23 MPa at the central region. The pure aluminum of the winding is stresses over the elastic limit. Figures 4.5 and 4.6 shows the obtained stress distribution in the two regions considered. The resulting stresses of the outer cylinder are 20 % lower than predicted by 1D shell model, so that we can consider substantially correct the information coming from the 1D analysis .

4.3.3 1-D analysis For Axial Stress

In order to carry out a 1-D analysis for axial stress we considered the winding and the cylinder as two homogeneous structures mechanically coupled. As first step, it was necessary to define the average thermal and mechanically properties of the winding.

Winding thermal contraction The axial thermal contraction from 300 K down to 4.5 K of the winding a_w was calculated according to:

$$a_w = (a_{Al}\, D_{Al} + a_{ins}\ D_{ins}) / (D_{Al} + D_{ins})$$

were a_{Al} a_{ins} are the thermal contraction of Aluminum and Insulating material (fiber-glass epoxy transverse to the fibers in this case) and D_{Al} and D_{ins} are the radial thickness of the two materials. With $a_{Al} = 4.15\ 10^{-3}$, $a_{ins} = 6\ 10^{-3}$, D_{Al} = 3.2 mm or 5.8 mm and D_{ins} =0.4 mm, we obtained $a_w = 4.35\ 10^{-3}$ for the end regions and $a_w = 4.27\ 10^{-3}$ for the central region. Considering the axial length of the two regions the winding contracts $a_{wtot} = 4.3\ 10^{-3}$, i.e. 0.53 mm more than the supporting cylinder.

Young moduli The axial Young modulus of the winding E_w is given by

$$E_w = E_{Al}\, E_{ins}\ (D_{Al} + D_{ins}) \ / \ (D_{Al}\, E_{Ins} + D_{ins}\ E_{Al})$$

Using E_{Al} (T=4.5 K)= 78 GPa and E_{Ins}(T=4.5 K)=15 GPa we found E_w =53 MPa for the end regions and 61 MPa for the central region.

Winding and cylinder thermal contraction. It is useful to calculate what is the total thermal contraction of winding and cylinder considered as coupled structures.

Let be E_W and E_C be the Young modulus of the winding and supporting cylinder and a_W and a_C the relative thermal contractions from 300 to 4.5K,. the thermal contraction of the two coupled structures is:

$$a_{tot} = (S_W\, E_W\, a_W + S_C\, E_C\, a_C) / (S_W\, E_W + S_C\, E_C)$$

Where S_W and S_C are the axial cross sections given by

$$S_W = 2\ p <R_w>\ DR_w$$

$$S_C = 2\ p <R_c>\ DR_c$$

Where $<R_w>$ is the winding mean radius = 1498.5 mm , $<R_c>$ is the cylinder mean radius = 1530 mm , DR_w is the winding thickness = 31.8 mm and DR_c is the cylinder thickness = 29.8 mm.
We find a_{tot}= 4.21 10^{-3}

Axial deformation and shear stress As shown in Section 2.4 the integral axial force is 3.2 MN given by the difference between a compressive force of 4.8 MN applied at the end regions and an outward directed force of 1.6 MN. The total axial force causes a compressive deformation of the winding of 0.65 mm corresponding to a strain of 1.9 10^{-4} , so that the pure Aluminum would be stressed just at the elastic limit.
Considering the resistant structure formed by the winding + cylinder, the magnetic load causes a compressive deformation of 0.27 mm, corresponding to a strain of 8 10^{-4} mm/mm , well below the elastic limit of pure aluminum. This means that, though the winding could support the axial alone force, it is recommended to couple the winding to the outer cylinder in order to operate in safety conditions.
A more detailed analysis of the axial stress must be carried out considering the stress coming from the differential thermal contraction of winding and supporting cylinder. There is a mis-matching of 9 10^{-5} mm/mm (equivalent to 0.31 mm on the whole solenoid length). Since the winding contracts more than the cylinder, the thermal stress acts in the same direction of the magnetic stress.
The total strain due to the mis-matching between cylinder and winding is given by the strains due to magnetic load + thermal contraction i.e. 1.9 10^{-4} mm/mm + 9 10^{-5} mm/mm = 2.8 10^{-4} mm/mm . The axial force related to this strain is 4.7 MN, which is applied to the winding, with respect the cylinder . If winding and cylinder are mechanically coupled (using an epoxy gluing), there is a shear stress at the border of average value given by the ratio of the total force (4.7 MN) and the total border surface (33 m^2), i.e. 0.14 MPa,.
Unfortunately the magnetic force is not equally distributed along the axial direction. There is a peak force of strength 1.3 MN applied at the solenoid ends, for an axial length of 52 mm. The resulting shear stress is 2.7 MPa. The epoxy impregnation can give an adhesion between metallic surfaces separated by fiberglass-epoxy as high as 20 MPa, so that the actual value of the shear stress seems to be a factor 8 below the attainable values of the epoxy gluing.
The shear stress can be reduced if an axial pre-stress is applied. The important feature is that the pre-stress must be applied when winding and cylinder are independent structures, because in this way we move the winding with respect the cylinder , compensating the deformations occurring for magnetic load and thermal contraction. The pre-stress can be applied at the winding before curing for resin impregnation using bolts to push inward the end flanges against the winding (as shown in fig. 4.7) .

Nevertheless we do not think that the required pre-stress can be completely applied. In fact using metric M20 bolts in aluminum alloy, a maximum force of 20 kN can be reasonably applied. Using 64 bolts (as allowed by the circumferential allowed space) we can apply a force of about 1.3 MN. Looking at the end regions , this pre-stress reduces the shear stress of 4% . Generally speaking, the benefits of the pre-stress are taken by the complete winding-cylinder bonding. In some parts the shear stress will be strongly reduced, in other parts (as the ends) the shear stress will remain essentially the same.

4.3.4 Finite Element Analysis Of The Overall Magnetic Stress

In the previous sections we developed simple 1D analyses, for both axial and radial stress, carried out separately. In this section we show the results of a stress analysis for only magnetic load carried out by modeling the complete cold mass (winding + supporting cylinder), so that radial and axial stress are considered at the same time. Two different models were made using ANSYS code:

i) Shell model: In the first model the cold mass was idealized according to the shell model presented before. Fig.4.8 shows a detail of the model with the meshed regions. For this finite element analysis both the real stress-strain curve of pure aluminum and the simpler linear stress-strain curve were considered.

ii) Linear model with sub-model and very fine mesh: With the second approach the cold mass was idealized through two models: a “coarse” model , which modeled each turn but not the insulation and a fine sub-model which modeled the insulation too. The details of this model are given in Appendix F.

The results obtained for the circumferential stress confirmed the ones from the 1-D analysis and are shown in tale 4.3 .

Table 4.3 Results for the circumferential stress of FE analysis of the overall magnetic stress (MPa)

Model --->	Shell (Al elastic)		Shell (Al plastic)		Linear with sub-models	
Component	Side[1]	Center[2]	Side[1]	Center[2]	End[3]	Border[4]
Cylinder	34	22	52	21		
Insulation	5	5	24	6	5	13
Pure-Al	28	17	14	14		
Rutherford	50	38	67	36	20	59

Notes to the table:

1- Maximum stress at the side regions
2- Maximum stress at the central region
3- Stress at the solenoid ends
4- Stress at the border between high and lower current density regions

The main results for overall stress are:

Outer cylinder - The maximum hoop stress (50 MPa) is located at the high current density regions, whilst at the central zone the stress is limited to 20 MPa. The radial displacement at the side is 1mm and at the central region 0.5mm as shown in fig.4.9 .

Aluminum stabilizer The hoop stress is limited to 15 MPa. The maximum axial stress at the aluminum stabilizer is 10 MPa.

Rutherford conductor The maximum von Mises stress is 59 MPa.

Shear stress The maximum shear stress in the insulation ranges from 1.1 MPa to 4.5 MPa depending on the model.

4.3.5 Overall Stress State In Cold Mass

In addition to the calculations above, we have performed a Finite Element analysis of the cold mass including the effect of gravity , magnetic offsets and supporting system. These calculations are described in more detail in APPENDIX C. This model (shown in figure 4.10) consists of a cylinder thickened at the ends and subject to the loads and supports described below. 4-Noded shell elements were used with the ANSYS program version 5.0. The windings were assumed to contribute to the stiffness and strength of the structure. Output studied included Von Mises equivalent stress and deflections. In all cases we were careful to look at both surfaces of the shell elements as well as at the center plane, in order to capture bending effects.

The support directions are shown in figure 4.11.

The most obvious features of the results is that the stress due to the magnetic pressure dominates, none of the other loads has a particularly large effect except for the axial magnetic force (loadcase 6). . Note that this axial force is the force acting on each individual conductor, as distinct from the net axial forces of 20t and 10t due to misalignment and asymmetry. The stress value of 39 MPa corresponds broadly with the value for the simple 1D analysis given above, remembering that this was a linear analysis which did not take account of the increase in stress in the outer shell when the conductor yields.

Table 4.3 Supports

Supports		
Type	Location and number	degrees of freedom
Axial	6, evenly distributed around one end of the coil, with two on the horizontal center plane	“Z”
Radial	4 each end, at +/1 45 degrees from the horizontal center plane	tangential movement = 0

Table 4.4 Loads and results (all loads included magnetic pressure, gravity)

Case	Description	Additional loads	Max. deflection (mm)	Max. stress (MPa)
1	Worst case axial loads with earthquake	30t plus 1.2g, axial	0.86	38.7
2	Worst case sideways loads with earthquake	20t plus 1.2g, sideways	1.04	39.6
3	Worst case vertical loads with earthquake	20t plus 2g, vertical	1.13	39.9
4		vertical 20t	1.02	39.5
5	Worst case of earthquake and magnetic force direction - puts most load onto only two supports each end	2g vertical plus 20t at 45 degrees	1.13	39.9

In addition to the axial offset magnetic load, there are large forces acting on the conductors which are reacted within the cold mass structure. In order to assess the effects of these, we tried one load case with those forces included:

6	Nominal load, but with axial magnetic forces included.	vertical 20t, axial 10t	1.02	39.5

4.3.6 Concluding Remarks

The general conclusions of this section dedicated to the stress analysis of the cold mass are:

i) Due to differential thermal contraction and to magnetic loads (which are the dominant effects) the designed cylinder is put in tension for a maximum value of 70 MPa , i.e. 43 % of the Yield strength.

ii) The Aluminum stabilizer of the conductor is stressed over the elastic limit only at the first solenoid energization. In the following magnetic cycles the stabilizer works between the compressive and tensile elastic limits

iii) The axial stress can be easily transmitted from the winding to the cylinder at a low value of shear stress ranging from 1.1 MPa to 4.5 MPa, depending on the model used for calculation. These values are much lower than the attainable shear stress given by an epoxy gluing (20 MPa)

iv) The application of a moderate axial pre-stress (through a force of 1.3 MN) could help in reducing the shear stress at the winding-cylinder boundary.

v) An interesting comparison can be made between the mechanical behavior of BABAR solenoid and a working thin solenoid as CDF. The important parameter is the radial displacement due to the magnetic load. For BABAR this displacement is 1 mm maximum, as shown above. For CDF the designed radial displacement was 0.67 mm as described in the CDF solenoid technical design. Nevertheless for CDF the aluminum stabilizer was considered elastic up to high value of the stress (40 MPa). In our case, Table 4.1 shows that for elastic response of pure aluminum the magnetic loads would cause a maximum stress on supporting cylinder of 37 MPa, corresponding to a radial displacement of 0.69 mm, very close to the design result of CDF.

4.4 MANUFACTURING METHOD

In the previous section we have stressed the importance of a good bonding between winding and supporting cylinder. There are two ways to couple the two structures corresponding to two different manufacturing approaches.

i) The first approach consists in the shrink-fit technology: the winding is wound and impregnated onto a removable mandrel, then machined and enclosed inside the outer cylinder through a shrink-fit operation. In this case the coupling is given by the mechanical interference. Nevertheless the winding could move with respect the coil so that the application of axial preloads is recommended.

ii) The second approach is based on the inner winding. The conductor is wound directly inside the cylinder, the winding is then mechanically compacted and impregnated. In this case the bonding is given by the epoxy-adhesion. In both solutions it is possible that the winding axially moves with respect the cylinder causing heat dissipation and premature quenching. The design should minimize such events both reducing the possible movements and increasing the stability against thermal disturbances.

We chose the solution ii , i.e. the winding should be made by the inner winding method, so that the cylinder is coupled to the winding by resin impregnation. This choice was determined by the fact that several thin solenoids, still working, were made by using this technique (ALEPH, DELPHI, ...). Furthermore this solution reduces the costs and risks of the solenoid manufacturing. In fact the solution based on the shrink-fit operation requires more tooling and presents some critical steps as the machining of the outer surface of the solenoid and the shrink-fitting.

After the winding completion, the turns should be axially pressed applying a pre-stress of strength 1.3 MN or more depending on the final design of the end flanges.

We have stressed that the axial force is transmitted to the outer cylinder through the epoxy bonding. Though the predicted shear stress is low (max 4.5 MPa), the epoxy impregnation must be very carefully carried out in order to minimize epoxy voids, which could lead to mechanical disturbances.

CHAPTER 5

5. CRYOSTAT

5.1 DESIGN CONCEPT

The cryostat concept is shown in figure 5.1. The cryostat consists of an annular vacuum vessel equipped with radiation shields and superinsulation. The vessel is designed in aluminum alloy AL5083.

5.2 VACUUM VESSEL

The vacuum vessel (see figure 5.2) consists of two cylindrical shells with end flanges. There are brackets between the inner and outer shell at the ends, which are the fixing points for the support rods from the cold mass.

The vacuum vessel is designed to satisfy a number of basic criteria:

1. to support vacuum loads to standard vessel codes
2. to carry the internal cold mass and radiation shield weight through the insulating supports
3. to operate with deflections <2mm under all loads when mounted in the yoke structure.
4. to operate under defined seismic loading.

The vacuum vessel also has to transfer the earthquake loads acting on the central detectors in the axial direction, to the iron yoke (see below for further discussion of the support system). The scheme is shown in figure 5.3, and because of the large forces involved it is one of the more significant load cases.

As for the design of the cold mass, Finite Element calculations backed up by simple checks have been done. These are described in the report number RAL/ASD/CME/misc/010 and in Appendix D, and are summarized for convenience here. When looking at the stresses, we distinguished between maximum localized stresses (which could be 'designed out' at the detail design stage) and maximum general stress levels (which would require significant design changes if they were to be removed) (see also figure 5.4).

Table 5.1		
Model details		
Inner shell	Mean Diameter Thickness Length	1395mm 10mm 1875mm to center of end flange
Outer shell	Mean Diameter Thickness Length Thickened region	1695mm 30mm 1875mm to center of and flange 50mm thick for 200mm each end
End flanges	Thickness	50mm
Supports - axial	Four at one end, at ±30° to the vertical	"Z" - axial. See note below.
Supports - radial	One each end, at one side, on the horizontal center line	"X" - sideways. See note below.
Supports - vertical	Two each end, on the horizontal center line	"Y" - vertical

Note: There will in fact be Eight axial supports (four each end) and four radial supports (two each end), but they will be made so that they only act in one direction. See the description of the supports in the next section.

Table 5.2					
Load cases and results. All load cases included 1g downwards and vacuum loads.					
Case	Description	Additional loads	Max. deflection	Max. general stress	Max. local stress
1	Nominal	None	0.4	16	23.9
2	Earthquake - vertical	2g downwards	0.5	18	32.2
3a	Earthquake - sideways	1.2g +X direction	1.47	23	40.6
3b	Ditto - opposite direction	1.2g -X direction	1.72	20	58.6
4a	Earthquake - Axial Middle support	1.2g axial plus 60t load from detectors at middle radius of end flange	1.05	16	46.1
4b	Earthquake - Axial Inner support	1.2g axial plus 60t load from detectors at inner radius of end flange	1.43	25	42.1

The overall deflections are less than 2mm in all cases. This is acceptable in terms of the job the vacuum vessel has to do.

The stresses are in all cases less than 65MPa, which is the design stress for 5083 specified in the Pressure Vessel standard BS5500.

In terms of this design study, these results show that the design is feasible. In order to satisfy pressure vessel regulations, it will probably be necessary to specify the geometry of the fixings, etc, more completely, and make a more careful assessment of the stresses in those regions.

Cases 4a and 4b enable a conclusion to be reached about the flange on the inner detectors, which bears on the vessel to transmit axial earthquake loads (see also section 6.1 and figure 6.3 below). The bearing points may be anywhere between the middle of the vacuum vessel end flange, and its inner edge.

5.3 RADIATION SHIELDS

The radiation shields can be of a conventional design, consisting of aluminum alloy plate 5mm thick, rolled to the appropriate sizes with cooling circuits attached for helium gas circulation. They may be supported by a simple tie rod system basically similar to that for the cold mass, the shields are illustrated schematically in figure 5.5.

5.4 SUPERINSULATION

The cryostat should be equipped with superinsulation using stand and techniques at the following levels:

300K - 80K	typically 30 layers of superinsulation
80K - 4K	between the coil and radiation shield the use of superinsulation is optional. Clean surfaces with low emissivity will give adequate performance (H1 Solenoid) otherwise a maximum of 5 layers of superinsulation may be installed.

CHAPTER 6

6. SUPPORTS

6.1 DESIGN CONCEPT

The concept developed (see figure 6.1) here features "internal" supports shown in figure 6.2, which support the cold mass within the vacuum vessel, and "external" supports shown in figure 6.3, which support the vessel and the inner detectors from the iron yoke.

The magnetic pressure and the vacuum loads are reacted within the cold mass and the vacuum vessel and require no additional support.

6.2 LOADS

6.2.1 Overview

The following loads are considered:

The **magnetic offset load**, due to the fact that the flux return is asymmetric. This load acts axially, on the coil.

The **magnetic alignment load,** caused by errors in the geometric alignment of the components of the magnet, or by variations in magnetic properties of the flux return. This acts on the coil, and could act in any direction.

The **weight** of the cold mass, radiation shields, and vacuum vessel, (together, "the solenoid") as well as the weight of the inner detectors.

Earthquake loads on the solenoid and the inner detectors. These could act in any direction but with different magnitudes in the different directions. For the purposes of the support system design, we have treated the **axial, radial** (or sideways), and **vertical** components separately (see also the section/appendix on earthquake loads, below)

6.2.2 Details

The masses and loading criteria assumed in this design study are outlined below, and the resulting loads are given in table 6.1.

<u>Masses:</u>

- Vessel mass = 5.2t, cold mass = 7t, radiation shields etc = 1.5t, detector mass = 50t.
- "Cryostat" means the vessel plus the radiation shields, 6.7t.

Other criteria:

- Earthquake loads are 1.2g sideways and 2g vertical. A fuller explanation of this is given in Appendix A.
- Alignment error of 2cm gives forces up to 20t. See section 2.5.
- Asymmetry of the flux return gives axial force of 10t. See section 2.5.

Table 6.1

	On cold mass (7 tonnes)	On cryostat (6.7 tonnes)	On detectors (50 tonnes)	Combined load on supports
Max. radial loads:				
g forces (1.2g)	8.4	8.0	60	
magnetic alignment errors	20			
Total	28.4	8	60	96.4t
Max. axial loads:				
g forces (1.2g)	8.4	8.0	60	
magnetic loads due to known geometry	10			
magnetic alignment errors	20			
Total	38.4	8	60	106.4t
Max. vertical loads:				
weight	7.0	6.7	50	
g forces (2g)	14.0	13.4	100	
magnetic alignment errors	20			
Total	41	20.1	150	211.1t

6.3 INTERNAL SUPPORTS (TIE RODS)

Inside the vacuum vessel, the cold mass is supported by six **axial tie rods** and eight **radial tie rods**. The concept is shown in figures 6.1 and 6.4. The six axial tie rods are positioned at one end of the cold mass, equi-spaced around the circumference. They take the axial forces, magnetic and earthquake. The eight radial tie rods are positioned four each end at 45° from the horizontal, aligned tangentially. They take the vertical and sideways forces, earthquake and magnetic.

6.3.1 Design Criteria

The design criteria we used were:

- **Strength**. The direct stress in each rod must be less than half the yield stress under normal loading, AND less than half the ultimate stress under earthquake loading. See also Appendix A)
- **Conduction.** The heat conduction over half the rod's length (assumed) must be acceptable between 80K and 4K. Total for all supports should be less than 10W.

- **Stability.** The buckling load of the rods must be at least twice the highest load they will see.

- It is common in design of structures to use a factor of safety of five on buckling, because the method used to calculate bucking loads in complex structures (the “bifurcation” or “eigenvalue” method) is known to be non-conservative. With the buckling of simple ball-ended rods such as these, however, we feel that the simple Euler formula we have used will be accurate enough to allow the lower safety factor. Tests could be used to confirm this. Clearly whoever does the final design will have to reconsider the matter, taking into account the requirements of relevant design codes. The stability of the vacuum vessel is an entirely different matter and should not be confused with the stability of the support rods.

- **Size.** The size of the rod ends may be a problem, and the overall length must be possible to accommodate in the design.

In order to meet the criteria we propose a Titanium alloy, which has a good combination of high strength and low thermal conductivity. The stiffness is moderate. The final design should consider the use of a tougher alloy than the one we have used for these initial calculations. The ELI (extra low interstitial) form may be suitable. We have also designed rods which are smaller at the ends than along the bulk of their length. This allows us to use reasonably small end fittings, whilst obeying the buckling criterion.

The design we show here will meet all the criteria. It may be, however, that changes need to be made at the detailed design stage. An option which could be used at that stage is to make the rods hollow - this allows low thermal conductivity with high buckling resistance.

6.3.2 Axial Rods

There are 6 rods at the turret end only. The magnetic offset load acts in such a way as to compress the rods. The actual load the rods will see depends upon how the other loads (magnetic alignment and earthquake) are applied.

Axial design load in normal operation is due to magnetic forces. The 10t magnetic offset load acts so as to compress the rods.

Max compressive load in normal operation = 20 + 10 = 30t

Max tensile load in normal operation = 20 - 10 = 10t

Max. load in earthquake (compressive) = magnetic load + 1.2*7t = 38.4t

The maximum tensile load the rods will see is given by the combined effect of alignment forces plus earthquake loads. The magnetic offset force will tend to reduce this tensile load, but since it may be less than 10t and may in any case be absent under earthquake conditions (the magnet may not be working!), it is safer to ignore it:

Max. load in earthquake (tensile) = 20t + 1.2*7t = 28.4t

If stability (buckling) is a problem at the detail design stage, then the rods could be moved to the other end of the solenoid but that has implications for the design of the cryogenics between the turret and the cold mass because of the contraction of the cold mass. As mentioned above, another possibility would be to use hollow rods.

6.3.3 Radial Rods

The design is based on eight radial tie rods, four at each end (see figure 6.2). The loads they each see will depend upon the direction of the magnetic alignment errors and the earthquake load vector. See figure 6.5, in which vector diagrams are given showing how the loads add up. This analysis leads to the following maximum loads:

Tensile: 39t shared between four rods

Compressive: 29t shared between four rods.

The maximum nominal (non-earthquake) load happens when the magnetic alignment forces act at 45° (compare with case 2 in figure 6.5), and is given by

Max nominal loads: 20t + .707 × 7t = 25t shared between four rods

6.3.4 Sizes, Details Of Stresses, Heat Loads, Etc.

Applying the design criteria and loads given above, the following design was arrived at:

Table 6.2			
	Units	Axial	Radial
Loads			
Nominal load - tensile	tonne	10	25.0
Nominal load - compressive	tonne	30	
Rods to resist nominal load		6	4
Quake load Tension	tonne	28.4	39.0
Compression	tonne	-38.4	-29.0
Rods to resist quake load		6	4
Material			
Material	Titanium alloy 6%Al, 4%V		
Ultimate stress	MPa	1000	
Yield stress	MPa	900	
Conductivity integral 80K to 4K	W/m	213	
Rod sizes			
Rod diameter - nominal. This is the diameter of the rod over all of its length except the ends, where it is turned down to M20.	mm	25	25
Rod length	mm	350	300
Rod diameter in thread root	mm	16.9 (M20)	16.9 (M20)
Stress, buckling			
Stress under Earthquake load in thread root			
Tension Compression	MPa	211 -285	434 -323
Factor of safety on ultimate stress under earthquake load		3.5 (compressive)	2.3 (tensile)
Factor of safety on buckling (using nominal diameter)		2.4	2.9
Thermal conductivity			
Rods in conductivity calculation		6	8
Total heat load over half the length of the rods	Watts	3.6	5.6

As the table shows, we have developed at a design which satisfies all the design criteria. The axial rods are closest to the stress limits with a factor of safety of 2.3 (we require at least 2.0) The radial rods have the lowest buckling margin with a factor of 2.3 (we require 2.0). This is a simple design, however, and if it were felt desirable to increase the safety margins, or the detailed design introduced other constraints, then there are several options available to the designer, two of which have been mentioned above.

6.4 External Supports

As described in more detail below, the design features four main support brackets, which take the vertical and radial loads. Because these could not be designed to take the axial loads, there are also eight subsidiary support brackets to serve that function.

Vertical and radial loads:

Four **main support brackets** which are fitted to the iron yoke at each end, on the horizontal center line (see figure 6.6). These take the vertical and radial forces on the solenoid and the inner detectors. The inner detectors rest on these brackets independently of the solenoid, so that the solenoid does not ‘see’ any of the vertical or radial forces on the inner detectors. It is not possible to do the same for the axial forces on the inner detectors. This is because the radial gap between the inner detectors and the iron yoke would have to be spanned by some structure which would be loaded in bending. Within the 100mm axial gap available, we could not design a structure to take those loads. For example, a solid plate 60mm thick with 60 tonnes at it’s inner radius of 1.3m, simply supported at it’s outer radius of 1.8m, has stresses of 200 MPa, ignoring any cut-outs, etc. The design we have used depends upon the vacuum vessel of the solenoid to span that gap as follows.

Axial loads:

Eight **subsidiary support brackets** are fitted to the iron yoke each end, at positions 30° from the vertical (see figure 6.7). These react axial loads, and are needed because of the magnitude of the axial earthquake forces on the inner detectors. Forces coming onto the end flange of the vacuum vessel (see figure 5.3) are reacted along the length of the vessel to the subsidiary brackets at the far end. This arrangement means that the plates fixed to the solenoid end-flange are loaded principally in shear. If we tried to react the loads at the same end as they appear, then the bolts holding the brackets to the end plate would be very highly loaded.

General:

Simple calculations have been done on the proposed external supports which show that they will survive the worst loads likely to act. These are detailed in Appendix E.

Note that the external supports must be fixed to the iron yoke, and we believe that some work is needed on the local design of the yoke to ensure that it can take the loads imposed on it.

CHAPTER 7

7. CRYOGENICS

7.1 BABAR Overall Cryogenic Concept

The BABAR Experiment will require cryogenic supplies to the superconducting solenoid and two future superconducting beam line focusing magnets. It is proposed to install helium plant consisting of a helium liquefier/refrigerator, a 4000 liter supply dewar and a distribution valve box adjacent to the experimental hall to meet the experiment cryogenic requirements. The helium plant will also be designed to supply an auxiliary dewar/trailer for all other SLAC experiments. The 4000 liter dewar will provide approximately 30 hours of autonomous operation in the event of refrigeration failure. The helium plant will be located approximately 60m from the magnet. The conceptual layout is shown in Figure 7.1.

A detailed description of the plant is not given in this report, however a schematic of the plant is shown in figure 7.6. Table 7.1. gives the major parameters of the cryogenics plant and Figure 7.2 the configuration of the compressor/cold box system. The interface between the solenoid and the refrigeration plant will be at the turret valve box which will contain the liquid helium reservoir for the thermo-siphon and the current leads. Connecting cryogenic and helium gas feed/return lines will be supplied up to this interface by SLAC, - the cryogenic concept and cool down times are based on the use of these existing lines.

The refrigeration plant will be equipped with control systems for all operation. The solenoid will be required to interface with this control plant through defined protocols.

7.2 Solenoid cryogenics concepts

This section outlines the proposed cryogenic concepts for the BaBar Solenoid. The detailed cryogenic and control configuration will need to be agreed between SLAC and the manufacturer. This final configuration will take into account details of the solenoid design and operational requirements, details of the final cryogenic plant and a full assessment of system operation under normal and fault conditions.

The solenoid cryogenic circuit concepts are illustrated in Figures 7.2. and 7.3. The cryogenic circuit concept assuming the cool down is carried out by mixing gas streams is shown in figure 7.4. The solenoid will be equipped with a chimney and turret valve box mounted at the backward end of the yoke as shown in Figure 7.5. All cryogenic and electrical services will pass through the chimney and terminate in the turret valve box. The turret valve box will represent the cryogenic and electrical interface with the auxiliary systems. It is proposed to mount the solenoid vacuum system adjacent to the turret valve box and to pump through the turret chimney.

The turret valve box will contain an intermediate helium reservoir through which the thermo-siphon cooling will operate. The turret valve box will also contain the bayonet connections for coolant supply and return for shields and cold mass. The helium reservoir will contain the current leads. The turret valve box will be the interface to the transducers within the solenoid.

The turret valve box will be connected to the distribution valve box by two main flexible cryogenic transfer lines.

- Line 1 will contain the helium supply to the turret helium reservoir and is shielded by the gas return from the turret helium reservoir.
- Line 2 will contain the gas supply to the radiation shields and is shielded by the gas return from the shields

These lines exist and the cool down analysis has been tailored to use these lines.

Since the total liquid helium inventory in the coil / turret helium reservoir system is comparatively small in comparison to the capacity of the liquefier and its storage dewar, then to avoid unnecessary perturbation to the cryoplant the liquid will be vented to atmosphere in the event of a coil quench.

7.2.1 Cool down

Cool down of the coil cold mass and radiation shields from room temperature will be made by circulation of He gas. The gas may be supplied to the coil and the shield either directly from the refrigerator at the specified temperature in a controlled cool down or by mixing shield supply gas at approximately 40K from the cryoplant and 300K gas from the compressor system to cool down the shields and by mixing coil supply gas at approximately 4.5K from the cryoplant and 300K gas from the compressor system to cool down the coil. The gas mixing option recommended is that the mixing be carried out in the turret valve box since this minimizes the diameter of the cryogenic transfer lines.

Typical parameters for the cool down of the solenoid are given in Table 7.2. The full spreadsheet analysis is given in Appendix 1. A constant cool down rate is assumed during the cool down.

This analysis shows that with an initial mass flow rate of ~5gm/sec rising finally to ~10 gm/sec in the coil circuit the coil cold mass can be cooled to 5K in approximately 7 days. The corresponding mass flow for the shields is ~2 gm/sec rising to ~4 gm/sec. The cool down analysis is based on a maximum temperature difference across the shield and the coil of 40K in order to minimize thermal stresses. Cool down curves for the coil and shields are shown in Figure 7.11.

7.2.2 Operational Mode

After cool down the cryogenic system will be switched to operational mode where the coil is cooled by circulation of 2-phase liquid helium. The conceptual layout of the cold mass cooling circuit is shown in Figure 7.7. The design is based on the thermo-siphon technique established for ALEPH and CLEO II. The coil circuit will be fed from the turret dewar through a large bore manifold at the bottom of the force support cylinder. The cooling circuits are welded to the cylinder surface with a spacing of ~0.3m. The cooling circuits terminate in the upper manifold which connects to the turret helium reservoir through a phase separator.

The thermo-siphon cooling circuit will be designed for high flow rates to ensure a high quality factor for the helium - low vapor content. The design should allow for a minimum flow rate of ~30g/sec.

Typical parameters for the thermo-siphon circuit are given in Table 7.3.

The circuit will be equipped with pressure relief valves for safe operation during quench.

The turret helium reservoir will be designed to contain the minimum helium volume compatible with reliable operation. The total estimated volume of the coil and turret helium reservoir in real terms is < 80 litters.

7.2.3 Heat Loads

The estimated heat loads for the solenoid are given in Table 7.4. Eddy current heating in the support cylinder will lead to additional heat loads during charging of the solenoid. For a solenoid charging time of 30 minutes the estimated transient power is ~2 watts.

7.3 Cooling Circuit Design

7.3.1 Radiation Shields

Using the estimated heat loads calculations show that five equi-spaced cooling circuits will be required on the radiation shields. The internal diameter of the circuits which gives the best compromise between the cool down and running conditions is ~10mm. The manifolds feeding and collecting the cooling circuits require an internal diameter of ~30 mm and the line diameter through the chimney requires an internal diameter of ~20 mm.

7.3.2 Coil

Again using the estimated heat loads calculations show that 10 equi-spaced coils will be required on the coil. The internal diameter of the circuits which gives the best compromise between the cool down and running conditions is ~10mm. The manifolds feeding and collecting the cooling circuits need to have an internal diameter of ~45 mm and the line diameter through the chimney needs to have an internal diameter of ~25 mm.

7.3.3 Transfer Lines

The design presented here is based on the use of existing SLAC flexible transfer lines which have a 10 mm bore. Initial analysis suggests that these lines will be suitable for BABAR.

7.4 LOCAL CRYOGENICS

7.4.1 Services Chimney

The solenoid will be connected to the turret dewar through the cryogenics chimney shown in concept in Figures 7.8. and 7.9. The chimney will be located at the backward end of the solenoid and accommodated in a customized channel in the backward end door. The chimney design is based on an outer diameter of 350mm. The cross-section of the chimney is shown in concept in Figure 7.10. The chimney will contain the following components:

i. superinsulation layers

ii. outer 80K shield cooled by main solenoid shield return gas.

iii. 4.5K feed and return

iv. 40K feed to the solenoid shields

v. current feeds to the coil

vi. instrumentation wiring

The current feeds will be cooled indirectly by the thermo-siphon feed and return and will terminate in the turret dewar at the base of the current leads.

7.4.2 Turret Valve Box and Helium Reservoir

The turret valve box and helium reservoir dewar is shown in concept in Figure 7.5. The associated parameters are given in Table 7.5.

7.5 CRYOGENIC CONTROL

The cryogenic control system and must be capable of ensuring the safe working of the solenoid during cool down, normal operation and quench conditions. Details of the control interface and the operational protocols will need to be agreed between the solenoid manufacturer and SLAC.

7.6 FAULT CONDITIONS

7.6.1 Warm Up after a Fault Condition

Calculations show that above a radiation shield temperature of ~110K the radiation heat loads to the shield and the coil are equal. The radiation heat load is the dominant heat load so above this temperature the temperatures of the shield and the coil become locked together and warm up as one entity. This is illustrated in Figure 7.12.

Table 7.1. Parameters of cryogenic plant.

Normal running	
Shield	351 W at 60K
Coil	39 W at 4.5K
Current leads	0.75 g/s (22.5 litters/hr)
Cool down	
Peak compressor output	~14 g/s
Peak 40K mass flow rate	~4 g/s
Peak 4.5K mass flow rate	~10 g/s
Rated power at 4.5K	**141 W**

Table 7.2. Parameters governing cool down of solenoid.

System	
Cool down time	~7 days
Peak demand from compressors	~14 g/s
Radiation shields	
Cold mass	1000 kg
Peak 40K demand	~4 g/s
Coil	
Cold mass	7000 kg
Peak 4.5K gas demand	~10 g/s

Table 7.3. Thermo-siphon parameters

Operational driving head	~3 m
Operational driving pressure	353 Pa
Diameter of turret chimney lines	25 mm I.D. ~30 mm O.D.
Diameter of manifolds	45 mm I.D. ~50 mm O.D.
Diameter of coil cooling circuits	10 mm I.D.

Table 7.4. Estimated heat loads on solenoid.	
Coil	
Radiation heat load	29 W
Conduction heat load from supports	10 W
Transient heat load while charging coil	2 W
Radiation shields	
Radiation heat load	301 W
Conduction heat load from supports	50 W
Current leads	
Current	5000 A
Heat load	15 W
Helium consumption	.75 g/s (21l/h)

Table 7.5. Turret valve box and helium reservoir parameters.	
Turret valve box	
Height of outer vessel	1500 mm
Diameter of outer vessel	1000 mm
Diameter of radiation shield	950 mm
Helium reservoir	
Height	1250 mm
Diameter	500 mm
Length of current leads	750 mm
Working volume (height)	55 litters (300 mm)

CHAPTER 8

8 VACUUM SYSTEM

8.1 Design Criteria

When designing the vacuum pumping system for the BABAR Solenoid the following criteria have to be considered:

- The system will have a large volume.
- During the initial pump down, if the system has been let up to air and when the magnet is warm there will be a high background out-gassing load 98% of which consists of water vapor.
- When the magnet is cold this out-gassing load will be negligible.
- A coating of oil will have disastrous effects on the effectiveness of multilayer superinsulation.
- The system will need to be roughed down to a relatively low pressure before the high vacuum pumps can be brought on line.
- A base pressure of at least 1×10^{-4} mbar must be attained by the high vacuum pumps before cool down can commence (Reference 1).
- The vacuum pressure will rise when the warm up of any internal component occurs.
- The high vacuum pumping system used to pump the insulating vacuum of the cryostat down to a pressure of 1×10^{-4} mbar when the system is warm must produce the necessary pumping speed **at the vacuum vessel**. This means the pumps must be connected to vacuum vessel by a pumping line which does not seriously reduce the pumping speed of the pumps themselves. In general this means a short, large diameter pumping line must be used.
- When the magnet is cold and the magnetic field is on a pumping system with a lower pumping speed could be used as the out-gassing load will be negligible.

8.2 Choice of High Vacuum Pumps

Superconducting magnets used in particle detectors have traditionally used diffusion pumps and more recently turbomolecular pumps for the insulating high vacuum of the cryostat. It is proposed to use turbomolecular pumps since used correctly they can be regarded as oil free. SLAC will be receiving a large number of them from SSC and it is intended to use whatever meets the BaBar solenoid insulating vacuum specification.

8.3 Design

The major system parameters for the vacuum system are given in Table 8.1.

It is proposed that the vacuum vessel is pumped down in two phases. The first phase is to rough pump the system to a pressure of 0.1 mbar. The second phase is to then to bring the high vacuum pumps on line and pump the system down to 1 x 10^{-4} mbar which is the pressure at which cooling can be deemed to start. It is intended to use two identical pumps. Both will be used to pump the system down to 1 x 10^{-4} mbar. When the system is cold and the out-gassing load is very much smaller one of the pumps can be switched off and isolated from the system. This pump can then take on the role of a redundant pump which can be brought on line if the other pump fails.

8.3.1 Design Calculations

Calculations show that the vacuum system of the BABAR Solenoid needs to produce a pumping speed at the vacuum vessel which is greater than 400 litters/sec.

The calculations have all been carried out using spreadsheet techniques and are summarized as follows:

- Figure 8.1 gives the expected out-gassing load.
- Table 8.2 gives the basic parameters of the pumping system.
- Figure 8.2 gives the expected pump down curve.

8.4 System Definition

SLAC is scheduled to receive a number of Varian V450 turbomolecular pumps and controllers along with their backing pumps. Two set of pumps will be used to pump the magnet/turret vacuum space. Varian V450 turbomolecular pumps have a pumping capacity of 470 liter/sec at pressures less than 1 x 10^{-3} mbar and ~300 liter/sec at 1 x 10^{-2} mbar. The backing pumps are expected to each have a pumping speed of ~18 m^3/hr which is the pumping capacity of Varian’s recommended two stage SD-300 backing pump. Two of these backing pumps have more than sufficient pumping capacity for the initial “rough” pump down of the solenoid vacuum vessel. The use of two identical units provides redundancy and the ability to operate only one system after magnet cool down.

8.5 System Installation

Figure 8.3 shows the vacuum system for the BABAR Solenoid.

Both pumping systems will be installed on the top deck of the solenoid close to the turret dewar. The turbomolecular pumps will be connected directly to the solenoid/dewar common interconnecting chimney in a vertical orientation either under or above the electronics platform. They will require some more metal shielding and will be fan cooled. The backing/roughing pumps will require spring/damper mounting for vibration isolation. The turbomolecular pumps have less than 0.02 micron vibration levels and it is not anticipated that they will require vibration damping.

All controls, including the turbomolecular pump controllers, and the instrumentation will be located in the liquefier control room, although some instrumentation may be provided near the turret dewar.

8.5.1 Location/Mounting

The proposed location and mounting of the turbomolecular pumps is shown in Figures 8.4. and 8.5. There are two possible variations either over or under the electronics platform

8.6 System Operation

Referring to Figure 8.3. valves designated MV are mechanical valves and those designated SV are solenoid controlled valves. Valves MV 11/22 are normally open and capped. Valves MV 21/22 are normally open. All the solenoid controlled valves are closed when not powered. Gauges designated TC are thermocouple gauges measuring in the range 10 - 0.02 mbar and those designated PEN are Penning gauges measuring in the range < 1 x 10^{-3} mbar.

The operational sequence is as follows:

1. Rotary pumps RP 1 and RP 2 are started.
2. When the pressures on gauges TC 12/22 read less than 0.1 mbar and if the pressure on gauge TC 31 reads greater than 0.1 mbar valves SV 13/23 open and roughing down of the vacuum vessel commences.
3. When the pressures on gauge TC 31 reads less than 0.1 mbar valves SV 13/23 close and then valves SV 12/22 open.
4. When the pressures on gauges TC 12/22 read less than 0.1 mbar turbomolecular pumps TP1 and TP2 start.
5. When TP1 and TP2 reach full operational speed gate valves SV 11/21 open.
6. When the pressure measured on PEN 31 reads less than 1 x 10^{-4} mbar cool down can start.
7. Finally when the pressure measured on PEN 31 reads less than 1 x 10^{-7} mbar after cool down has started one of the valves SV 11/21 can be closed and its corresponding turbomolecular pump and backing valve can be switched off and closed respectively. The vent valve will open automatically to let the turbomolecular pump up to a predetermined pressure to prevent oil migration to the top of the pump nearest the pumping line.

8.7 Turret Valve Box

The turret valve will be connected directly to the solenoid vacuum vessel forming a common system. Under this scheme the turret valve box will not require a separate vacuum pumping system.

8.8 Transfer Lines

It is proposed that the vacuum insulation space of each of the flexible transfer lines has its own pump out port and if deemed necessary their own dedicated pumping systems. These transfer lines are the most exposed components and need to have self contained vacuum pumping systems to ensure they do not affect the safety of the magnet system.

8.9 Relief Valve

The purpose of the relief valve on the vacuum system common to the solenoid and the turret dewar is to relieve any pressure rise which might occur if a rupture occurs in the internal cryogenic system thus leading to a potentially high pressure. It is a large bore device designed to open at a differential pressure of more than ~100 mbar. Its location is shown in Figures 8.4. and 8.5.

8.10 Control System

The control system will have two modes automatic and manual.

The automatic mode will run through the sequences detailed in Section 8.6. It will be able to pick up the sequence at any stage so that all that is required to initiate the pump down is to switch from manual to automatic.

The manual mode allows any operation to be carried out by manual switching but does not allow strategic interlocks to be overridden.

Manual mode will be used for the initial rough pumping from atmospheric pressure if lots of moisture is present. Backing valves SV 12/22 are kept closed when the roughing valves SV 13/23 are opened. The system is pumped down to a low a pressure as possible, valves SV 13/23 are closed, the system is back-filled with dry nitrogen and the oil in the roughing/backing pumps changed. The latter is usually necessary since the gas ballast on the pumps will not usually handle the amount of water vapor involved. This process is repeated as many times as is necessary until a pressure of ~ 0.05 mbar read on TC 31 can be obtained. The automatic pump down can then be initiated.

The automatic mode will be used throughout a pump down after letting the system up with dry gas.

8.10.1 Interlocks

For safety the following interlocks are incorporated:

- Roughing valves, SV 13/23, can only open if the backing valves SV 12/22 are closed and vice versa and if the pressure registered on gauge TC 31 is more than 0.1 mbar.
- A turbomolecular pumps can only be started if the pressure registered on gauge TC 12/22 is less than 0.1 mbar and the corresponding backing valve SV 12/22 is open.
- The gate valves SV 11/12 can only be opened if the turbomolecular pumps are running at full speed and the system pressure registered on TC 31 is less than 0.1 mbar.

8.11 REFERENCES

Thermal and Vacuum Insulation for Large Cryostats.
D.A. Cragg, April 1994.

8.12 Tables

Table 8.1 - System parameters of the vacuum system.	
Working volume	11 cubic meters
Surface area of multilayer insulation	2262 square meters
Surface area of aluminum alloy	300 square meters
Water vapor fraction in out-gassing load	98%
Working pressure	< 10^{-7} mbar
Cool down commencement pressure	< 10^{-4} mbar (See Ref. 1)
Vessel roughing pressure	< 0.1 mbar
Inner diameter of pumping line	210 mm
Internal diameter of outer pumping line with turbo pumps above the electronics platform.	~400 mm.
Length of pumping line with turbo pumps above the electronics platform.	3.4 m.
Internal diameter of outer pumping line with turbo pumps below the electronics platform.	~360 mm
Length of pumping line with turbo pumps below the electronics platform.	2.0 m
Pump down time to 1 x 10^{-4} mbar	200 hours

Table 8.2 - Parameters of the pumping system.	
Pumping speed of high vacuum pumps	2 X 470 litters/sec
Pumping speed of roughing pump	2 X 18 cubic meters/hr
Roughing time to 0.1 mbar. (Pump and purge 4 times)	5.5 hours
Nominal backing pressure	< 0.01 mbar
Critical backing pressure	~ 0.5 mbar
Maximum steady state turbo inlet pressure for effective use.	~ 0.05 mbar
Diameter of roughing line	40 mm
Diameter of backing line	40 mm
Length of roughing line	~5 meters
Length of backing line	~5 meters

CHAPTER 9

9. DC POWERING AND QUENCH PROTECTION

9.1 CONCEPT

The DC power and quench protection concept is shown in figure 9.1.

The solenoid will be powered from a 7.5kA DC power supply with sufficient voltage capacity to ramp a full field in 30 minutes.

Two modes of discharge are incorporated in the circuit shown in figure 9.1.

1. Slow Dump Discharge Circuit

   Slow discharge - normal ramp down of the current - will be initiated by opening CB1. This will allow the solenoid to discharge through the diode/slow dump resistor combination. This passive type of circuit allows controlled ramp down of the solenoid current under power failure conditions i.e. loss of main power supply. Clearly it is also possible to ramp down the solenoid field by reversing the power supply but this would probably initiate a fast dump (quench) in the event of a mains power failure.

2. Fast Dump (Quench) Discharge Circuit

   Fast discharge will be initiated by opening the linked circuit breaker system CB2 allowing the solenoid to discharge through the fast dump resistor. Fast discharge parameters are given in Table 9.1. The fast discharge concept is based on two main criteria:

   i. A voltage limit of 500 volts across the solenoid during fast discharge. Center-tapping of the fast dump resistor will limit the voltage to ground to 250V. The center-tapped resistor will also allow the measurement of earth leakage currents as a safety and diagnostic tool.

   ii. The protection concept is based on an upper temperature limit of 100K for the cold mass under quench conditions. This limit will give a very good safety margin against peak temperature rise and thermally induced stresses at quench.

Fast discharge of the solenoid will be initiated by the quench detection system or by certain interlocks designed to protect the overall solenoid system.

9.2 QUENCH ANALYSIS

Quench Parameters are given in Table 9.1

Table 9.1 - Quench Parameters

Parameter		Value
Operating Current		6.83 kA
Stored Energy		30 MJ
Inductance		1.0 H
Quench Voltage		500 V
Protection Resistor		0.070 Ohm
Time Constant		15 sec
Adiabatic Peak Temperature		100 K
Overall Current Density	Conductor 1	67 A/mm^2
	Conductor 2	35 A/mm^2
Aluminum Stabilizer RRR Zero Field		500

9.2.1 Adiabatic Hot Spot Analysis

The minimum section of high purity aluminum required for quench protection is given by the adiabatic criterion:

$$J^2 = \frac{G(\theta)\, I\, V}{E}$$

$$G(\theta) = \int_o^{\theta_{max}} \frac{Cp\ (\theta)\, d\theta}{\rho\ (\theta)}$$

Cp (θ) is the specific heat of the conductor

$\rho\ (\theta)$ is the resistivity of the conductor

θ_{max} is the peak temperature

J is the overall current density in the aluminum substrate

I is the operating current

V is the protection voltage

E is the stored energy

Applying the adiabatic criterion to the two conductors proposed for the BABAR solenoid fields.

Conductor I	3.2 x 32mm^2	$G(\theta) \simeq 5 \times 10^{16}$	$\theta_{max} \simeq$ 90 - 100K
Conductor II	5.8 x 32mm^2	$G(\theta) \simeq 1 \times 10^{16}$	$\theta_{max} \simeq$ 30K

for aluminum with a resistivity ratio RRR = 500.

This highly conservative adiabatic approach yields hot spot temperatures which are within the design criteria.

9.2.2 Quench Analysis

A quench analysis of the BABAR solenoid has been made using a code developed for the DELPHI Solenoid design. The code models the thermal and inductive behavior of the solenoid in order to take into account the effects of the quench back and heat transfer to the support cylinder.

The code has been used to model a number of quench scenarios and demonstrate that the BABAR Solenoid is conservative.

9.2.2.1 Quench Code Outline

Normal Zone Propagation

In order to simplify the modeling the normal region growth is assumed to be one dimensional i.e. the initial normal zone is assumed to occupy the full coil circumference - this is a reasonable assumption for such a solenoid.

Thermal Model

It is assumed that the coil is thermally bonded to the force support cylinder.

For any coil element the temperature is described by the heat balance equation:

$$C(\theta) \frac{d\theta}{\partial t} = Gc - Hc$$

where $C(\theta)$ is heat capacity of the coil elements.
Gc is heat generation in the coil
Hc is heat transfer to the shell

For any shell element the temperature variation is described by the heat balance equation:

$$C \frac{d\theta}{\partial t} = Gs - Hs$$

where Gs is inductive heat generated in the shell
Hs is heat transferred to the coil

Cooling of the shell during a quench is neglected.

Electro-magnetic Model

The support shell is represented as a shorted secondary winding with close inductive coupling to the main coil. The dump resistor and shell resistance are assumed to be constant with temperature while for the coil conductor temperature dependence of resistivity is included fully.

Numerical Modeling

The electro-magnetic and thermal models are solved by numerical integration.

9.2.2.2 Quench Studies

Three basic cases have been considered in this initial study.

i. standard protection as shown in figure 9.1 with a circuit breaker delay of 2 seconds.

ii. standard protection with a breaker delay of 5 seconds.

iii. circuit breaker/quench detection failure i.e. no energy extraction.

In all cases the quench is assumed to start at the end of the coil. The coil model is assumed to be made up of the <u>smaller conductor section</u> in all cases.

i. Breaker delay 2 seconds

Figure 9.2 shows the coil and shell temperature in the superconducting region of the coil over the initial transient quench onset, breaker opening. Quench back after ~ 1 second from breaker opening.

Figure 9.3 shows the coil and shell peak temperature over the full discharge. The peak temperature in coil and shell is approximately 45K. Figure 9.4 shows the current decay for the coil and the induced current in the shell.

ii. Breaker delay 5 seconds

Figure 9.5 shows the coil and shell peak temperatures. The peak temperature rise is increased to 60K which is still acceptable.

iii. Breaker failure/quench detection failure

Figure 9.6 shows the coil/shell peak temperature.

The peak temperature in the coil rises to ~210K in this mode. Figure 9.7 shows the current decay in the coil and induced current in the shell. Figure 9.8 shows the quench back of the coil under this mode after 15 seconds.

Quench analyses show that the design of the BABAR solenoid is conservative. Peak temperature rise under design conditions is <50K which gives a large safety margin. The coil is also shown to be safe under failure of the quench detector and circuit breaker operation.

9.3 QUENCH DETECTION

Quench detection can be made by a number of techniques. The multiple bridge circuit shown in figure 9.9 is a typical system with redundancy.

CHAPTER 10

10. INSTRUMENTATION AND CONTROL

10.1 GENERAL

Transducers will be installed in the BABAR Solenoid system for two main functions:

i. Control

ii. Monitoring and Commissioning

The transducers should monitor the following main system parameters:

i. Temperature

ii. Forces

iii. Voltages/Currents

iv. Cryogens - liquid flow

- gas flow
- pressures
- liquid levels

v. Magnetic Fields

vi. Cold Mass/Toroid Position

Vacuum system instrumentation is identified in Section 8.

Transducers for essential functions in inaccessible locations should be duplicated.

10.2 TRANSDUCER FUNCTIONS

10.2.1 Temperature Monitoring

Temperature measurement can be made by two types of transducer dependent on the temperature range:

25K - 300K 1000Ω PRT

4K - 25K 220Ω Carbon Resistors

These transducers will normally be mounted as a single unit.

10.2.1.1 Cold Mass Temperature

The cold mass temperature will be monitored at two points on each coil

- inlet of cooling

- outlet of cooling

This will allow full checking of coil temperatures during cooldown and operation. These transducers will be used for initial commissioning and diagnostics and a selected number ~4 will be logged during normal operation.

10.2.1.2 Radiation Shield Temperature

Temperature transducers will be installed on the radiation shields for commissioning, diagnostics and for control purposes.

The transducer package can be a single 1000Ω PRT.

Approximately 16 transducers will be installed on the radiation shield.

Temperature transducers will be installed as a package with protective covers and heat-sinking of leads.

10.2.2 Force/Loads

Forces/loads will be monitored using strain gauges. These will be applied to all cold mass supports and restraints - axial and lateral.

Force transducers will be used mainly in commissioning although certain transducers will be logged on a long-term basis and used for control purposes through links to interlocks.

10.2.3 Voltages/Currents

Solenoid current will be monitored using a DCCT and will be logged on a continuous basis.

Voltages will be measured on:	current leads coil for protection/detection busbars

10.2.4 Cryogen Flow

Mass flow meters will be included in the cryogen system for monitoring and cooldown control. This will include mass flow rates for gaseous helium and pumped liquid helium.

Pressures - cryogenic systems will include pressure measurement for setting up and monitoring.

10.2.5 Magnetic Fields

The solenoid will be equipped with hall probes to measure and monitor fields within the structure.

10.2.6 Position Sensors

Position sensors may be included to accurately monitor the position of the cold mass and the solenoid.

10.3 TRANSDUCER LISTING

Table 10.1 gives an outline listing of the transducers, function and location.

10.4 MONITORING AND CONTROL

The control and monitoring system should be set up to allow adequate diagnostics at subsystem level for commissioning purposes.

With this principle in mind a degree of subsystem control and monitoring capability should be installed e.g. for vacuum, electrical and cryogenic systems. The control system must have enough memory to store at least one month of operation. Periodic backup will be required.

Fundamental control systems for protection interlocks should be hard wired.

The solenoid control system will be required to interface with the overall control system for the experiment.

Figure 10.1 shows a concept for solenoid monitoring and control. This concept is based on the system implemented for the H1 Solenoid at DESY and shows the principle outline above.

The details of the control structure, interface protocols and monitoring/data logging at the various levels will need to be agreed between the manufacturer and SLAC.

CHAPTER 11

11. INTERFACES

11.1 SPACE

The limiting volume occupied by the coil is that shown in figure 11.1, plus that occupied by the external supports and the chimney. Space is very tight in the vicinity of the four main support brackets, which may need redesigning once the earthquake criteria are more accurately defined. If large flanges are used on the inner detector, as we have assumed (figure 11.1), then it is not clear how the cabling from the inner detector will be routed out.

11.2 STRUCTURAL

The design of the external supports is linked to that of the iron yoke. For example, it would appear that the 1" thick plates in the regions where the main support brackets attach (see figure 6.1) are probably not sufficiently strong or stiff to support the earthquake loads. For this reason we have not used a lot of effort to analyze the design of the main brackets.

A suggested way forward would be to define the solenoid interface such that the solenoid includes the plates fastened to the end flange, but the rest of the support brackets are left as the responsibility of the BABAR team.

The interface between the inner detector end plates and the solenoid end flanges needs to be defined. We have looked at the way the axial load is passed into the solenoid (figure 5.3). Our calculations (section 5, table 5.2, cases 4a and 4b) suggest that the axial load may be put onto the vessel anywhere on the end flanges, provided it is reasonably well spread around the circumference.

We recommend that the design of the end plates for the inner detectors (figure 11.2) be progressed, as our simple calculations suggested there may be high stresses present.

11.3 ELECTRICAL/CONTROL

The detailed electrical and control interfaces remain to be defined. At this stage the interface between the solenoid and the ancillary equipment is at the top of the turret dewar.

The solenoid control system should incorporate the necessary interfaces to link the solenoid to the cryogenic electrical and vacuum systems.

11.4 CRYOGENICS

The physical interface of the solenoid to the BABAR cryogenics plant is defined at the top plate of the turret dewar for the purposes of their design study. In practice the interface will need to be carefully defined to take account of all operational and fault conditions - quench, power failure, refrigeration failure etc.

11.5 ASSEMBLY REQUIREMENTS

The solenoid will be delivered with the chimney and turret dewar detached. Assembly of the solenoid into the iron flux return and is alignment and mounting will require special handling jigs and tools.

Adequate clearances must be defined for assembly and installation at the interfaces to the iron and inner detectors.

Installation of the chimney and turret must be part of the experimental assembly planning.

11.6 INTEGRATED EARTHQUAKE DESIGN

We have used a very simple approach to earthquake loading. Consideration should be given to a more rigorous frequency-response-based analysis when the details of the rest of the experiment are clearer.

We have not looked in detail at the overall earthquake design philosophy as regards tolerable damage levels for various parts etc. Whilst we believe we have done enough to demonstrate that a feasible design exists, some more work is probably needed in this area to fully define the specification.

11.7 GENERAL

The interfaces are many and varied, we have only mentioned a few here. When the contract is placed for manufacture of the coil, an essential part of that contract will be a complete specification of the interfaces.

CHAPTER 12

12 ASSEMBLY, TRANSPORT AND INSTALLATION

12.1 ASSEMBLY OF THE COIL AND TESTS

The coil will be assembled inside the cryostat at the manufacturers and the electrical and hydraulic connections made at the chimney, so that the coil can be tested before shipping.

A complete cooldown will be carried out from room temperature to the operating temperature of 4.5K. The cooldown will allow checking of cooldown time, temperature control, heat loads and full operation of sensors.

A magnetic test will be performed at low field (30% of the operating current) to check superconductor operation, the joint resistance and the additional losses due to the energization.

12.2 TRANSPORT

Before delivering the magnet, after the tests at the factory, the end flanges could be dismounted to allow a hard connection of the cold mass to the cryostat walls. Depending on the transport facilities, the chimney could require to be dismounted too. In this case the electrical and hydraulic connection must be disconnected and protected against breakage.

APPENDIX A

A. EARTHQUAKE LOADS ON BABAR SOLENOID

A.1 DESIGN APPROACH

In designing against earthquakes, one can use either a full dynamic calculation of the effects of the loads, or else a simple static equivalent approximation.

The full dynamic calculation is beyond the scope of this study; we have used the simple static equivalent approximation.

A.2 MAGNITUDE OF EARTHQUAKE ACCELERATIONS

The magnitude of the accelerations used in the static equivalent calculation depends upon a variety of factors, not all of which are as yet fully specified.

We used accelerations of 1.2g in the horizontal directions (radial and axial), and 2g in the vertical direction. These figures assume that anti-vibration mounts will be used to absorb the horizontal motions, which would otherwise be bigger than the vertical ones.

Direction	Acceleration
Radial	1.2g
Axial	1.2g
Vertical	2g (+ weight)

A.3 APPLICATION OF FORCES

We have considered the earthquake loads to act in only one of the three directions at a time. In practice the real loads will be a vector whose direction and magnitude changes and whose limits we assume are given by the values above.

A.4 SURVIVAL

We will use a survival design stress of half the ultimate stress.

A.5 DAMAGE

We cannot guarantee there will be no damage - especially if one considers the effects of, for example, some or all of the central detector parts falling onto the inside of the vacuum vessel. As an indication, a timescale of a year to have the experiment working again after a major earthquake seems realistic.

We should attempt to ensure that only those parts which could reasonably be repaired or replaced are damaged. The only major component of the coil that does not fall into that category is the cold mass - because it is bonded together making repair difficult and because the lead times on supply of the cable are likely to be long.

APPENDIX B

B. MATERIAL PROPERTIES

TABLE B.1 - Mechanical and physical properties of some aluminum alloys

Material	Temp K	Yield MPa	Tensile MPa	Elong. %	Weld-ability	Young GPa	Therm. Conduct. $Wm^{-1}K^{-1}$	Electr. Resist. μW cm
5083	295	235	335	15	Excell.	71.5	120	5.66
	77	274	455	31.5		80.2	55	3.32
	4	279	591	29		80.9	3.3	3.03
6061	295	291	309	16.5	Good	70.1		3.94
	77	337	402	23		77.2		1.66
	4	379	483	25.5		77.7		1.38
2219	295	371	466	11	readily	77.4	120	5.7
	77	440	568	14		85.1	56	
	4	484	659	15		85.7	3	2.9

TABLE B.2 - Mechanical and thermal properties of some materials used in the solenoid design

Material	Temp K	Yield MPa	Tensile MPa	Young GPa	Thermal contraction $\Delta L/L$ 300 ->4.2 k	Therm. Conduct. $Wm^{-1}K^{-1}$
Al 99.998	295	13	45	71	0.00415	240
	4	14	50	78		4000
Fiberglass epoxy I	295	245		12	0.006	
	4			18		
Fiberglass epoxy //	295	280		27	0.0021	
	4			33		
Ti (6Al4V)	295	890		109	0.00152	7.5
	4	1725		129		1.3
Cu/NbTi 1.8/1	295		530	110	0.00231	
	4	360	800	130		

TABLE B.3 - RADIATION LENGTH

Component		Thickness (mm)	N I L
Cryostat Al inner wall		10	0.0255
Cryostat Al outer wall		30	0.0765
Al shields		20	0.051
Al stabilizer		32	0.0816
NbTi/Cu	min	-	-
	max	9	0.06
Al cylinder	min	30	0.0765
	max	70	0.1785
Insulation		2	-
Total	min		0.31
	max		0.47

APPENDIX C

C. BABAR Earthquake Load Cases: The Cold Mass

This information is found in Report number RAL/ASD/CME/Misc/9 with selected diagrams.

C.1 THE MODEL

C.1.1 Finite element model

The Finite Element program ANSYS was used for this analysis. This was a model of the coil and the supporting cylinder with the windings modeled as if they were aluminum. The geometry was meshed into four-noded quadrilateral elements.

Constraints were applied on the four points at each end at ± 45° to the horizontal. These points were constrained tangentially to the coil. In addition six points at 60° intervals on the +Z end were constrained axially. The constraints are shown in the first plot.

C.1.2 Dimensions

Dimensions used were the cold dimensions as of 9 February 1995. These were:

Coil	
Length	3.455m
Radius	1.4835m
Thickness	0.03m
Support cylinder	
Overall length	3.515m
Flange	0.03m thick x 0.1032m radially
Thickened Length	0.375m at each end
Thickness of thickened length	0.0732m
Thickness of standard length	0.033m
Material Properties	
Modulus of elasticity	70GPa
Poisson's ratio	0.3
Density	2970Kg/m^3

C.2 LOAD CASES

Six load cases were considered. In each case loads of 1g gravity, radial magnetic forces and an axial force of 10 tonnes due to magnetic errors were applied. The 10 tonnes load was distributed equally over all nodes. In order to apply the magnetic forces the structure was divided axially into cylindrical bands each consisting of four areas. The radial magnetic forces were a function of axial position and were integrated to give a pressure which was constant within each band but varied from band to band as shown in the table below. The width of each band is also shown in the table.

band 1	0.405m	1.73MPa
band 2	0.24425m	1.52MPa
band 3	0.24425m	1.4MPa
band 4	0.288m	0.859MPa
band 5	0.288m	0.7515MPa
band 6	0.288m	0.73MPa
band 7	0.288m	0.73MPa
band 8	0.288m	0.7515MPa
band 9	0.288m	0.859MPa
band 10	0.24425m	1.4MPa
band 11	0.24425m	1.52MPa
band 12	0.405m	1.73MPa

The other loads varied from case to case and are detailed below with the results.

C.3 RESULTS

Maximum displacements and Von Mises stresses are given for each case. The plots show the displacements and Von Mises stresses. They show the Von Mises stress at whichever surface of the element gives the largest stress. The largest stresses and displacements are due to the radial magnetic pressure. This is at a maximum near the ends of the coil, but because the ends of the support cylinder are thickened the maximum stress and displacements occur near this thickened region thus producing the patterns of the form shown in the plots.

i. Loadcase c1
This case had additional axial loads of 20 tonnes plus 1.2g.
Maximum displacement 0.86mm
Maximum Von Mises stress: 38.7MPa

ii. Loadcase c2
This case had additional sideways loads of 20 tonnes plus 1.2g.
Maximum displacement : 1.04mm
Maximum Von Mises stress : 39.6MPa

iii. Loadcase c3
This case had additional vertical loads of 20 tonnes plus 2g.
Maximum displacement : 1.13mm
Maximum Von Mises stress : 39.9MPa

iv. Loadcase c4
This case had additional vertical loads of 20 tonnes:
Maximum displacement : 1.02mm
Maximum Von Mises stress : 39.5MPa

v. Loadcase c5
This case had additional vertical loads of 2g plus 20 tonnes at 45° to the vertical:
Maximum displacement : 1.13mm
Maximum Von Mises stress : 39.9MPa

vi. Loadcase c6

This was case c4 with axial magnetic forces. These were a function of axial position and were applied by distributing the summed force over each band over the ring of 48 nodes at the center of the band. The force on each node in was as follows, starting at the -Z end.

band 1	$1.153 \; 10^{5}$ N
band 2	$1.635 \; 10^{4}$ N
band 3	$-2.177 \; 10^{4}$ N
band 4	$-2.35 \; 10^{4}$ N
band 5	$-7.562 \; 10^{3}$ N
band 6	$-2.861 \; 10^{3}$ N
band 7	$2.861 \; 10^{3}$ N
band 8	$7.562 \; 10^{3}$ N
band 9	$2.35 \; 10^{4}$ N
band 10	$2.177 \; 10^{4}$ N
band 11	$-1.635 \; 10^{4}$ N
band 12	$-1.153 \; 10^{5}$ N

Maximum displacement : 1.19mm
Maximum Von Mises stress : 47.5MPa

C.3.1 Files

The files are stored in directories named /home/elnath/john/ansys/babar3/casei where i is the loadcase number.

APPENDIX D

D. EARTHQUAKE LOAD CASES; THE VACUUM VESSEL

This information is found in Report number RAL/ASD/CME/Misc/10 with selected diagrams.

D.1 THE MODEL

The Finite Element program ANSYS was used for this analysis. This was a model of the vacuum vessel . Dimensions were taken from the attached figure with dimensions to mid-surface being used. The geometry was meshed into four-noded quadrilateral elements. Vertical constraints were applied on the two points at each end on the horizontal center line at the middle of the end flange and horizontal constraints were applied at one point at each end at the same place. In addition four points at ±30° to the vertical at the outer edge of the end flange on the +Z end were constrained axially. The constraints are shown in the first plot.

Material Properties	
Modulus of elasticity	70 GPa
Poisson's ratio	0.3
Density	2970 Kg/m^3

D.1.1 Load Cases

Six load cases were considered. In each case loads of 1g gravity and the vacuum load were applied. Lumped masses were applied at the 45° positions at each end to simulate the mass of the cold mass and shield. The total lumped mass was 7600 kg. The other loads varied from case to case and are detailed below with the results:

D.1.2 Results

Maximum displacements and Von Mises stresses are given for each case. The plots show the displacements and Von Mises stresses. The Von Mises results are taken at whichever surface gives the largest stress. The largest stresses occur near the constraints.

i. Loadcase v1
This was the nominal case with no additional loads.
Maximum displacement : 0.4mm
Maximum Von Mises stress : 23.9MPa
The general level of Von Mises stress was less than 16MPa

ii. Loadcase v2
This case had additional vertical gravity load of 2g.
Maximum displacement : 0.5mm
Maximum Von Mises stress : 32.2MPa
The general level of Von Mises stress was less than 18MPa.

iii. Loadcase v3a
This case had additional sideways load of 1.2g in the +X direction.
Maximum displacement : 1.47mm
Maximum Von Mises stress : 40.6 MPa
The general level of Von Mises stress was less than 23MPa.

iv. Loadcase v3b
This case had additional sideways load of 1.2g in the -X direction.
Maximum displacement : 1.72mm
Maximum Von Mises stress : 58.6MPa
The general level of Von Mises stress was less than 20MPa.

v. Loadcase v4a
This case had an additional load of 60 tonnes distributed at the middle of the -Z end flange. The lumped masses were replaced by forces at six positions on the -Z end to simulate the axial load from the cold mass plus shield and forces were applied to simulate the weight of the cold mass plus shield at the eight positions corresponding to the support positions of the cold mass. It will be noticed that the displacements are asymmetrical with regard to left and right. This is seen most clearly in the plot of axial (Z) displacement. The reason for this can be seen in the next plot which shows a cross-section through the displaced and undisplaced structure. The constraints in X prevent the -X side of the structure from bending as much as the +X side.

Maximum displacement : 1.05mm
Maximum Von Mises stress : 46.1MPa
The general level of Von Mises stress was less than 16MPa

vi. Loadcase v4b
As case v4a except that the 60 tonnes was distributed at the inner edge of the end flange.

Maximum displacement : 1.43mm
Maximum Von Mises stress : 42.1MPa
The general level of Von Mises stress was less than 25MPa

D.1.3 Files

The files are stored in directories named /home/elnath/john/ansys/babar3/casei where i is the loadcase number.

APPENDIX E

E. Manual Check of Stresses in External Supports

APPENDIX F

F. BABAR Cold Mass - detailed analysis of the structure

This is reproduced from the RAL report number RAL/ASD/CME/MISC/012.

F.1 Introduction

A Finite Element analysis was made of the BABAR cold mass in order to determine in detail the stress distribution in the coil.

F.2 The Model

In order to calculate the stress distribution in the coil, particularly in the insulation , a very detailed FE model is required. It would not be feasible to model the entire structure in such detail so two models were made, one modeling each turn of the coil individually but not modeling separately the insulation or the conductors (the "coarse" model), and the other a submodel of the region of interest where both insulation and conductors were modeled in sufficient detail to allow accurate stress calculations to be made (the "fine" submodel). Because the structure has rotational symmetry about the axis and mirror symmetry about mid-axis an axisymmetric model of half the structure was made.

F.3 The Calculations

Appropriate forces and constraints were applied to the coarse model. The forces were the magnetic forces interpolated from the table shown below and applied to the center point of each coil and the constraints were those to provide the symmetry condition. This model was then run to calculate stresses and displacements. In particular the displacements of the coarse model at the positions corresponding to the boundaries of the fine submodel were calculated. The fine submodel was then run using the displacements from the coarse model as boundary conditions and with the same magnetic forces applied to the conductors in the submodel, this time the forces were distributed uniformly over the conductors.

F.4 Material Properties

For the fine model the relevant material properties were as follows:

Matrix	E = 70 GPa
	ν = 0.3
Conductor	E = 130 GPa
	ν = 0.3
Insulation	E// = 33 GPa
	E⊥ = 18 GPa

	ν = 0.3
Coil support	E = 80.9 GPa
	ν = 0.3

For the coarse model the coil was given weighted average values of the matrix, conductor and insulator properties. These differed between the regions with wide and narrow turns, because the proportions of the materials differed:

Coil (narrow turns)	E (radial) = 76.02 GPa
	E (axial) = 75.72 GPa
	E(circumferential) = 76.65 GPa
	ν = 0.3
Coil (wide turns)	E (radial) = 76.41 GPa
	E (axial) = 75.43 GPa
	E(circumferential) = 77.04 GPa
	ν = 0.3
Coil support	E = 80.9 GPa
	ν = 0.3

F.5 Results

The overall Von Mises stress for the coarse model in shown in the first plot. Two regions were examined in detail using a fine model, the transition region between the wide and narrow turns, and the region near one end. Plots two and three show Von Mises and shear stress respectively in the transition region The maximum Von Mises stresses (up to 58.8 MPa) are in the conductors and the lowest (13 to 18 MPa) are in the insulation. This is as expected, the stresses are approximately in proportion to the respective moduli of elasticity. The fourth plot shows detail of the shear stress in the insulation.

The results for the region near the end are shown in plots five and six. Here Von Mises stresses range up to 20 MPa in the conductors and about 5MPa in the insulation. Plot seven shows the shear stress in the insulation.

F.6 Magnetic Forces

Axial Position (m)	Radial Force (N)	Axial Force (N)
0.038798	4.8415e+05	2233.3
0.10259	4.8443e+05	7396.3
0.16637	4.8501e+05	12666
0.23016	4.8589e+05	18122
0.29395	4.8710e+05	23859
0.35774	4.8863e+05	29989
0.42153	4.9051e+05	36666
0.48532	4.9274e+05	44105
0.54911	4.9532e+05	52636
0.61289	4.9824e+05	62781
0.67668	5.0150e+05	75512
0.74047	5.0509e+05	92409
0.80426	5.0894e+05	1.1819e+05
0.86805	5.1389e+05	1.6088e+05
0.93183	5.1803e+05	2.8105e+05
0.98320	7.3067e+05	4.0092e+05
1.0346	7.3319e+05	1.9313e+05
1.0859	7.3563e+05	94438
1.1373	7.3466e+05	18269
1.1887	7.3171e+05	-46295
1.2401	7.2659e+05	-1.0666e+05
1.2914	7.1924e+05	-1.6581e+05
1.3428	7.0952e+05	-2.2619e+05
1.3942	6.9723e+05	-2.8998e+05
1.4455	6.8209e+05	-3.5971e+05
1.4969	6.6380e+05	-4.3924e+05
1.5483	6.4193e+05	-5.3405e+05
1.5996	6.1645e+05	-6.6068e+05
1.6510	5.8361e+05	-8.4303e+05
1.7024	5.5166e+05	-1.2900e+06

The files are to be found in /home/elnath/john/ansys/babar4

The transition region fine model is in sub-directory cased1 and the end region fine model in sub-directory cased3.

APPENDIX G

G. ADDENDUM 2: CHANGES TO SUPPORTS, ETC.

Since the main body of this study was written, there have been some changes to the design requirements. This addendum describes the effect of certain changes on the stresses in the vacuum vessel, on the design of the support system, and on the interfaces.

G.1. The changes to the requirements

The earthquake load criteria have changed as a results of the decision to use compliant mounts for the experiment. The design loads are now 0.2g lateral (radial or axial) and 1.4g vertical. (This compares with 1.2g and 2.0g respectively in the previous design).

A knock-on effect of this change is that the axial earthquake loads on the inner detectors no longer need to be supported through the vacuum vessel as described in section 6.4 above.

Another change is to the geometry of the iron yoke; this will now be eight-sided, not six-sided. This affects the positioning of the external supports which can no longer be on the horizontal center line but must occupy spaces 22.5 degrees below it. The supports have been moved to allow for this, and also so that instead of supporting the end-flange of the vacuum vessel they are now located on the outer wall of the vessel. This is shown in the new interface diagram, figures 2a and 2b to this addendum.

The size of the coil has been changed to allow more space inside; all the radii have been increased by 30mm.

G.2. Changes to the overall stresses in the cold mass

The effect of the changes in size will be small. The principal source of stress in the cold mass is the magnetic load, which has not changed. For these reasons, we did not re-run the stress analysis on the overall stress state in the cold mass.

G.3. Changes to the stresses and deflections in the vacuum vessel

(This compares to section 5.2 above)

We have re-run the finite element model of the vacuum vessel with the new dimensions, new support positions, and new loads. There were just four external support positions, placed 22.5 degrees below the horizontal center line on the outer wall of the vessel, in the thickened portion near the end. All four supports were constrained vertically, two (at one end) axially, and two (on one side) horizontally.

Loads from the cold mass were transferred as before, using new values for the mass of the cold mass.

Three load cases were considered. In all cases, the weight of the vessel and the cold mass was included.

Table G.1					
Case	Description	Additional Loads	Max deflection (mm)	Max general stress (MPa)	Max local stress (MPa)
1	Axial	30t + 0.2g on cold mass, 0.2g axial on vessel.	0.71	17	38.1
2	Radial	20t +0.2g on cold mass, 0.2g radial on vessel	2.34	16	70.8
3	Vertical	20t + 1.4g on cold mass, 1.4g on vessel.	0.68	17	37.7

The work is fully written up in report number RAL/ASD/CME/Misc/015.

G.4. Changes to the loads on the supports

This compares with section 6.2.2

The masses are now vessel mass = 5.4t, cold mass = 7.3t, radiation shields etc = 1.5t. Thus cryostat = 6.9t.

The earthquake loads are now 0.2g sideways and 1.4g vertical.

The new table of loads is

Table G.2 (compare with table 6.1 of the design study)			
	On cold mass (7.3 tonnes)	On cryostat (6.9 tonnes)	Combined load on supports
Max. radial loads:			
g forces (0.2g)	1.5	1.4	
magnetic alignment errors	20		
Total	21.5	1.4	22.9t
Max. axial loads:			
g forces (0.2g)	1.5	1.4	
magnetic loads due to known geometry	10		
magnetic alignment errors	20		
Total	31.5	1.4	32.9t
Max. vertical loads:			
weight	7.3	6.9	
g forces (1.4g)	10.2	9.7	
magnetic alignment errors	20		
Total	37.5	16.6	54.1t

The loads on the rods (compare with section 6.3, and see figure 1 to this addendum) are now:

Table G.3

		Axial	Radial shared between 4 rods
Nominal	Tensile	10t	20 + .707*7.3 = 25.2
	Compressive	30t	
Earthquake	Tensile	20 + 0.2*7.3 = 21.5t	35.4t
	Compressive	30 + 0.2*7.3 = 31.5t	25t

And the design is now:

Table G.4 (compare with table 6.2 of the design study)

	Units	Axial	Radial
Loads			
Nominal load - tensile	tonne	10	25.2
Nominal load - compressive	tonne	30	
Rods to resist nominal load		6	4
Quake load Tension	tonne	21.5	35.4
Compression	tonne	-31.5	-25.0
Rods to resist quake load		6	4
Material			
Material	Titanium alloy 6%Al, 4%V		
Ultimate stress	MPa	1000	
Yield stress	MPa	900	
Conductivity integral 80K to 4K	W/m	213	
Rod sizes			
Rod diameter - nominal. This is the diameter of the rod over all of its length except the ends, where it is turned down to M20.	mm	25	25
Rod length	mm	350	300
Rod diameter in thread root	mm	16.9 (M20)	16.9 (M20)
Stress, buckling			
Stress under Earthquake load in thread root			
Tension Compression	MPa	160 -234	395 -278
Factor of safety on ultimate stress under earthquake load		4.2 (compressive)	2.5 (tensile)
Factor of safety on buckling (using nominal diameter)		2.9	3.4
Thermal conductivity			
Rods in conductivity calculation		6	8
Total heat load over half the length of the rods	Watts	3.6	5.6

The changes to the loads are not very great, and serve to increase the safety margins in all cases. It might be possible to reduce the size or number of the rods, but it was not considered worthwhile to pursue that option here.

G.5. Changes to the design of the external supports

The major change results from the fact that the coil no longer has to transfer the loads from the inner detectors. We have done away with the subsidiary axial support plates, and simplified the design of the main support brackets. Conceptually, the design is now as shown in the interface diagram.

G.6. Changes to the interfaces

The interface diagram (figure 11.1 of the design study) has been modified to reflect the new dimensions and support design, see figure 2 of this addendum. The design of the inner detector no longer needs to allow for the support brackets as was shown in figure 11.2. The supports no longer occupy part of the axial space between the end flange of the vacuum vessel and the end-cap of the yoke.