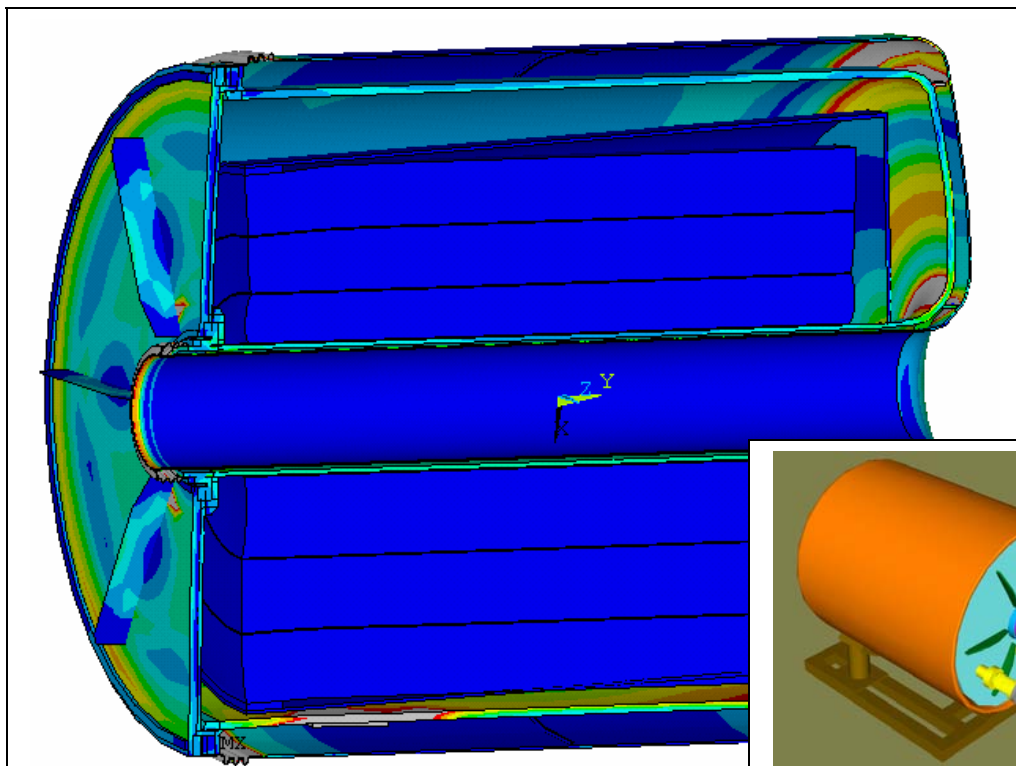


**Design Report
MERIT (MERCURY Intense Target)
BNL - E951 15T Pulsed Magnet for Mercury
Target Development
Neutrino Factory and Muon Collider
Collaboration
March 2007 (Shipped with Magnet)**

**P.H. Titus
MIT Plasma Science and Fusion Center**



**MERIT Pulsed Magnet –Structural Model
Inertially Cooled , 80K LN2 Cooled Between Shots**

Preliminary Design of BNL Pulsed Magnet –Inertially Cooled , 30K
He Gas Cooled Between Shots



Table of Contents

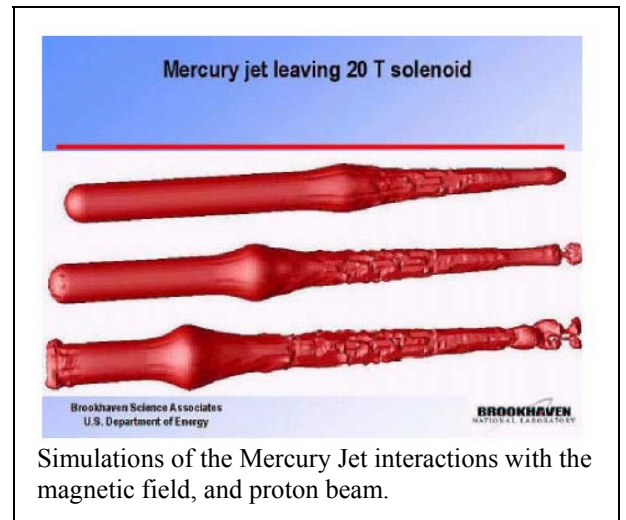
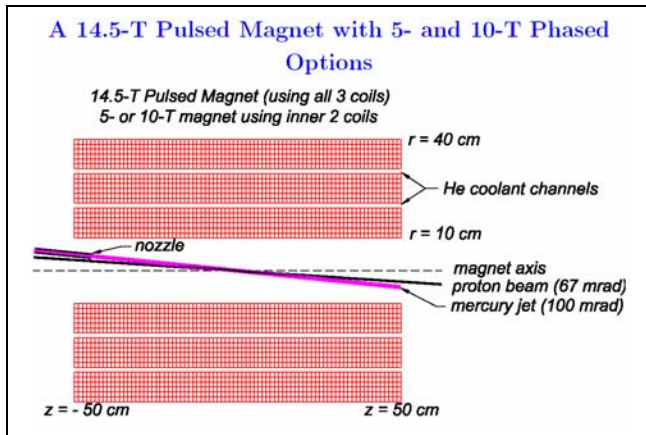
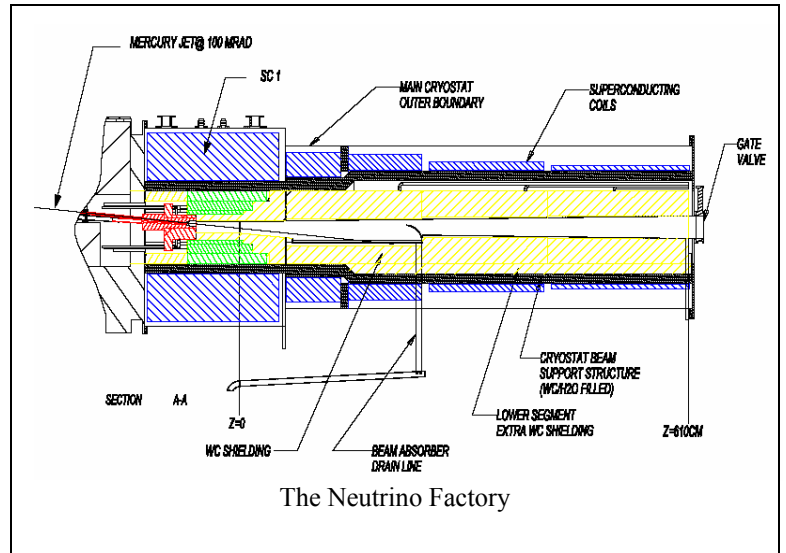
Introduction	1.0
Design Input/Specifications	2.0
Circuit Simulations	2.2
LN2 82K Baseline	2.2.1
Conductor non-conformance simulation	
References	3.0
Structural Design Criteria	4.0
Conductor and Insulation Systems Design	5.0
Electrical Simulations, Field and Iron Material Force Calculations	6.0
Coil Lorentz Force Stress Analysis	7.0
Nominal – Three Segment Analysis	
Optional – Two Segment Analysis	
Elastic Plastic Analysis of Inner Segment Mandrel	
Coil Heat-Up and Thermal Stress	8.0
Break-outs, Leads and Penetrations	9.0
Cryogenic Electrical Break	9.1
He/Vacuum/LN2 can pressure design.	10.0
Pressure Test	10.1
Simplified Stress Calculations	10.2
Vessel Simulation Model	10.3
Vessel Simulation Model Results	10.4
Cryostat Bolting Calculations	10.4.5
Cooldown Displacements	10.5
Vacuum Jacket	11.0
Vacuum Boundary Support	11.1
G-10 Support Rings	11.2
Vacuum Jacket Linear Elastic Buckling (Eigenvalue Method)	11.3
Vacuum Jacket Elastic Buckling (Large Displacement Method)	11.4
Bellows Design	12.0
Eddy Currents, Loads and Stresses in Vacuum and He Can	13.0
Steady State Heat Gain.	14.0
Cooldown between shots	15.0
Temperature profile, and time-temperature calculations	
Pressure drop calculations	
Instrumentation	16.0

Support Frame	17.0
Support Frame	17.1
Gravity Supports	17.2
Lift Fixtures	17.3
Manufacturing/Assembly Approaches	18.0
Winding Procedures	
Phased Construction requirements	
Safety, and Failure Analyses	19.0
Cost Studies	20.0
Operational Experience	21.0

1.0 Introduction

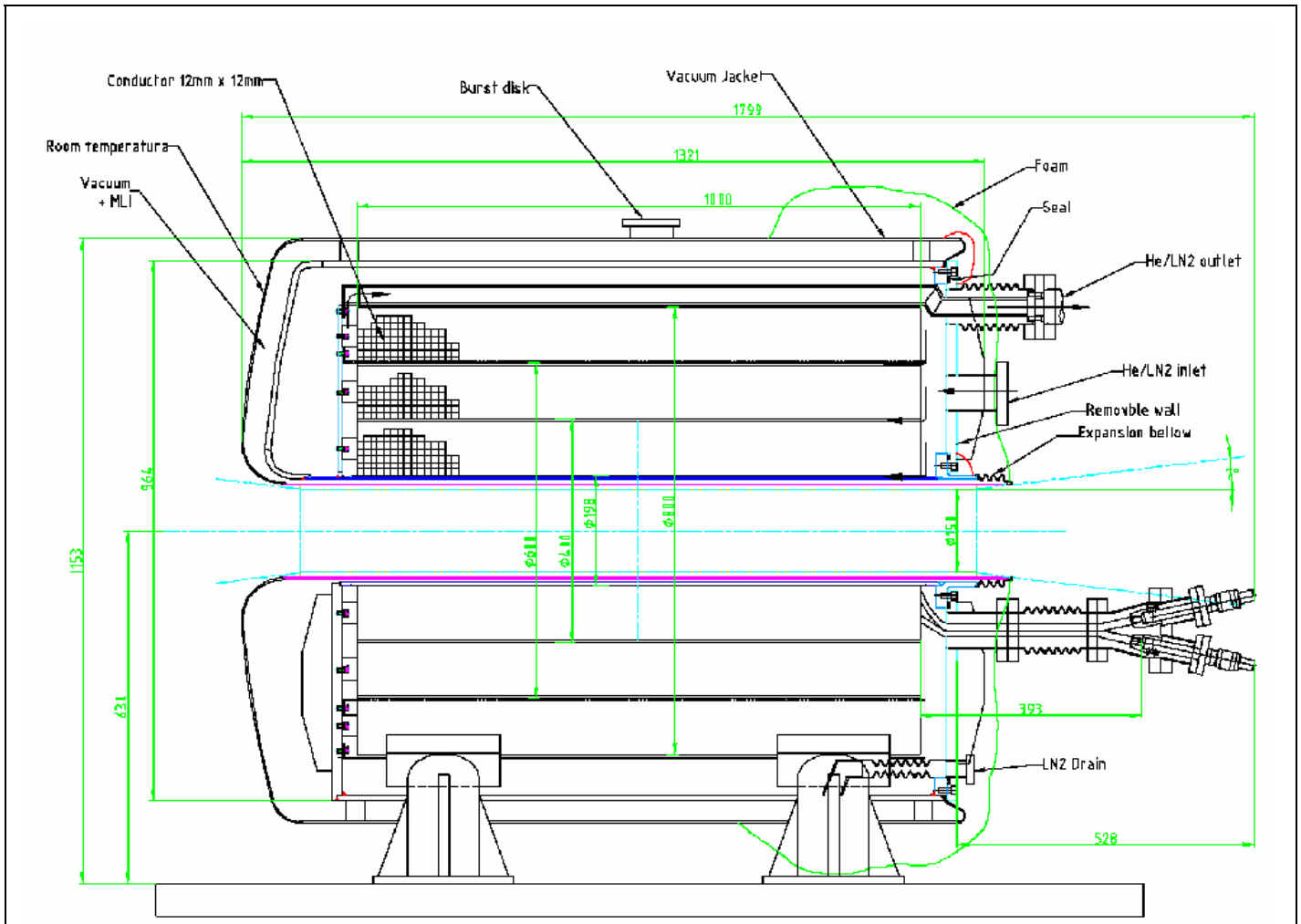
The purpose of the experiment is to study mercury targets for neutrino beams and a muon collider source. In these experiments, a mercury jet intersects a proton beam in a high field. The particles that are produced are confined by the magnetic field. The behavior of the mercury jet in this environment needs to be understood before resources are committed to the larger experiments.

Cost issues dictate a modest coil design. Power supply limitations dictate a compact, low inductance, high packing fraction design. A three segment, layer wound solenoid is proposed for the pulsed magnet. Phased manufacture is supported. The third segment may be purchased and installed in the cryostat later. The conductor is half inch square, cold worked OFHC copper. The coil is inertially cooled with options for liquid nitrogen or gaseous Helium cooling between shots. Coolant flows through axial channels in the coil. For the same packing fraction, a hollow conductor would have a 1.4mm diameter hole. This is judged difficult to manufacture and difficult to maintain helium flow over the conductor length, without intermediate gas connections. The coil will be epoxy impregnated. Wound coils of this small radius, using cold worked conductor, retain internal elastic stresses from the winding process, and if not impregnated, require elaborate clamping mechanisms to have the coil retain its shape.



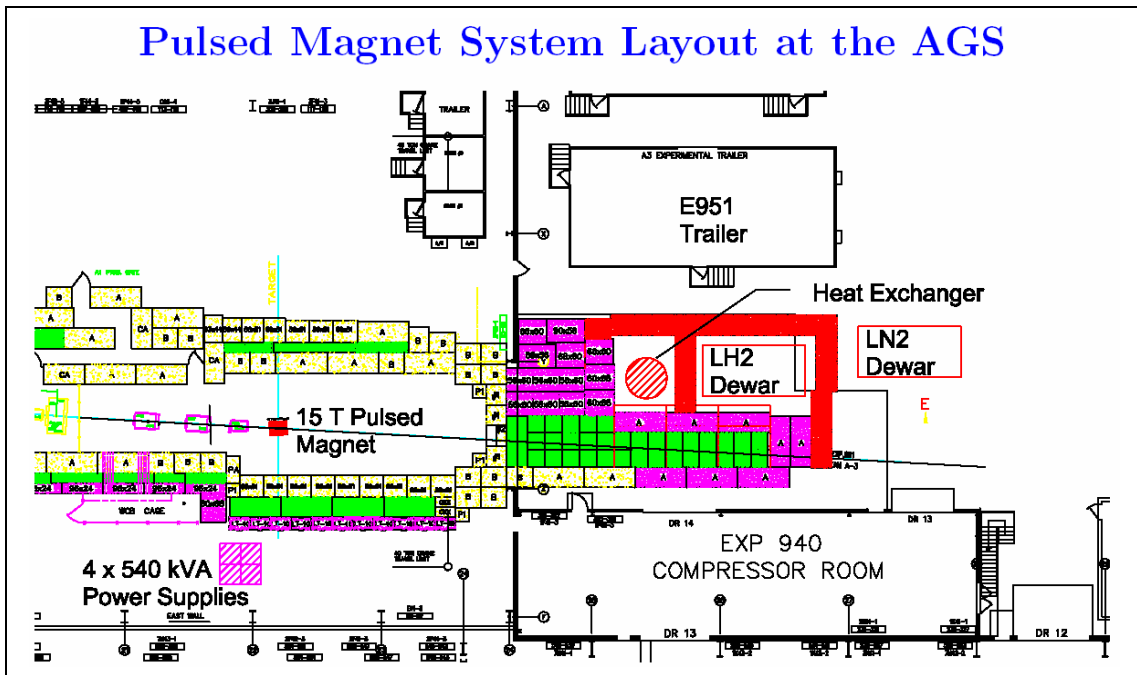
Proposed Operational Scenarios:

Case #	Peak Field	Coolant	T after pulse	T coolant	Start Bulk Temp
1	5T	Helium Gas	90K	66K	84K
1a	5T	LN2	90K	66K	84K
2	10T	Helium Gas	96K	66K	74K
2a	10T	LN2	96K	66K	74K
3	14.5T	Helium Gas	78K	22K	30K

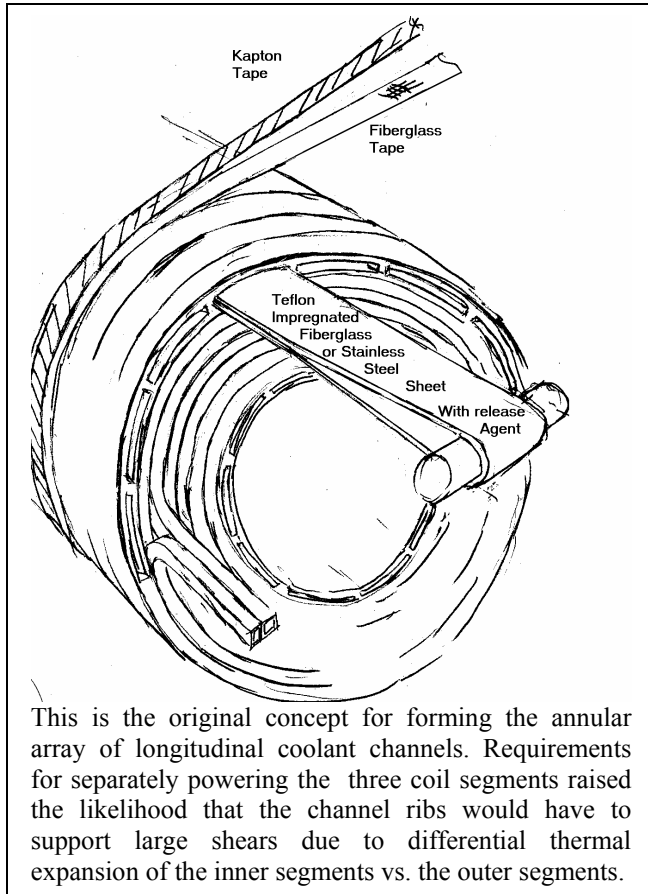


Elevation/Assembly Drawing of the BNL - E951 15T Pulsed Magnet, used for mercury target development for the Neutrino Factory and Muon Collider Collaboration

Pulsed Magnet System Layout at the AGS



Dual operational modes require special design of the cryostat/helium can. This is discussed in the section on cooldown behavior.



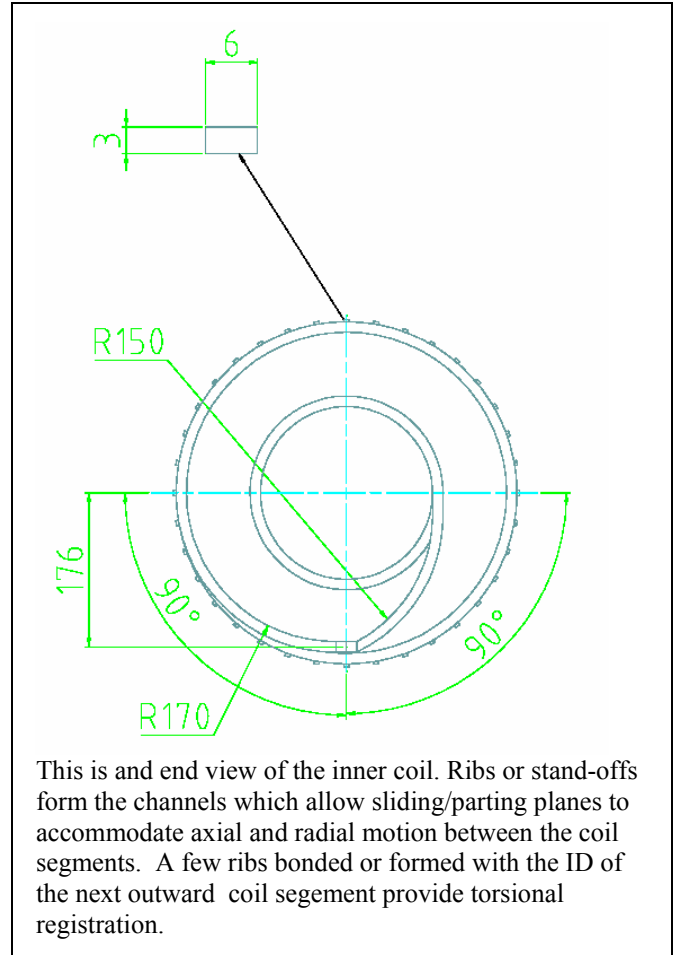
Insulation design impacts the conduction cooling behavior:

- Kapton is the limiting element in the thermal conduction through the coil.
- Kapton initially was expected to be wound around the conductor. This produced the equivalent of 5 mils of Kapton between layers.
- To improve conduction, Kapton is used only between the layers. Turn to turn voltage is lower than layer to layer. The turn to turn voltage is less than the rule of thumb for He breakdown voltage (1 volt/mil at 1 atmosphere) for the insulation thickness proposed. Note that the He operating pressure is expected to be 15 atmospheres, the pressures inside the epoxy winding pack may be substantially lower, depending on Helium diffusion, making the 1 atm breakdown voltage for the conductor, reasonable.
- The layer to layer voltage exceeds this rule of thumb, however, and would need the Kapton if there was an imperfection in the epoxy/glass insulation. Half laps of kapton and fiberglass, similar to the CS model coil will retain some structural integrity.

- Once a layer of conductor is wound, a layer of Kapton/glass would be wound on the completed layer of conductor. This produces the equivalent of 3 mils of Kapton rather than 5 if the conductor were wrapped individually. Every 6 to 8th layer some sort of preformed channel array would be layed on, then wrapped with glass/Kapton to hold it in place, and as the layer insulation for the next layer of conductors. .

Thermal contraction/shock of channel presents some reliability concerns. It was originally conceived that the channels might be preformed and the layer winding would be applied over them. Thermal differentials could produce separation and loss of conduction. In the final design each of the three coil segments is manufactures separately, and slipped over the next inner coil.

Experience with Alcator C-Mod indicates that for final magnet temperatures at or below 100K, the channels will not need “turbulators” or surface trips to break-up the film boiling layer. This will have to be reviewed during the design phase. If some form of surface roughness will be required in the 2mm channels, then the method of forming the channels with removable strips will have to be re-visited.



This is an end view of the inner coil. Ribs or stand-offs form the channels which allow sliding/parting planes to accommodate axial and radial motion between the coil segments. A few ribs bonded or formed with the ID of the next outward coil segment provide torsional registration.

2.0 Design Input/Specifications

2.1

It is important that we have the expected (design) operating conditions for, especially, the following parameters:

Mode	Field	T ₀	Energy	T ₂	ΔT	Cycle
Units >	T	K	MJ	K	K	minutes
LN ₂	5.0	84	2.4	90	6	6
LN ₂ Pumped	10.0	74	7.9	95	21	20
LH ₂	14.5	30	13.5	78	48	20

LN ₂	5.2	81	2.6	87	6	10
LN ₂ Pumped	10.4	70	7.9	91	21	30
LH ₂	15.0	26	15.0	80	54	30

Table 1. The initial temperature, energy deposition, final temperature, temperature rise and maximum rep rate for ca. 5, 10 and 15 T central field. The first set of three rows is for coils with 75 turns per layer. The second set is for coils of 78 turns per layer and a less conservative estimate of magnetoresistance. All coils can maintain a flat top of one second. Decreasing the flat top to about half a second would allow one to achieve full field from a starting temperature about 1 K higher. (From Bob Weggel, 7/16/02)

Builds used in the finite element models:

#	r	z	dr	dz	nx	ny
1	.15	0	.098	1.0	16	16
2	.2	0	.002	1.0	1	16
3	.25	0	.098	1.0	16	16
4	.3	0	.002	1.0	1	16
5	.35	0	.098	1.0	16	16

Coil Description:

	Mode 1	Mode 2	Mode3
Number of Segments operating:	2	2	3
Coil Weight (kg)	1994.79	1994.79	3740.23
Number of turns per segment	624	624	624
Total number of turns active	1248	1248	1872
Layers in each coil segment	8	8	8
Turns per layer	78	78	78
Conductor radial t	.0116698 m .45944 in	.0116698 m .45944 in	.0116698 m .45944 in
Conductor Axial t	.012516m .49274359 in	.012516m .49274359 in	.012516m .49274359 in

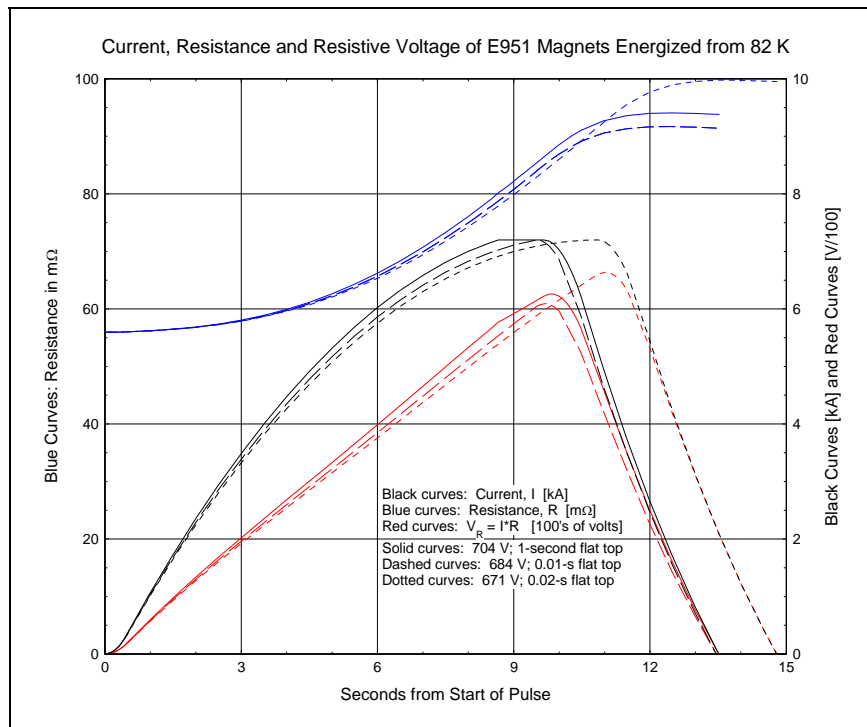
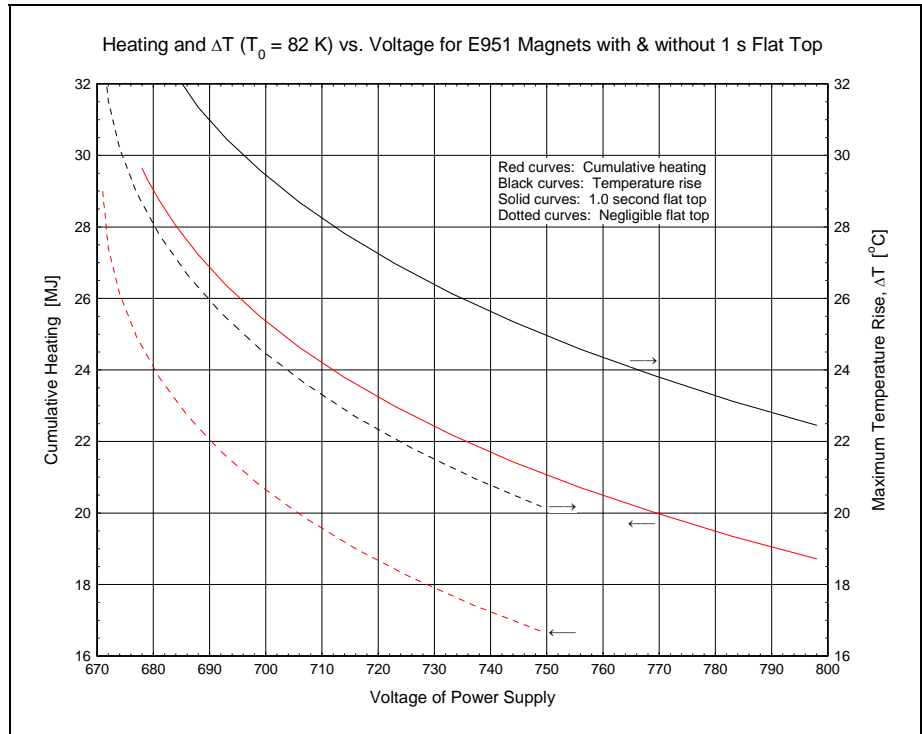
Max Operating Field Bore CL	5T	10T	15.0T
Max Field at Magnet			
Max Terminal Current	3600A	7200A	7200A
Coolant Working Fluid	77K LN2	65K LN2	30 K Helium Gas
Terminal Voltage	150V	300V	300V
Layer to Layer Volts	18	36	24
Turn-to-Turn Volts	0.12	0.24	.16
Design Life			1000 full power pulses
Cryostat Pressure - Initial Operating			15 atm
Number of .54 MVA power supplies	1	4	4
Mode of Ganging Supplies	None	2 X 2	2 X 2
Charge Time	7.2	6.3	15.3 sec
Initial Temperature	84K	74K	30K
Temp Rise	5.8K	21.7K	48.3K
Final temperature			78.3
Cumulative heating at end of pulse	2.7MJ	9.1MJ	15.2MJ

1. Longevity goal for the magnet is ~1000 pulses of 15.0 T peak central field.
2. Magnet should be capable of operation in three modes: 1) 5 T at 3600 A, with inner windings (inner two shells) pulsed at 150 V from ≤ 84 K; 2) 10 T at 7,200 A, with inner windings pulsed at 300 V from ≤ 74 K; 3) 14.5 T at 7,200 A, with entire magnet pulsed at 300 V from ≤ 30 K.
3. All vessels and piping, including the bore tube, should satisfy ASME pressure and vacuum codes.

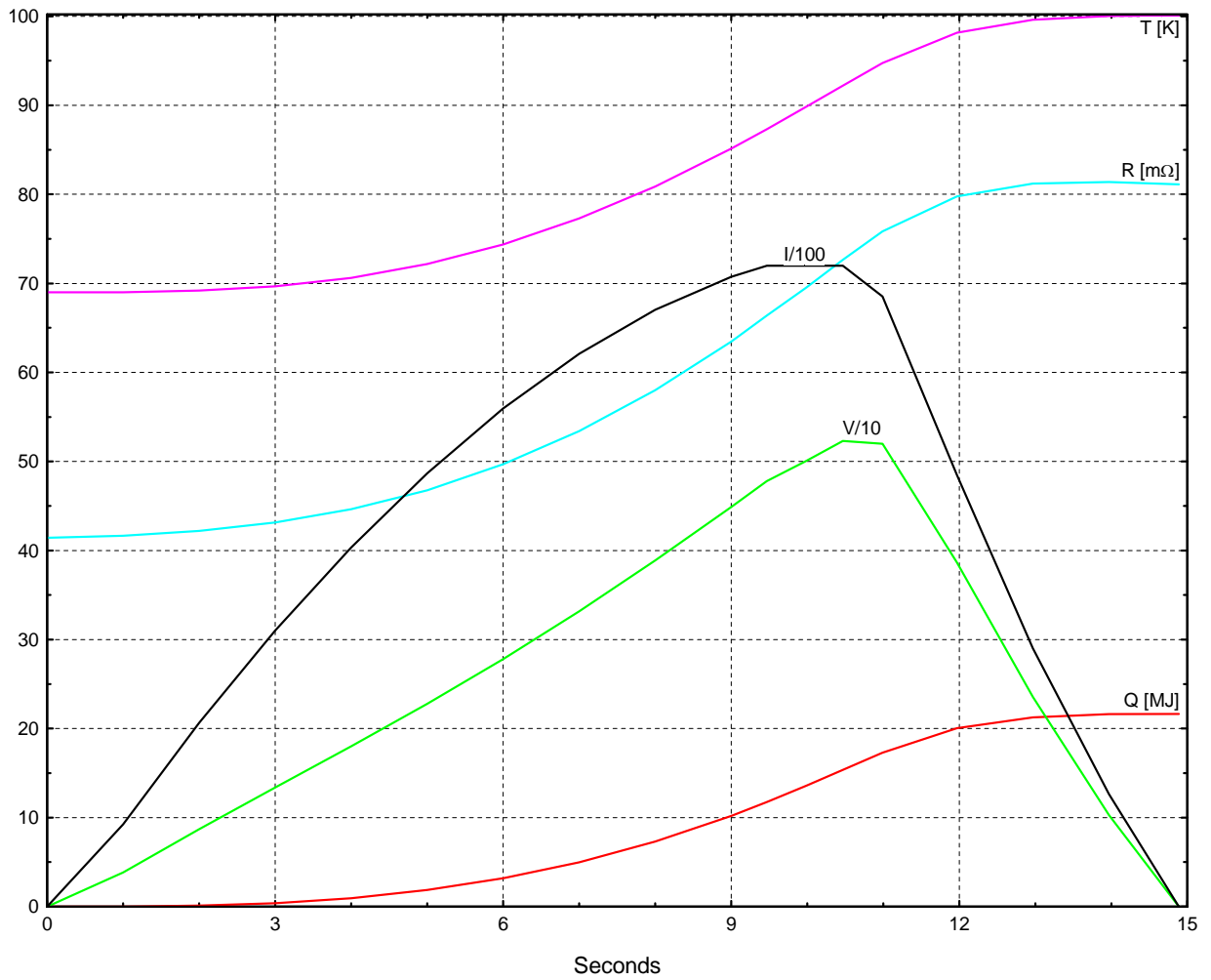
2.2 Circuit Simulations

2.2.1 LN2 82K Baseline

Bob did several calculations for us. His results show that it is better to run with as high a voltage as possible. For your cooldown calculations I recommend using as a baseline: 82deg K, 704 V, 1.0 sec flat top, 25 MJ energy deposition, del T 29 deg K.



Parameters of Pulse Coil Precooled to 69 K and Energized at 600 V to 7200 A



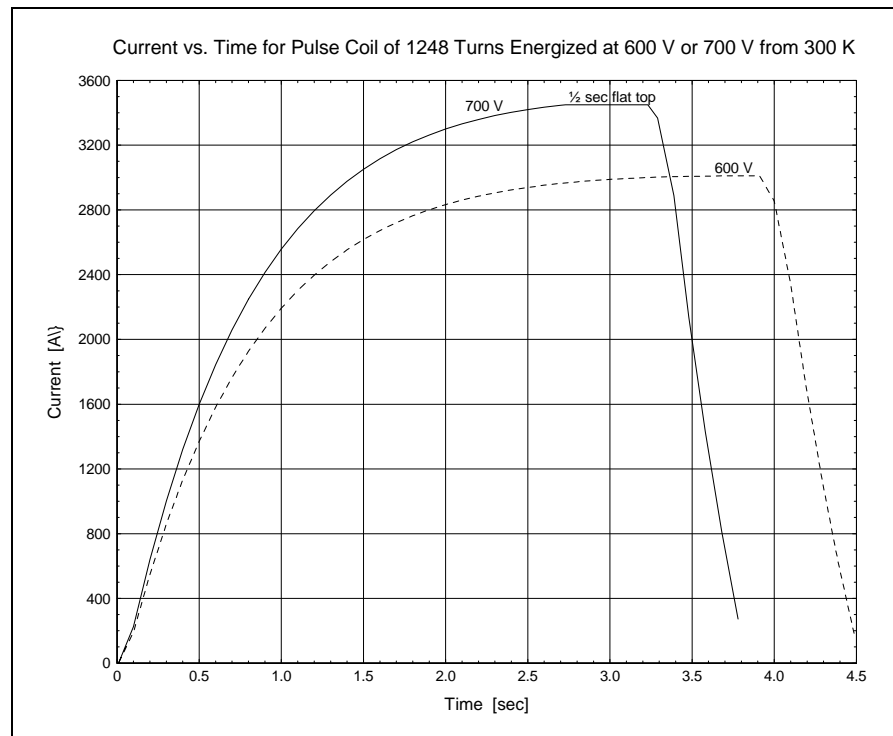
Bob Weggel's 10-14 analysis of the LN2 magnet operation

Room Temperature Operation

During tests at MIT a number of tests were performed at room temperature. The simulations for these and the resulting fields and measured temperatures are included in the MIT test results report, PSFC/JA-06-18

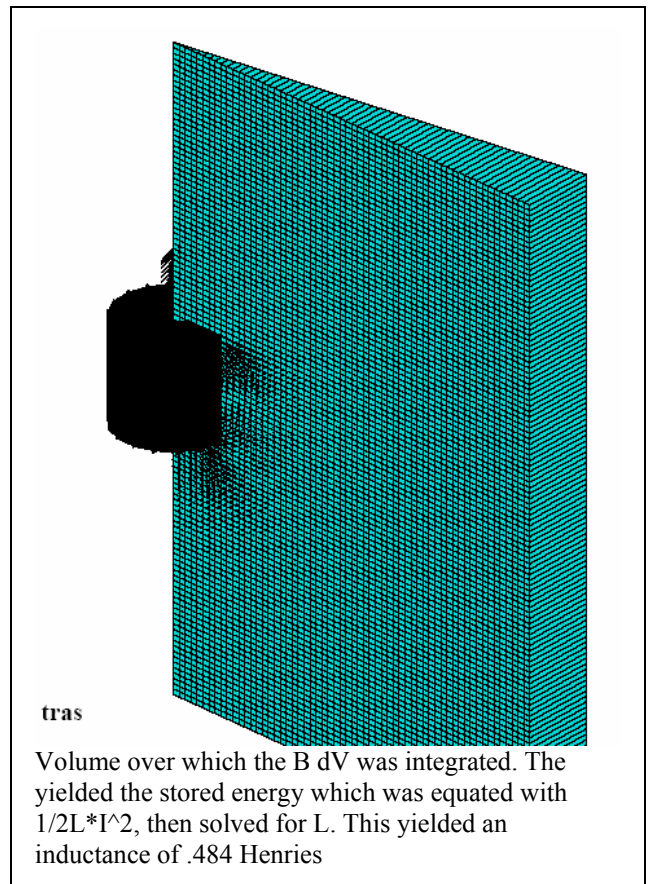
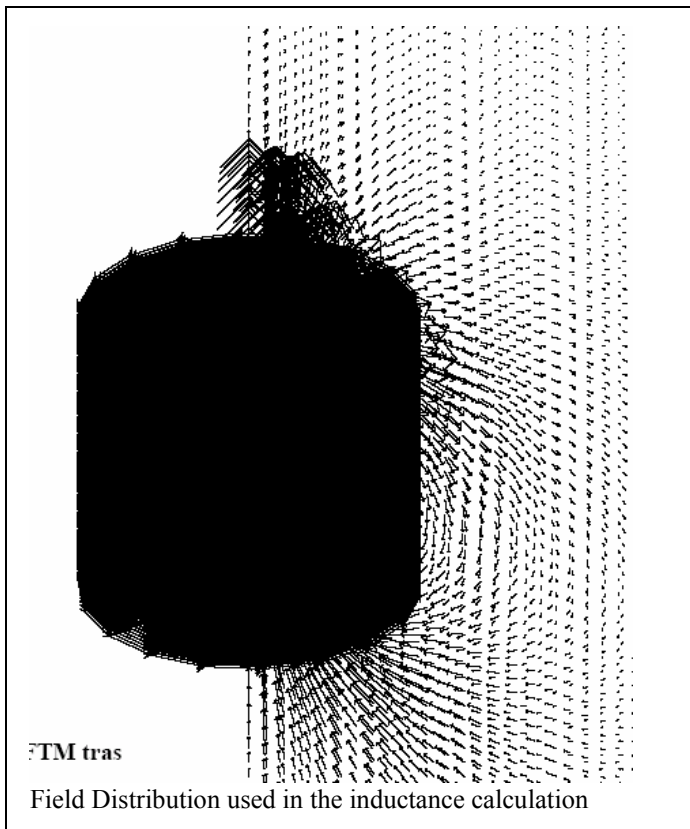
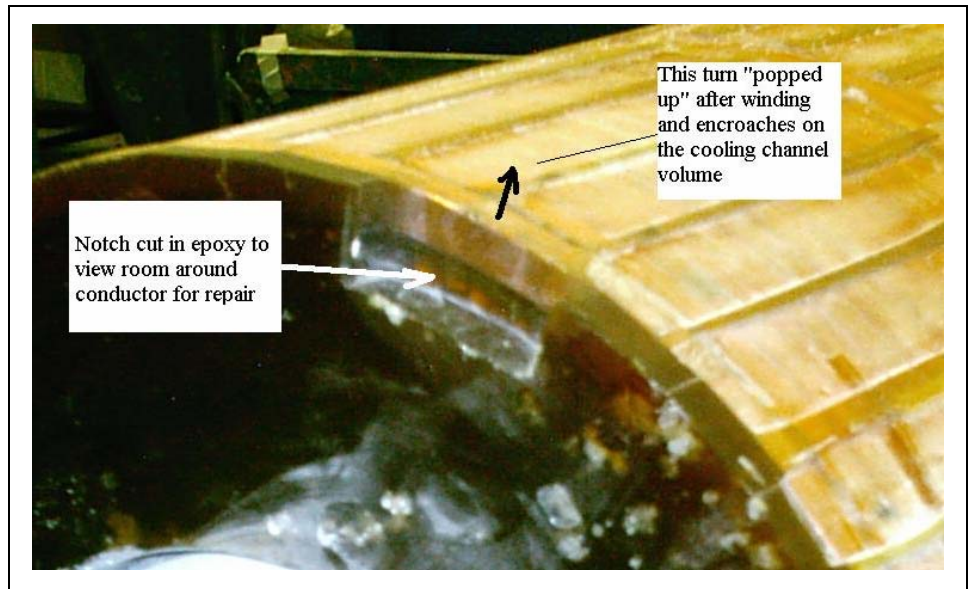
From a June 2003 correspondence between Bob Weggel and Harold Kirk,

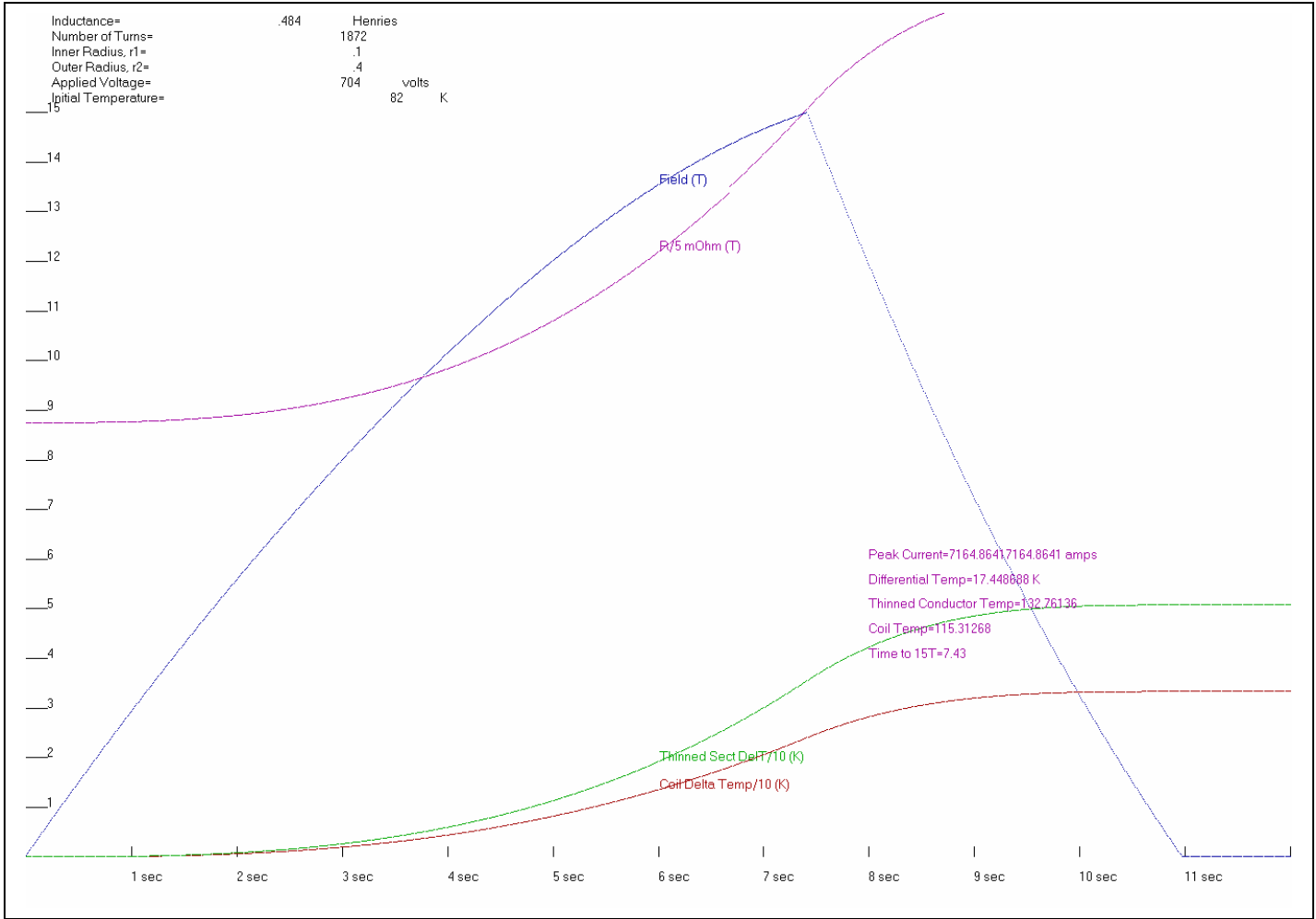
In reply to your request, the attached MS Word file P1room.doc graphs the current vs. time for a two-shell, 1248-turn pulse magnet operated from room temperature (300 K). The dotted curve is for a power supply of 600 V, such as for Cases #2 and #3, but with their two 3.6-kA, 300-V modules in series instead of parallel. At 600 V, the current tops out at 3011 A; the field is only 4.37 T. The solid curve is for 700 V, in order to reach 5.01 T (3450 A) with a flat top of $\frac{1}{2}$ second. For each magnet the temperature rise is about 7 K and the energy dissipated about $5\frac{1}{2}$ MJ.



Conductor Non-Conformance Simulation Circuit Simulation for the Displaced Segment #1 Conductor

Bob Weggels electrical simulations are the simulations of record. However a conductor in segment 1 shifted radially outward and encroached on the channel spaces. It was damaged during the machining operation. The preferred repair option is to





From a January 11 2006 email:

Dear Peter,

Thank you very much for showing Carl and me our magnet (and the Levitated Dipole Experiment as well). Please send me, at your convenience, the as-built parameters of the magnet. With the dimensions as designed, with 78 turns per layer, the inductance is 470 millihenries. Let me know if you would like a table or graph of its current versus time. If so, please send me values for the parameters that I will need concerning your power supply.

Bob Weggel
bob_weggel@mindspring.com

Magnet Quantities

	Segment 1	Segment 2	Segment 3	
Volume (m ³) 9.2362824e-2	.15393804	.21551326		
numturn	624	624	624	
weight (kg)	748.04651	1246.7442	1745.4419	3740.23
conlen (M)	641.4085	1069.0142	1496.6198	
tapelen (M)	5131.268	8552.1133	11972.959	
epoxyvol (M ³)	8.3126542e-3	1.3854424e-2	1.9396193e-2	
Num Ramp/Trans	32	32	32	

Cryostat Bore Tube Geometry		
Building from the Magnet ID and working towards the centerline:		
Component	Thickness (m)	Radius (m)
The ID of the magnet winding		.15-.98/2= .101
Coolant Channel	.002	.099
Cold Cryostat Shell	.004762(3/16in.)	.094237
Vacuum Space	.008	.086237
Vacuum shell	.0005	.085737
Strip heater	.001	.084737
This leaves a clear bore diameter of .16947m		

3.0 References

- [1] Fusion Ignition Research Experiment Structural Design Criteria; Doc. No. 11_FIRE_-DesCrit_IZ_022499.doc; February, 1999
- [2] "General Electric Design and Manufacture of a Test Coil for the LCP", 8th Symposium on Engineering Problems of Fusion Research, Vol III, Nov 1979
- [3] Product Literature, Inco Alloys International, Inc Huntington West Virginia 25720, USA
- [4] "Mechanical, Electrical and Thermal Characterization of G10CR and G11CR Glass Cloth/Epoxy Laminates Between Room Temperature and 4 deg. K", M.B. Kasen et al , National Bureau of Standards, Boulder Colorado.
- [5] "US ITER Insulation Irradiation Report Program Final Report" August 31 1995 Reed, Fabian, Shultz
- [6] Effect of Face Compression on Interlaminar Shear Strength of Polyimide/S2 Glass Laminate Insulators - Preliminary Report" H.Becker, T. Cookson (GDC) June 24 1985
- [7] "Shear Compression Tests for ITER Magnet Insulation" Simon, Drexler, Reed, Advances in Cryogenic Engineering Materials Vol 40 (1994)
- [8] "Case Studies in Superconducting Magnets" Design and operational Issues, by Yukikazu Iwasa , Plenum Press, New York and London 1994
- [9] CryoCoat™ UltraLight™ Insulation presented by:Michael L. Tupper COMPOSITE TECHNOLOGY DEVELOPMENT INC. 1505 Coal Creek Drive Lafayette, Colorado 80026 January 2001
- [10] <http://www.lakeshore.com/temp/sen/crtd.html>
- [11] CPC Cryolab Vacuum Tight Pressure Relief Discs PR-3 Series Data Sheet
- [12] Mark's Standard Handbook for Engineers, Sevnth Edition, Pages 3-60 through 3-63, Pipe friction factors and entry/exit losses.
- [13] Air Liquide Web Site <http://www.airliquide.com> - For some Helium and Nitrogen Properties
- [14] MLI Specification from: "LHC Interaction Region Quadrupole Cryostat", Thomas H. Nicol, Fermi National Accelerator Laboratory December 3, 1998, P.O. Box 500, Batavia, IL 60510

Table 4.8: "Typical" Values of Radiation Heat Flux [4.14]

<i>Material</i>	$T_{cl} \sim T_{wm}$ [K]	$[e_r]_{cw}$	q_r [mW/m ²]	
Copper as received	4~20	0.03*	0.3	
	4~80	0.06	140	
	80~300	0.12	55,000	
	mechanically polished	4~20	0.01*	0.1
		4~80	0.02	46
		80~300	0.06	27,000
Stainless steel as received	4~20	0.06*	0.6	
	4~80	0.12	280	
	80~300	0.34	155,000	
	mechanically polished	4~20	0.04*	0.4
		4~80	0.07	160
		80~300	0.12	55,000
	electropolished	4~20	0.03*	0.3
		4~80	0.06	140
		80~300	0.10	46,000
Aluminum as received	4~20	0.04*	0.4	
	4~80	0.07	160	
	80~300	0.49	224,000	
	mechanically polished	4~20	0.03*	0.3
		4~80	0.06	140
		80~300	0.10	46,000
	electropolished	4~20	0.02*	0.2
		4~80	0.03	70
		80~300	0.08	37,000
	foil	4~80	0.01	20
		80~300	0.06	28,000

* One half of the 4.2~80 K value.

From [8]: page 152 "Case Studies in Superconducting Magnets"
Design and operational Issues, by Yukikazu Iwasa, Plenum Press,
New York and London 1994

4.0 Structural Design Criteria, Materials and Conductor Specifications

Lacking a specific design code jurisdiction, fusion project criteria are used for guidance in coil design [1]. The referenced FIRE design document allows the primary membrane stress to be based on the lesser of 2/3 of the Yield Strength (S_y) or 1/2 of the Ultimate Strength (S_u). The ASME Code bases the primary stress on 1/3 ultimate. The fusion project based criteria is based on a distinction between coils that are supported by cases and those that are not. For structural elements ASME-like criteria are adopted with membrane stresses remaining below the maximum allowable stress, S_m , where S_m is the lesser of $2/3 \cdot \text{yield}$ or $1/3$ ultimate. Bending discontinuity, and secondary stresses are treated in a manner similar to the ASME Code. Guidance for bolting and column buckling is taken from AISC, with average net section bolt stresses kept below $0.6 \cdot \text{yield}$. Yield Strength and Tensile Strength properties are taken at the loaded temperature. However because the cryostat must be safely pressure tested at room temperature, stresses will be checked based on the room temperature properties. The room temperature physicals for the proposed 316 bolts are:

High Strength Bolts Specs:
 ASTM A193 Grade B8M - Class 2 - Type 316
 for 3/4" diameter and under:
 $S_u = 110,000$ psi ,
 $S_y = 95,000$ psi

The cryostats are to be qualified in accordance with ASME VIII. Qualification of all the weld details, shell thicknesses, nozzle reinforcements, and saddle or support details of these vessels will be done at the final design stage. The conceptual design sizing presented here is intended to ensure adequate space allocation and cold mass performance.

The magnet is to be seismically qualified in accordance with the Uniform Building Code.

4.1 Conductor Properties and Allowables

Mechanical properties for some of structural materials and coil components are given in this section. Tensile yield strength for oxygen free copper is given in Fig. 1 at 4 K, 76 K and at 296 K. Yield and Ultimate strength data is given for several variants of copper at RT and 77K in Table 1.

The inner skin of the bore of the solenoid is allowed to reach the yield stress. - Treating this stress as a bending stress with a $1.5 \cdot S_m$ allowable with S_m based on $2/3$ Yield.

The maximum stress in the three segment coil is 166 MPa. Half hard copper should satisfy this requirement at liquid nitrogen temperatures. In order to minimize the difficulties of winding the first layers of the inner solenoid, It is intended that the conductor physical properties and degree of cold work be selected to just satisfy a 166 MPa yield stress.

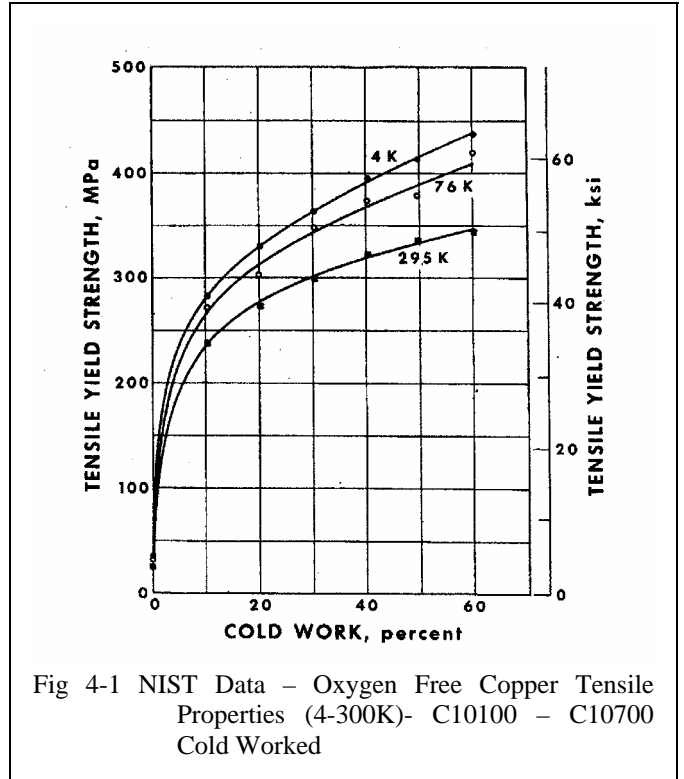


Fig 4-1 NIST Data – Oxygen Free Copper Tensile Properties (4-300K)- C10100 – C10700 Cold Worked

Table 4-1. Properties of Variants of Copper

Variant	Yield Mpa at RT	Yield Mpa at 77 K	Ult., Mpa at RT	Ult, Mpa at 77K

C10100/C10700 80%CW	380		420	500
C10100 Becker/C- Mod 60%CW	308	373	350	474

Table 4-2 Interpolated values:, Work hardened copper-, OFHC c10100 60% red

temp deg k	77	90	100	125	150	200	250	275	292
yield	374	369.	365.	356.	347.	328.	317.	312.	308.
ultimate	476.	466.	458.	439.	420.	383.	365.	356.	350.

The conductor is specified as half hard in the spec. Everson has purchased ¼ hard conductor to ease the bending operation, with the expectation that the cold work associated with the forming process will produce an adequate yield. From Table 4-3, ¼ hard copper would have a yield of 30 ksi or 207 MPa. From Figure 4-1 this would correspond to cold work of about 15%. The bending operation would introduce an additional 6% (see section 4.4) Hardness is assumed to correlate with %cold work. Figure 4-1 plots yield as a function of cold work. Full hard copper would have a yield of about 350 MPa at 77°K, and an ultimate of about 450 MPa which gives a primary membrane allowable (lesser of 2/3Sy or 1/2Su for externally supported coils) of about 200 MPa, At 4°K 30% CW reaches these levels.

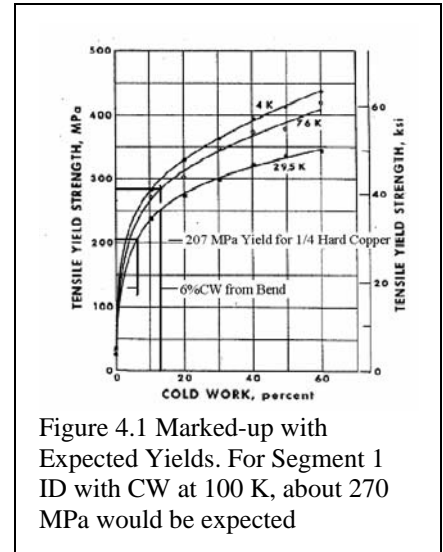


Table 4-3 Room Temperature Properties of Copper, C10200 (OFHC)

Standard	Former	Yield Min	Yield Max	Tensile Min	Tensile Max	Elong %	Rockwell F
O25	Hot Rolled-Annealed	11 ksi (2) 76 MPa		34 ksi (2) 234 MPa(2)		45	
H00	Eighth Hard	28 ksi (2) 193		36 ksi (2) 248 MPa (2)		30	54-82 (1) 60 (2)
H01	Quarter Hard	30 ksi (2) 207		38 ksi (2) 262 MPa (2)	25	25	60-84 (1) 70(2)
H02	Half Hard	36 ksi (2) 248 MPa (2)		42 ksi (2) 290 MPa (2)		14	77-89 (1) 84(2)
H03	¾ Hard						82-91 (1) 85(2)
H04	Hard	40 ksi (2) 276 MPa (2)		45 ksi (2) 310 MPa (2)		6 to 20	86-93 (1) 87 (2)
H06	Extra Hard						88-95 (1)
H08	Spring	50 ksi (2) 345 MPa (2)		55 ksi (2) 379 MPa (2)		4	90-97 (1)

(1) ASTM B152

(2) Copper Development Association Web Page www.copper.org flat form

Table 4-4 Room Temperature Properties of Copper, C10100 (OFHC) (Identical to C10200)

Standard	Former	Yield Min	Yield Max	Tensile Min	Tensile Max	Elong %	Rockwell F
O25	Hot Rolled-Annealed	11 ksi (2) 76 MPa		34 ksi (2) 234 MPa (2)		45	
H00	Eighth Hard	28 ksi (2) 193		36 ksi (2) 248 MPa (2)	30	30	54-82 (1) 60 (2)
H01	Quarter Hard	30 ksi (2) 207		38 (2) 262 MPa (2)	25	25 to 35	60-84 (1)
H02	Half Hard	36 ksi (2) 248		42 ksi (2) 290 MPa (2)		14	77-89 (1)
H03	¾ Hard						82-91 (1)
H04	Hard	40 ksi (2) 276 MPa (2)		45 ksi (2) 310 MPa (2)		6 to 20	86-93 (1) 87 (2)
H06	Extra Hard						88-95 (1)
H08	Spring	50 ksi (2) 345 MPa (2)		55 ksi (2) 379 MPa (2)		4	90-97 (1)

(1) ASTM B152

(2) Copper Development Association Web Page www.copper.org flat form

It would be wise to do a tension test of the full copper channel section prior to soldering the superconductor. Surface hardness may not be a reliable indicator of the full section strength.

Typical values of mechanical strength for several different types of composite insulating materials is given in Table 2.

Copper Fatigue

The design requirement for the magnet is 1000 full field shots at cryogenic Temperatures. The following data is for room temperature fatigue. Fatigue at LN2 to 120 degrees K is better.

CRYOGENIC PROPERTIES OF COPPER AND COPPER ALLOYS

MATERIAL	C10100, C10200, C10700 COPPER (COLD-WORKED)	PROPERTY STRESS-CONTROLLED AXIAL FATIGUE LIFE; AIR, 295 K
-----------------	------------------------------------------------	---------------------------------------------------------------------

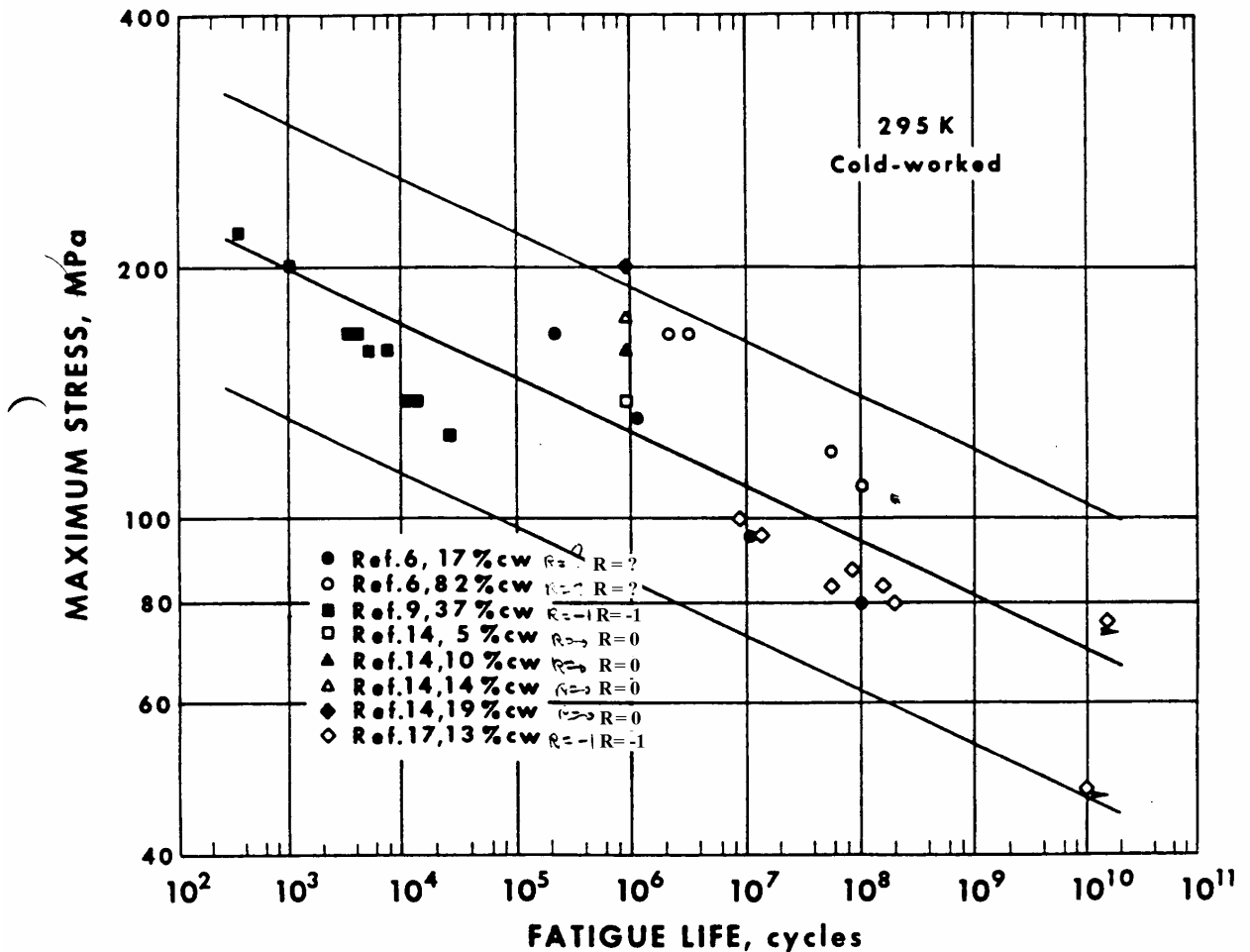


Figure 3. The data shown were used to compute the regression of maximum stress upon the number of cycles to failure [Equation (2)]. All data are presented in Table 2. Tests discontinued before failure are marked by an arrow. Product forms include bar and sheet. The percent of cold work refers to reduction of area or thickness. The R ratios are discussed in the text.

4.2 Insulation Allowable

Warp and Fill tensile allowable with a Factor of Safety F.S.=3 or 1/3 ultimate is 167 Mpa. Note that for tension normal to the reinforcement, there is no capacity listed. Normally primary tension in this direction is not allowed. Secondary or strain controlled tensile stresses are nearly unavoidable in bonded conductor arrays. Small cracks often develop near corners, and multi-layered insulations are used to limit the likelihood of through cracks. For MECO magnets, Kapton tape is specified to mitigate the effects of these cracks on electrical insulating integrity and this has the effect of reducing the tensile bond strength. It also reduces the winding pack modulus in the direction normal to the Kapton tape. The interplay between modulus and secondary displacement controlled strains will be resolved with measured properties and strengths of a representative impregnated array of conductors.

Table 4-5. Insulating Material Strengths (Ref 27)

	Mpa @4°K	Mpa @77°K	Mpa @292°K
Comp.Strength Normal to Fiber			
G-10CR	749(Ref 4)	693(Ref 4)	420 (Ref 4)
G-11CR	776(Ref 4)	799(Ref 4) 900(Ref 7)	461 (Ref 4)
CTD 101K AR irradiated	1260 (ave) (Ref 5)		
CTD-112P irradiated	1200 (ave) (Ref 5)	1150(Ref 5 p 47)	
Polyimide/S2 Glass Laminate			1033 Mpa, Ref [6]
Tensile Strength (Warp)			
G-10CR	862 (Ref 4)	825(Ref 4)	415 (Ref 4)
G-11CR	872(Ref 4)	827(Ref 4)	469 (Ref 4)
Tensile Strength (Fill)			
G-10CR	496(Ref 4)	459(Ref 4)	257 (Ref 4)
G-11CR	553(Ref 4)	580(Ref 4)	329(Ref 4)

Tensile Strain Allowable Normal to Plane

No primary tensile strain is allowed in the direction normal to the adhesive bonds between metal and composite. Secondary strain will be limited to 1/5 of the ultimate tensile strain. In the absence of specific data, the allowable working tensile strain is 0.02% in the insulation adjacent to the bond.

Table 3 provides modulus data for G-10 composite materials.

Table 4-6. Modulus of Elasticity for G-10 at several temperatures.

Temp Deg. K	G-10 Warp/Fill Gpa	G-10 Normal Gpa	Epoxy Only Gpa
295	27.8	14.0	3.81
250	29.5	16.5	5.25
200	31.3	18.8	6.69
150	32.5	20.5	7.84
100	33.0	21.5	8.54
76	33.5	21.8	8.68

September 11 2007:

I just got off the phone with Matt Hooker from CTD. In some cure temperature profiles, they recommend a 175C post cure for CTD 101K. We used 130C for our magnet, but could have gone to 175C – so the chemistry of the epoxy was not degraded. – But the glass transition temperature is 110C which means the epoxy loses much of it's strength above 110 deg C. This shouldn't be a problem because there were almost no loads on the magnet. Small strains could have occurred – spring back from the winding process, gravity loads maybe a little Lorentz force, so Matt recommended and I concur, that the hi pot tests be re-done.

Maybe the Teflon in the gland nut should be checked? The expansion may have disturbed things.

The large bellows on the vacuum jacket should be able to absorb the expansion of the cryostat, but vacuum should be re-checked to be sure.

-Peter

I would let it cool gradually. If the magnet is still beyond the glass transition temp of 110C then the differential temperature of the cooldown could do more harm than just being at an elevated temperature. -Peter

4.3 Stainless Steel Properties and Allowable

Additional properties for materials used in the BNL magnet system conceptual design are provided in Tables 4-7 to 4-10.

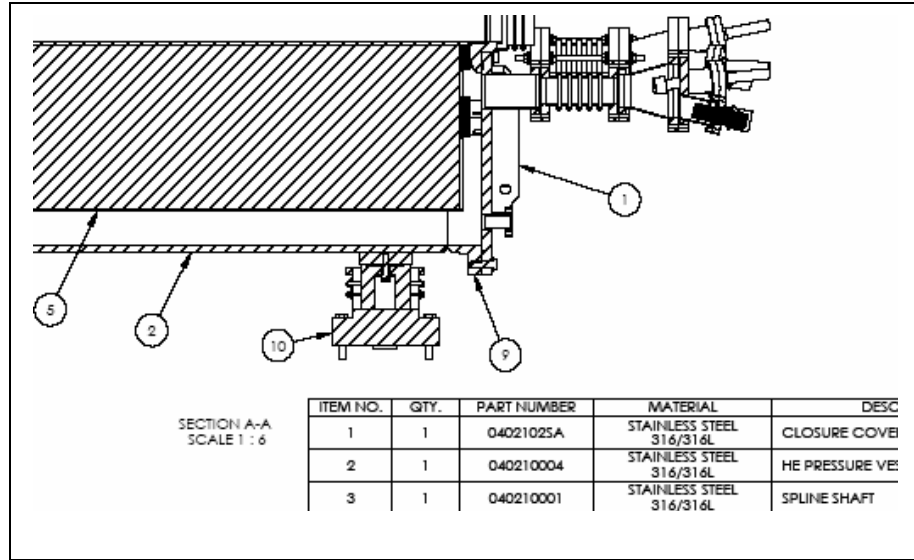


Table 4-7. Tensile Properties for Stainless Steels and Aluminum

Material	Yield 4 deg K (MPA)	Ultimate 4 deg K, (Mpa)	Yield, 80 deg. K (Mpa)	Ultimate, 80 deg. K (Mpa)	Yield, 292 deg K (Mpa)	Ultimate, 292 deg K (Mpa)
316 LN SST	992	1379			275.8]	613]
316 LN SST Weld	724	1110			324	482
304 SST 50% CW			1344 (195 ksi)	1669	1089	1241
304 Stainless Steel (Bar, annealed)			282 (40.9ksi)	1522	234	640
Aluminum 6061T6	362(20K)	496(20K)	275.8		275 Mpa 40ksi	310 Mpa 45ksi
Alum 6061 Weld	259(4K)	339(4K)				

Table 4-8. Coil Structure Room Temperature (292 K) Maximum Allowable Stresses, S_m = lesser of 1/3 ultimate or 2/3 yield, and bending allowable=1.5* S_m

Material	S_m	1.5 S_m
316 LN SST	183Mpa (26.6 ksi)	275Mpa(40ksi)
316 LN SST weld	160MPa(23.2ksi)	241MPa(35ksi)
304 SST 50% CW		
304 Stainless Steel	156MPa(22.6ksi)	234 MPa (33.9ksi)

(Bar,annealed)		
Aluminum 6061T6		

Table 4-9. Coil Structure Cryogenic (80 K) Maximum Allowable Stresses, S_m = lesser of 1/3 ultimate or 2/3 yield, and bending allowable= $1.5 \cdot S_m$

Material	S_m	$1.5S_m$
304 SST 50% CW	556 Mpa (80 ksi)	834 (120 ksi)
304 Stainless Steel (Bar,annealed)	188MPa (27ksi)	281 MPa (40.9ksi)

Table 4-10. Coil Structure Cryogenic (4 K) Maximum Allowable Stresses, S_m = lesser of 1/3 ultimate or 2/3 yield, and bending allowable= $1.5 \cdot S_m$

Material	S_m	$1.5S_m$
316 LN SST	459.6 Mpa (66.7 ksi)	689 (100 ksi)
316 LN SST Weld	366MPa (53.2ksi)	550 MPa (79.7ksi)
Alum 6061T6	165 MPa	248 MPa
Alum 6061T6 Weld	113 MPa	169.5 MPa

Inconel 718 was used in the C-Mod Drawbars and oblong Pins. Physical properties were measured from samples of the forgings used for these drawbars. This data is summarized in Table 8 . These values are representative of strengths expected for the 718 bolting specified for MECO.

Table 4-11. Inconel 718 Tensile data (hardening for 1 hr. at the most favorable temp, either 1750 or 1950deg F from ref[3])

	4 in. Bar	5/8in Bar	5/8in. Bar
	292 degK	292K	77degK
Yield	165ksi	180 ksi	173ksi
Ultimate	195ksi	208 ksi	237ksi
Modulus of Elasticity	29.8e6	29.8e6psi	31.e6 psi
Density	.296 lb/in ³	.296 lb/in ³	.296 lb/in ³

Used as bolting, 718 would have a $0.6 \cdot 173 = 100$ ksi maximum allowable operating stress

4.4 Conductor and Insulation systems design and packing fraction

Conductor Keystoning:

$H/(2*r) = .012/.1/2 = 6\%$ (elastic strain) For Plastic bending, (poisson=.5) the Keystoning contraction is 3%

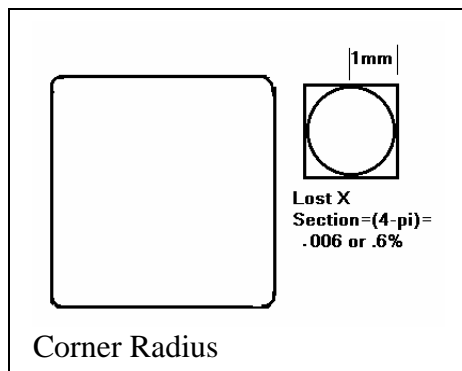
This is a function of radius. Three Keystone specs are suggested. The keystone geometry for the first segment should be $.012/.15/2 * .5 = 2\%$

The worst case loss in packing fraction is 1%, Average loss is .5%

Keystone allowances in outer two segments are 1.2%, and .86%. Packing fraction losses in outer two segments are .15%, and .007%

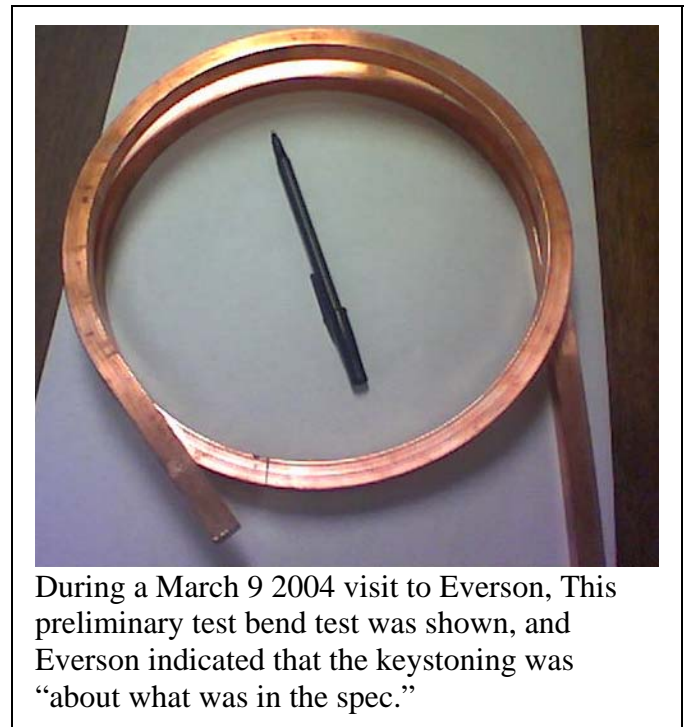
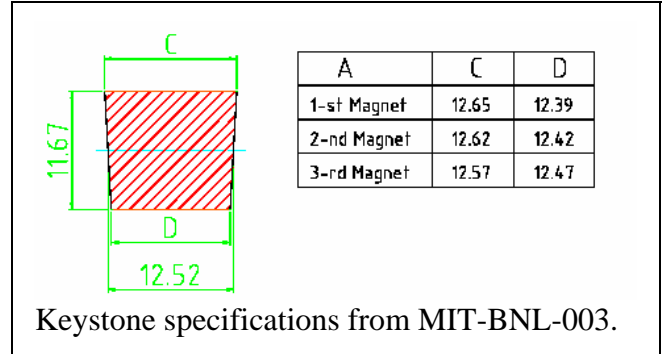
Effect of Corner Radius:

Whole Magnet loss of .2% + Corner Loss of .6% = .8%



Insulation System:

In the axial or "Z" direction, there are four thicknesses of 3 mil glass tape between each conductor. In the radial direction, there are an additional two layers of glass tape, interleaved with Kapton tape.

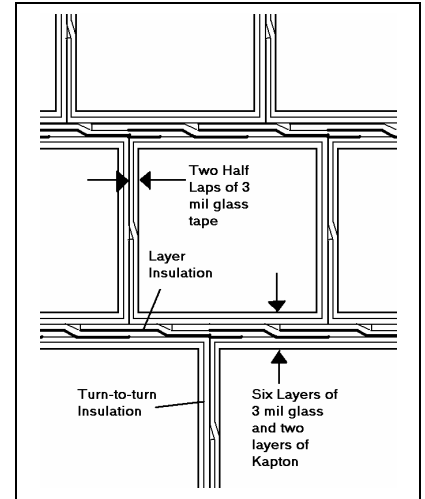


```

_data 1,.15,0,.098,1.0,16,16
_data 2,.2,0,.002,1.0,1,16
_data 3,.25,0,.098,1.0,16,16
_data 4,.3,0,.002,1.0,1,16
_data 5,.35,0,.098,1.0,16,16
_read n,r,z,dr,dz,nx,ny
_print "current Densities"
_print 4.5e6/dr/dz
_print 4.5e6/dr/dz/100/100
_clear
let ntpc=624
print "conductor dimensions"
let rdim=(dr-(.003*2+.001*2)/39.37)/8-.003*4/39.37
let zdim=dz/(ntpc/8)-.003*4/39.37
print "radial dim";rdim;"m";rdim*39.37;"in"
print "Axial dim";zdim;"m";zdim*39.37;"in"
print "packing fraction=";ntpc*rdim*zdim/dr/dz
print
print "conductor dimensions with 2 millimeter channel tolerance"
let dr=dr-.002
let rdim=(dr-(.003*2+.001*2)/39.37)/8-.003*4/39.37
let zdim=dz/(ntpc/8)-.003*4/39.37
print "radial dim";rdim;"m";rdim*39.37;"in"
print "Axial dim";zdim;"m";zdim*39.37;"in"
print "packing fraction=";ntpc*rdim*zdim/(dr+.002)/dz
end
conductor dimensions
radial dim 1.1919799e-2 m .4692825 in
Axial dim 1.2515712e-2 m .49274359 in
packing fraction= .94991124

conductor dimensions with 2 millimeter channel tolerance
radial dim 1.1669799e-2 m .45944 in
Axial dim 1.2515712e-2 m .49274359 in
packing fraction= .92998827

```



These packing fractions are based on the coil winding pack and exclude the channel. If the 2 mm channel is included, the packing fraction drops to .911.

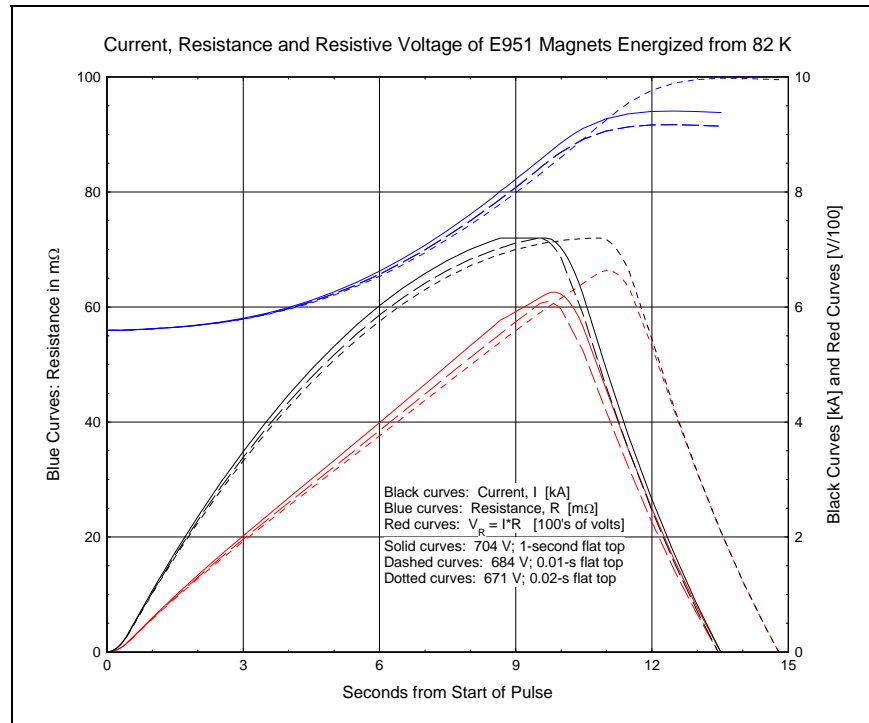
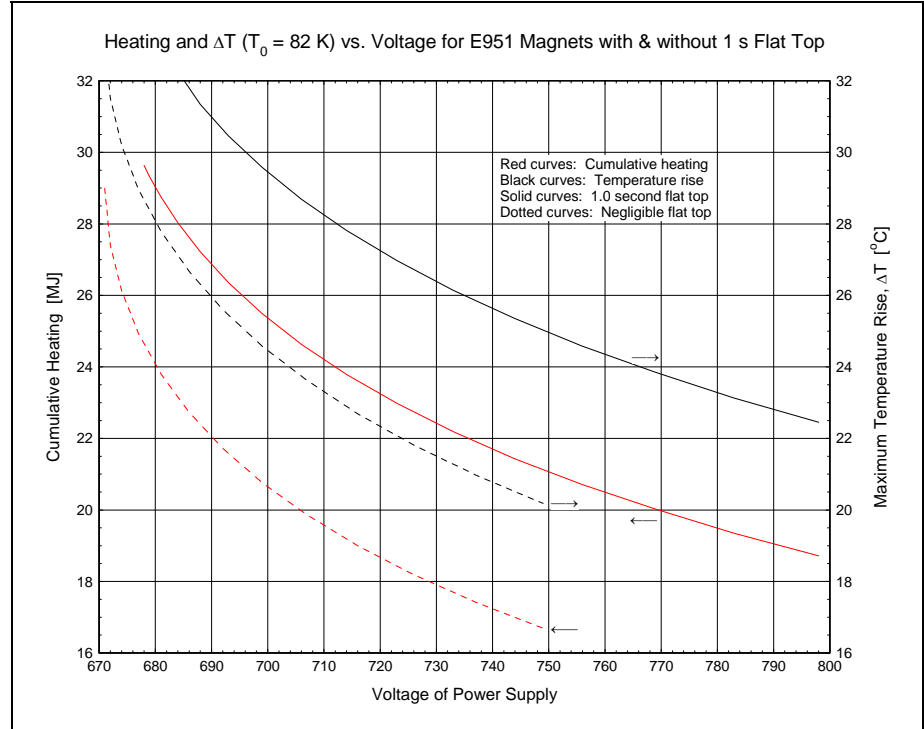
6.0 Electrical Simulations, Field and Iron Material Force Calculations

6.1 LN2 82K Baseline Circuit Simulation

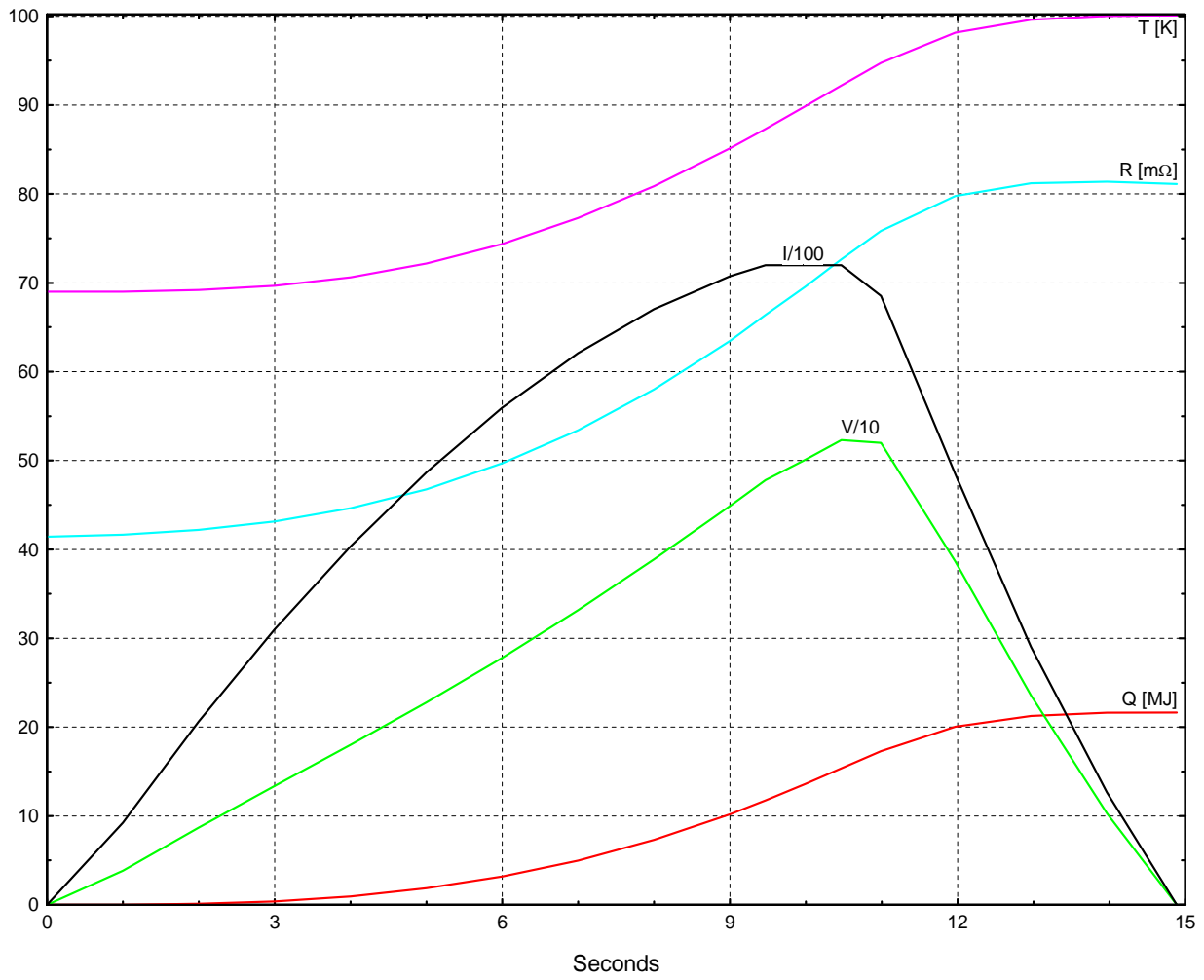
Bob Weggels electrical simulations are the simulations of record. Some of the results are repeated here for reference.

Bob did several calculations . His results show that it is better to run with as high a voltage as possible.

For your cooldown calculations Harold Kirk recommends using as a baseline: 82deg K, 704 V, 1.0 sec flat top, 25 MJ energy deposition, del T 29 deg K.



Parameters of Pulse Coil Precooled to 69 K and Energized at 600 V to 7200 A

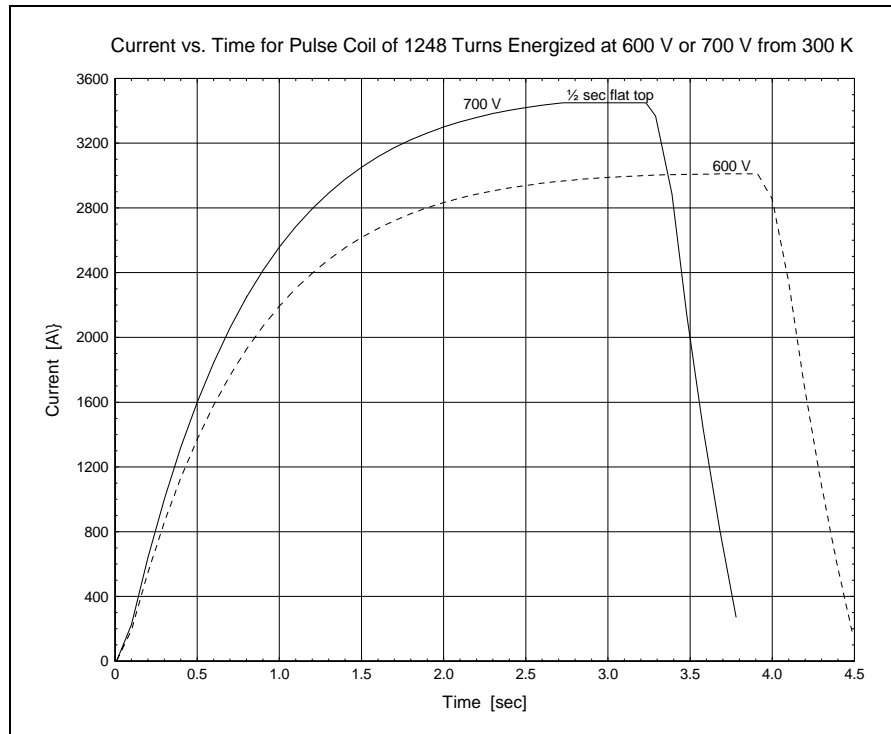


Bob Weggel's 10-14 analysis of the LN2 magnet operation

Room Temperature Operation?

From a June 2003 correspondence between Bob Weggel and Harold Kirk,

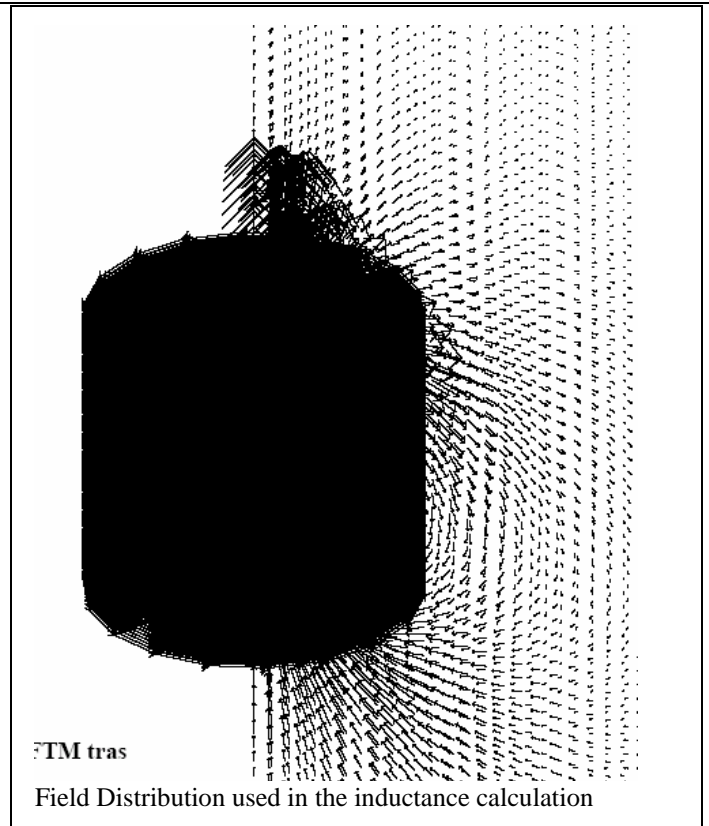
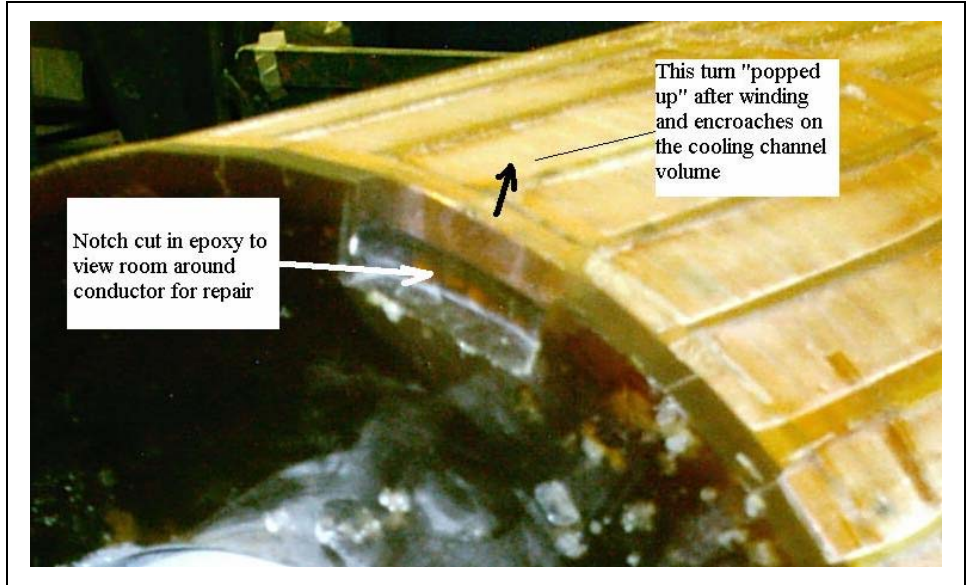
In reply to your request, the attached MS Word file P1room.doc graphs the current vs. time for a two-shell, 1248-turn pulse magnet operated from room temperature (300 K). The dotted curve is for a power supply of 600 V, such as for Cases #2 and #3, but with their two 3.6-kA, 300-V modules in series instead of parallel. At 600 V, the current tops out at 3011 A; the field is only 4.37 T. The solid curve is for 700 V, in order to reach 5.01 T (3450 A) with a flat top of ½ second. For each magnet the temperature rise is about 7 K and the energy dissipated about 5½ MJ.

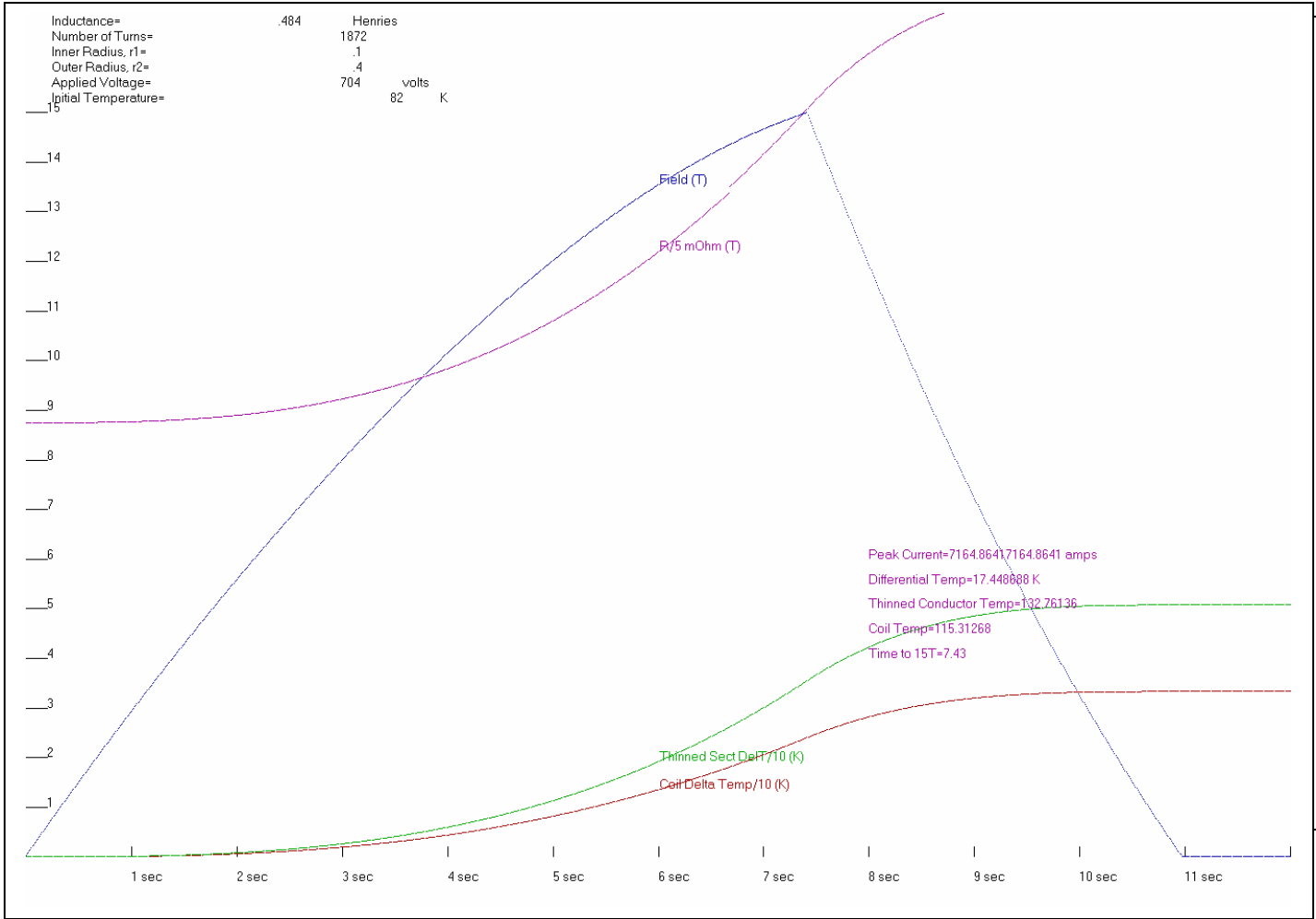


6.2 Conductor Non-Conformance Simulation

Circuit Simulation for the Displaced Segment #1 Conductor

A conductor in segment 1 shifted radially outward and encroached on the channel spaces during impregnation, or possibly lead break-out formation. . It was damaged during the machining operation. The intention of this section is to qualify the small local loss of conductor cross section. A full repair recovering the full cross section would be difficult, would endanger the schedule and expose neighboring conductors to risk of damage. The analysis approach is to reasonably duplicate Bob Weggels simulations, and then track the temperature that a reduced conductor cross section would develop. It is assumed that this small length of conductor will not effect the total resistivity of the coil and does not need to be included in the circuit calculations. The problem is assumed adiabatic – no thermal conduction along the conductor length is included.





From a January 11 2006 email:

Dear Peter,

Thank you very much for showing Carl and me our magnet (and the Levitated Dipole Experiment as well). Please send me, at your convenience, the as-built parameters of the magnet. With the dimensions as designed, with 78 turns per layer, the inductance is 470 millihenries. Let me know if you would like a table or graph of its current versus time. If so, please send me values for the parameters that I will need concerning your power supply.

Bob Weggel
bob_weggel@mindspring.com

6.3 Circuit Simulation with 6061-T6 Buss Bar

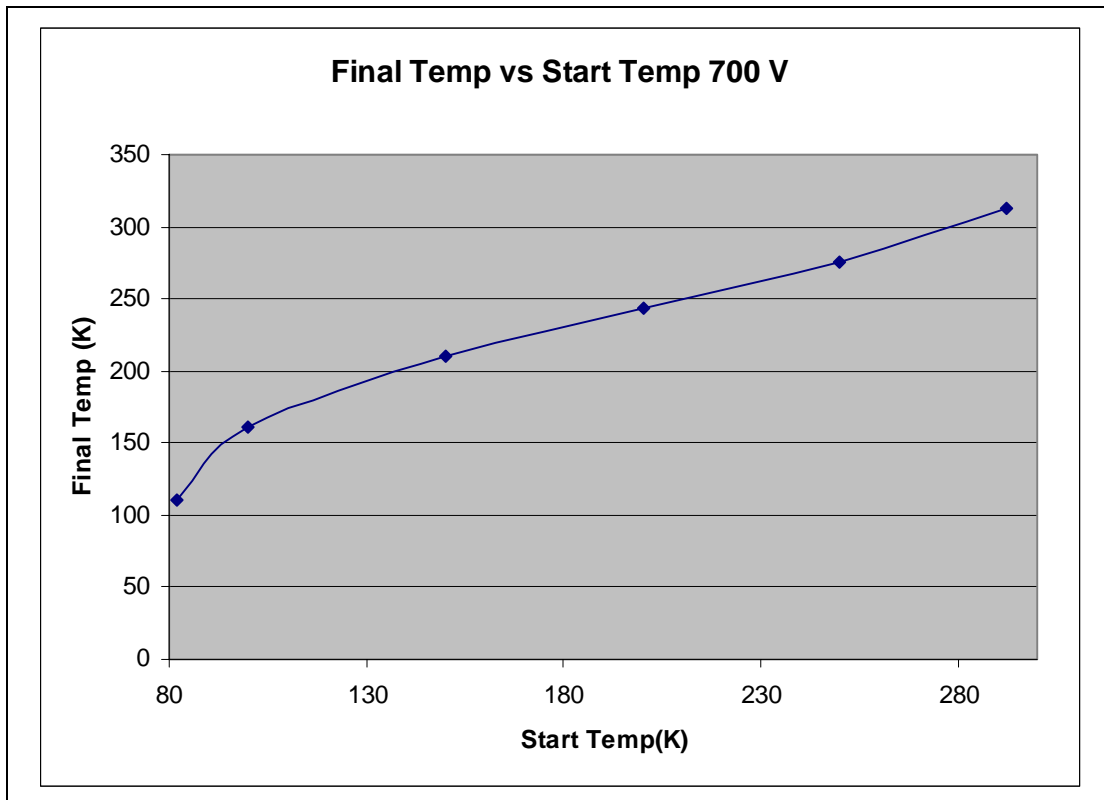
Material	Conductivity		Resistivity	Reference
	(% IACS)	(Siemens/m)	(Ohm-m)	(See Endnotes)
<i>Aluminum</i>				
Pure	61.00	3.538E+07	2.826E-08	ECTM
99.99%	64.94	3.767E+07	2.655E-08	CSNDT
99.99%	64.94	3.767E+07	2.655E-08	ALASM
6061-T4	37.60 - 40.50	2.265E+07	4.415E-08	NDT Mag
6061-T4	40.00		4.300E-08	ALASM
6061-T6 and -T9	40.00 - 44.80	2.459E+07	4.066E-08	NDT Mag
6061-T6	43.00		4.000E-08	ALASM
6062-F	47.00 - 51.00	2.842E+07	3.519E-08	NDT Mag
6062-T4	43.50 - 44.00	2.538E+07	3.941E-08	NDT Mag
6062-T6	44.70 - 49.50	2.732E+07	3.661E-08	NDT Mag
6063-O	58.00		3.000E-08	ALASM
6063-T1	50.00		3.500E-08	ALASM

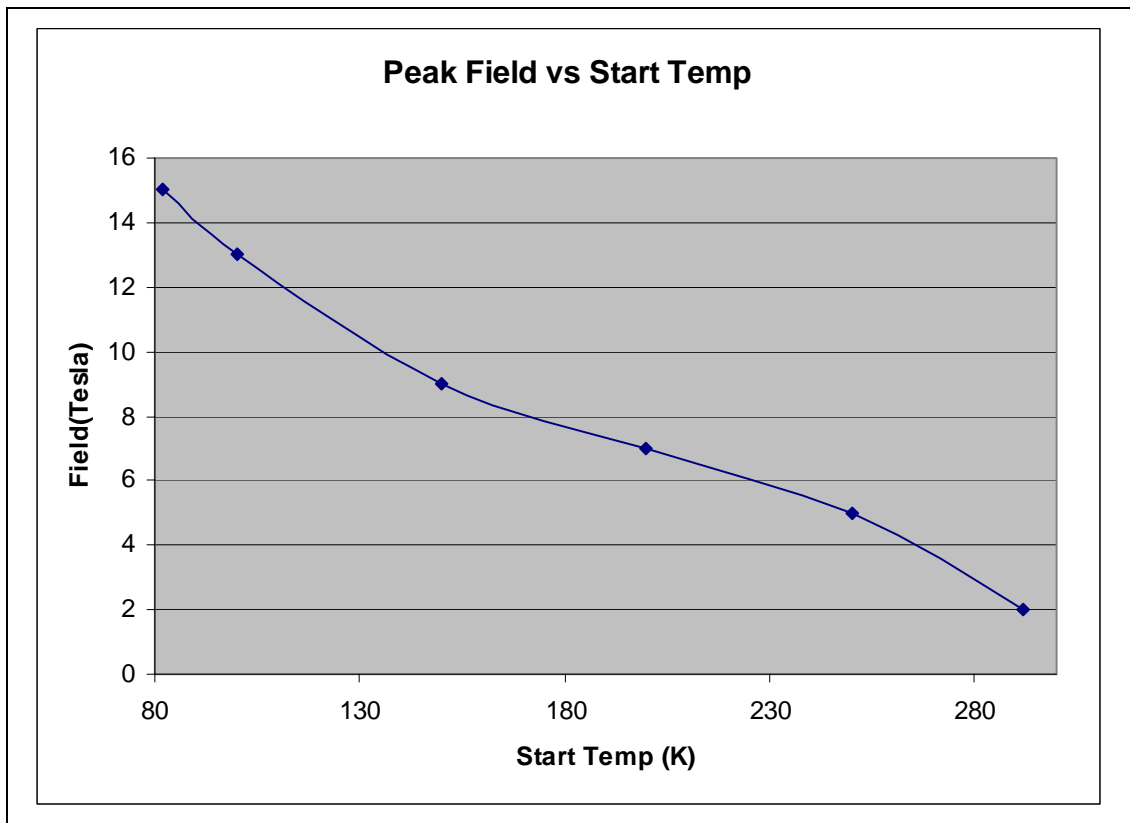
There is 4 by .75 inch 6061T6 bar available in the storage area of the Technology and Engineering Div of PSFC. The run from the 6 by .75 inch buss bar that feeds the split pair magnet is about 10 feet.

6.4 Circuit Simulations with Higher than Nominal Start Temperatures

If the Power Supplies are Voltage Limited to 700 V, the response of the magnet is simply to reach lower fields and modest delta T's

If the power supplies try to follow the current profile, and starts at RT it will overheat the magnet (550K) and the voltage will be too high.

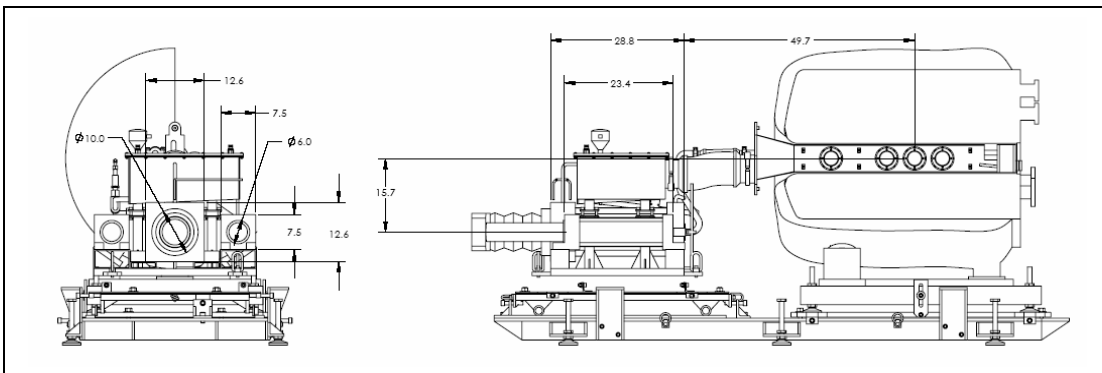
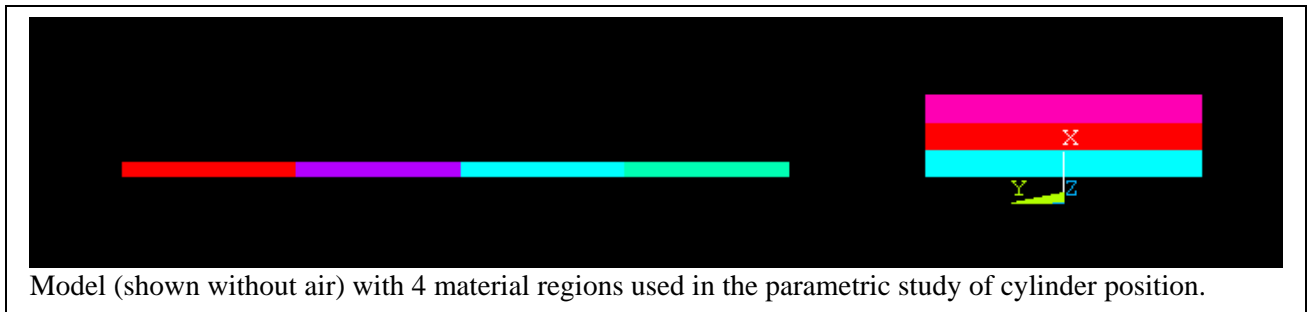
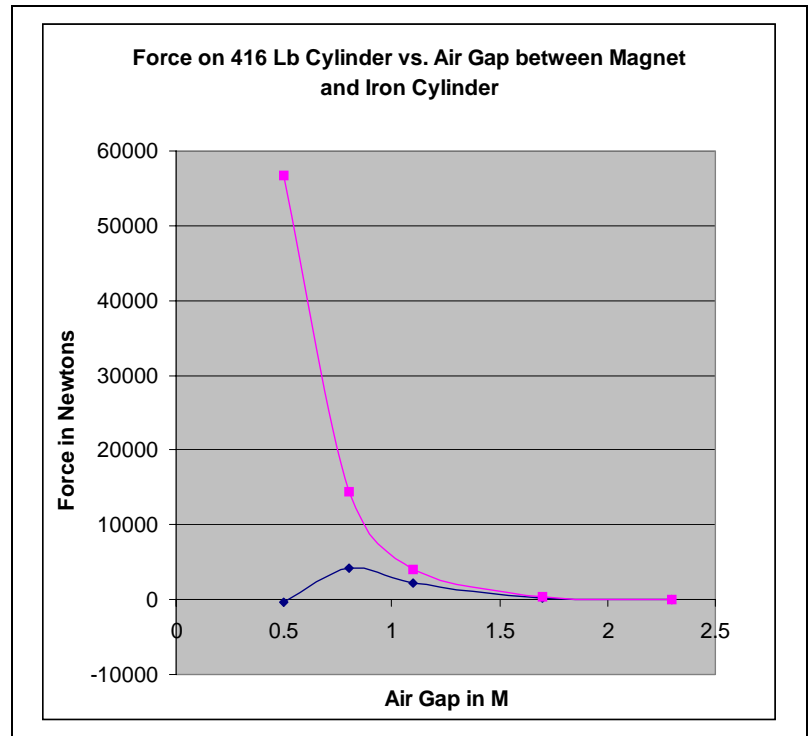




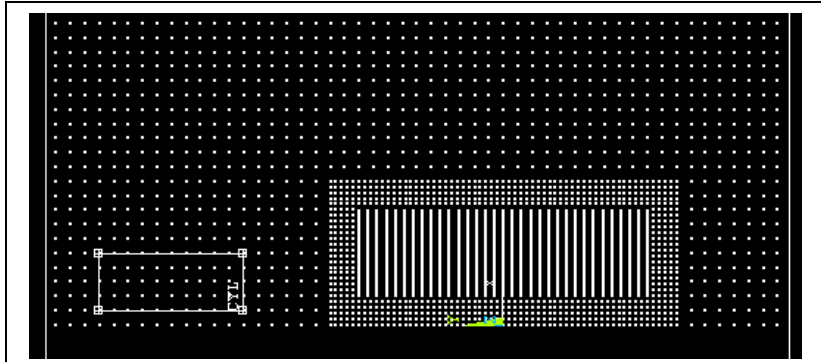
6.5 Magnetic loads on Mercury Jet Components

Material in this section comes from an August 10 2005 email from Van Graves.

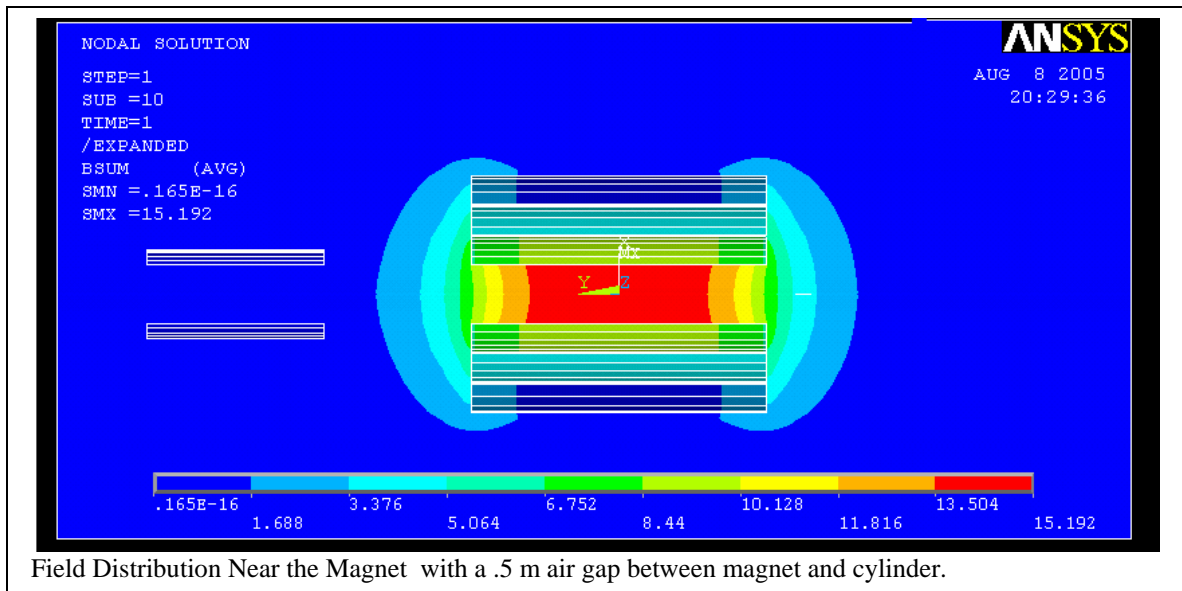
An iron cylinder of .6 m length, an inner radius of .1 m and thickness of .05 m was used to model the inventory of magnetic material in the mercury jet system. Initially, the cylinder edge was placed at .5 m from the end of the magnet or 1 m from the centerline of the magnet. The iron modeled, weighs 416 lbs. The magnetic force developed was 57438N or 12900lbs. This would probably be unacceptable for the magnet cold mass supports even if the cylinder could be adequately supported. The air gap between iron cylinder and magnet was then varied from .5 to 2.3m in steps of .6m by introducing a series of 4 materials modeling the cylinder that were sequentially made either air or iron. The loads are a very strong function of this air gap. When the gap was reduced to .2 meter, the load was .4MN. One solution to the excessive loading on the iron cylinders is to increase the air gap separation between the magnet and mercury injection cylinders to a meter or more. This raises a safety concern. The large loading at a gap of 0.2m, would produce loads that would be impossible to restrain with structures we are contemplating. If these components or other iron components like gas bottles or tools are allowed near the magnet they could easily become disastrous projectiles. If you choose to control the loading via separation, we should consider some extra framing that would guarantee the mercury skid could not be positioned closer than a meter to the magnet.



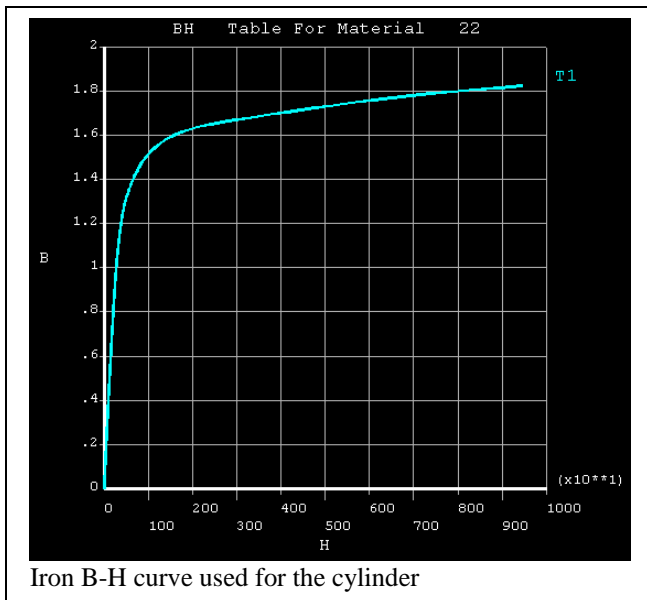
Load calculations require a non-linear magnetic solution and a path integral through the air surrounding the iron. This is done with an ANSYS macro, FOR2D. The path is defined in the graphic user interface.



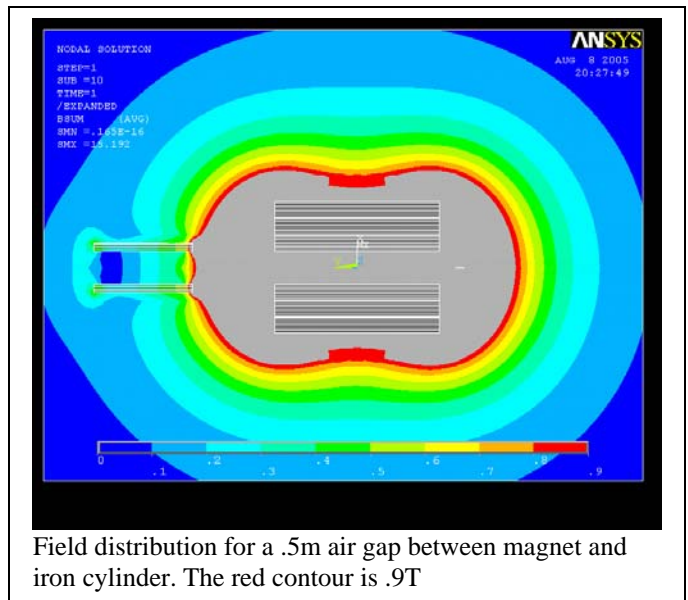
Typical Path used in the FOR2D macro



Field Distribution Near the Magnet with a .5 m air gap between magnet and cylinder.



Iron B-H curve used for the cylinder



Field distribution for a .5m air gap between magnet and iron cylinder. The red contour is .9T

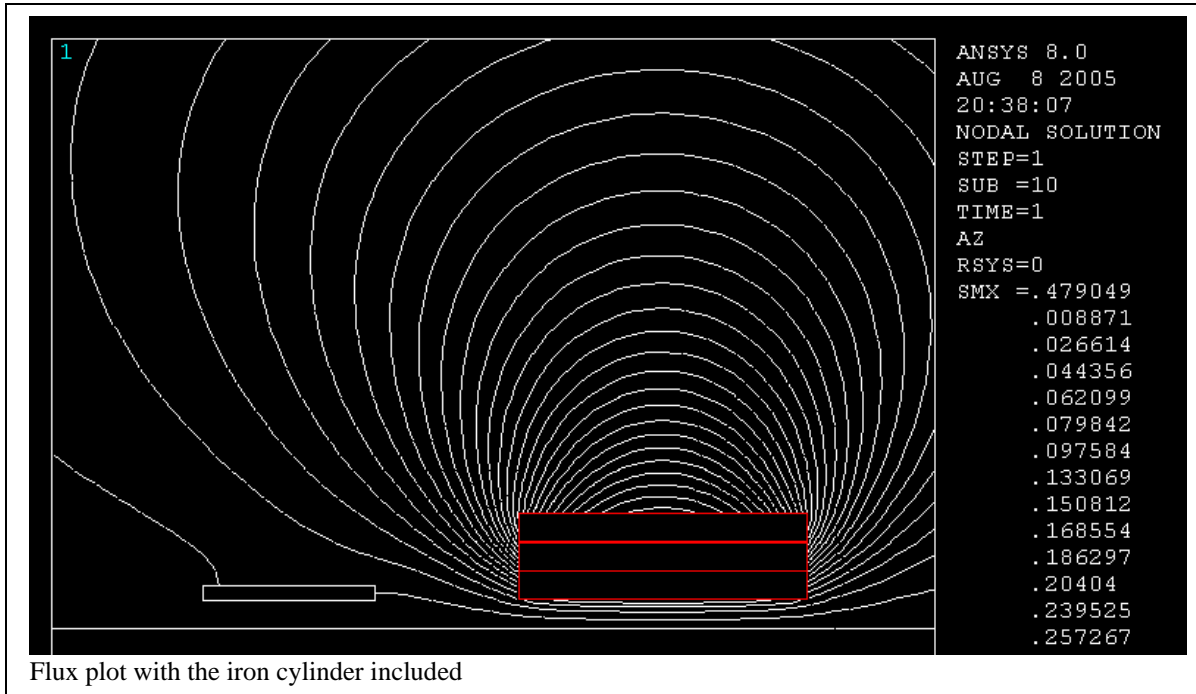
ANSYS output for the FOR2D Macro:

____SUMMARY OF FORCE CALCULATIONS BY MAXWELL STRESS TENSOR____

Force in x-direction = 1138.51061 N/m.

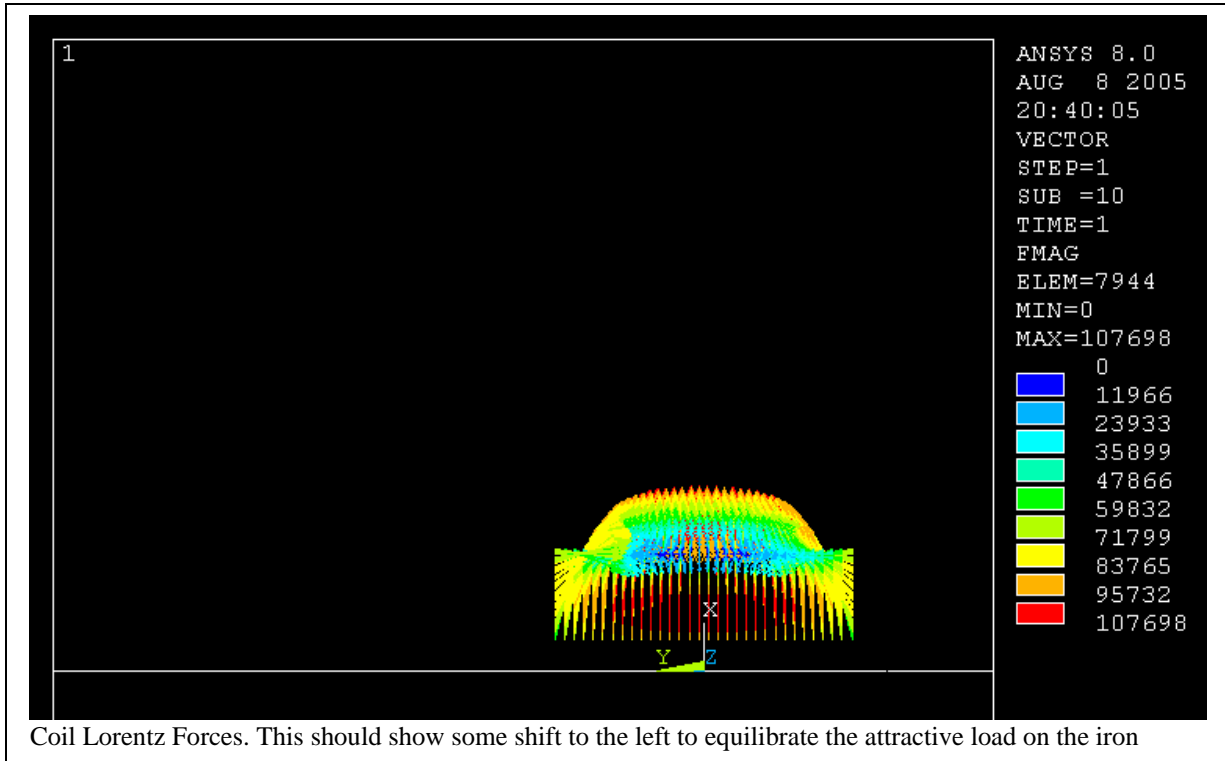
Force in y-direction = -57438.5828 N/m.

In this model, the x direction is radial and the y direction is axial The N/m unit is somewhat of a mystery because the macro should compute the force integrated over 2π and the units should be in N for an axisymmetric model, The N/m unit is probably an error in the macro. But the magnitude of the force would



Flux plot with the iron cylinder included

be similar if multiplied by the cylinder circumference.

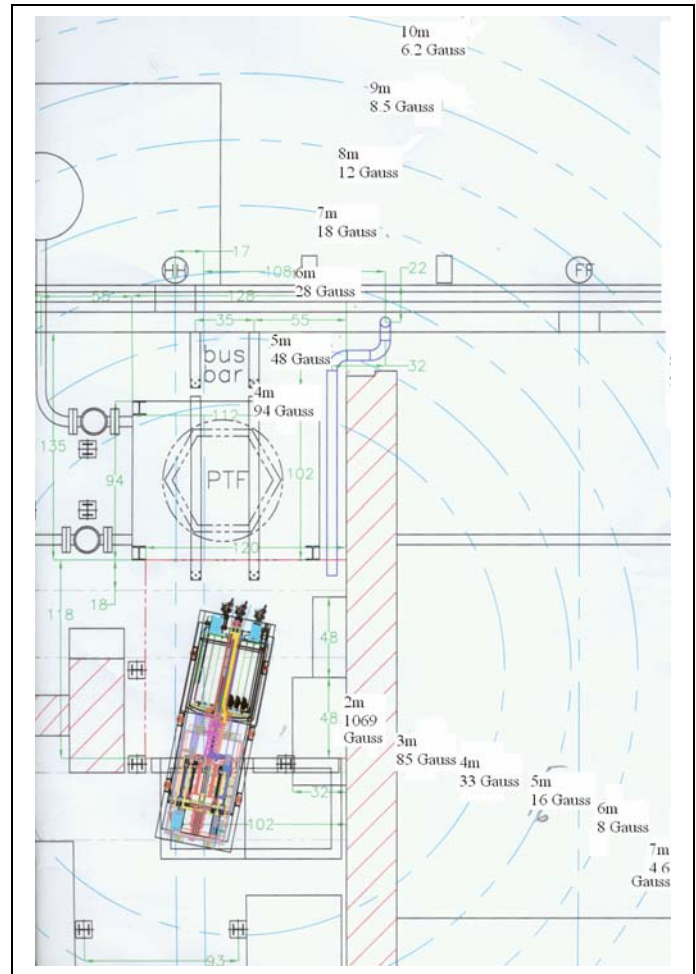
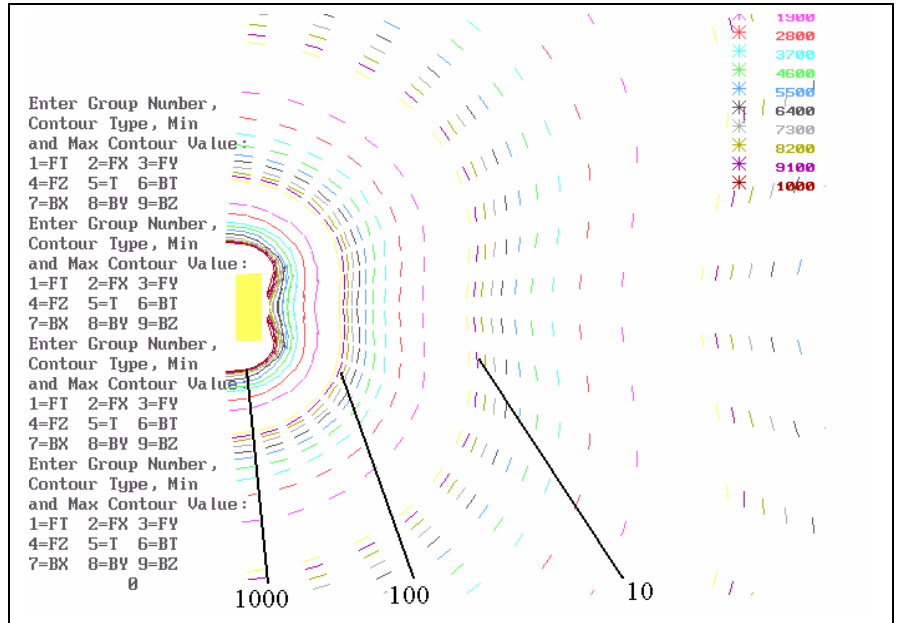


6.6 Stray Field Calculations

```

! BNL Pulsed magnet
axisymmetric analysis
gpla
-.1,.2,1,.5,.25,40
rotz
9,-90
rotx
9,90
repla
beam
zero
doread
beam,18,10
roty
9,-95
rotx
9,90
repla
beam
zero
pfc b
5
1,.15,0,.098,1.0,16,16
2,.2,0,.002,1.0,1,16
3,.25,0,.098,1.0,16,16
4,.3,0,.002,1.0,1,16
5,.35,0,.098,1.0,16,16
1 , .20425 ,0, .2415, .97
,16,16
snal
1
irdt
2
pfcu
5,2,1,1.0
1,4.5,4.5
2,0,0
3,4.5,4.5
4,0,0
5,4.5,0.0
1 , 9.090909,1.0
seal
1
grp mat
1,1
grpt
1,1
snal
1
merge
1,.000001

```




```
redu
egrp
7
ngrp
7
snal
1
seal
1
ccur
1,1,2,3,4,0,0,0,0
!1,1,2,3,4,1,2,3,4
```

```
repla
tmod
!exit
srel
1,5
srel
3,5
srel
5,5
snal
0
ngrp
1
egrp
1
read
beam
```

```
sfield
1
!sfield
!5
stype
7,7
gerase
7
redu
mfor
5,1,2,3,4,0,0,0,0
!1,1,2,3,4,1,2,3,4
srel
2,2
srel
4,2
grpm
2,2
snal
1
bscale
1,10000,10000,10000
repla
18mv
tmsa
18ma,3
```

```
!plce
!popt
!b
!pl
!zero
!zecp
plce
pl
zoin
zoin
!vect
1
seal
0
smat
1,1
snel
1,1
snel
0,0
clear
plot
1
cont
0,6,0,10
cont
0,6,10,100
cont
0,6,100,1000
cont
0,6,1000,10000
!vect
2
exit
```

7.0 Coil Stress Analysis

The coil stress has been analyzed using ANSYS. Fields and forces are computed using Elliptic integrals in a code external to ANSYS. The model is axisymmetric. The figures show a 3D representation from a symmetry expansion.

Table 5.1 Coil Build used in the Stress Analysis

Seg#	r	z	dr	dz	nx	ny
1	.15	0	.098	1.0	16	16
chan	.2	0	.002	1.0	1	16
2	.25	0	.098	1.0	16	16
chan	.3	0	.002	1.0	1	16
3	.35	0	.098	1.0	16	16

For Fusion magnets the inner skin of the solenoid is allowed to reach the yield - Treating this stress as a bending stress with a $1.5 \cdot S_m$ allowable with S_m based on $2/3$ Yield. The allowables are discussed in more detail in section 4.

Table 5.2 Stress Summary for 1/4 Hard Copper Specified by Everson/Tesla

Location	Load Case	Peak Von Mises Stress	Allowable	Factor of Safety @100K
ID Segment 1	3 Coils fully energized No Gaps	166(Tresca)	270 MPa at 100K 1/4 Hard Copper w/ID CW	1.6
ID Segment 2	3 Coils fully energized w/ Gaps	107	207MPa RT ~250 MPa at 100K, 1/4 Hard Copper	2.33
ID Segment 3	3 Coils fully energized w/ Gaps	40	207MPa RT ~250 MPa at 100K, 1/4 Hard Copper	6.25

If the highly cold-worked copper is chosen for the winding, the conductor allowable near the inside radius of the coil would be 365MPa. The max stress in the three segment coil is 166 MPa. With this stress level, it is expected that half hard copper could be used, simplifying the winding process. The three segment coil has three operational modes, two of which are structurally significant. The full performance configuration is limiting in terms of hoop stress and equivalent stress. It also has some radial stresses that will have to be mitigated with parting planes at the segment boundaries, or within the winding.

In the initial operating mode the outer coil segment is not energized. This induces some differential Lorentz forces and differential temperatures, that cause shear stresses between segments.

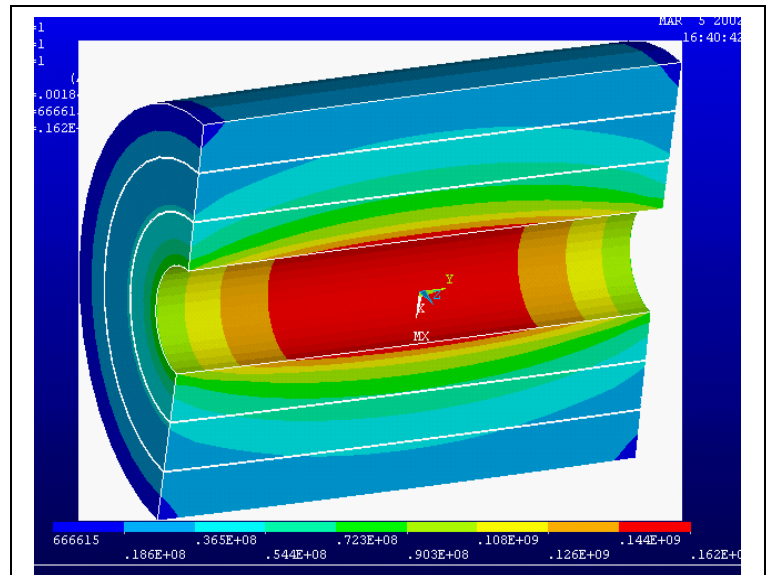


Figure 7.1 Hoop Stress, all coil segments fully energized. The Von Mises stress plot is similar with a peak of 165 MPa, Tresca is 166 MPa.

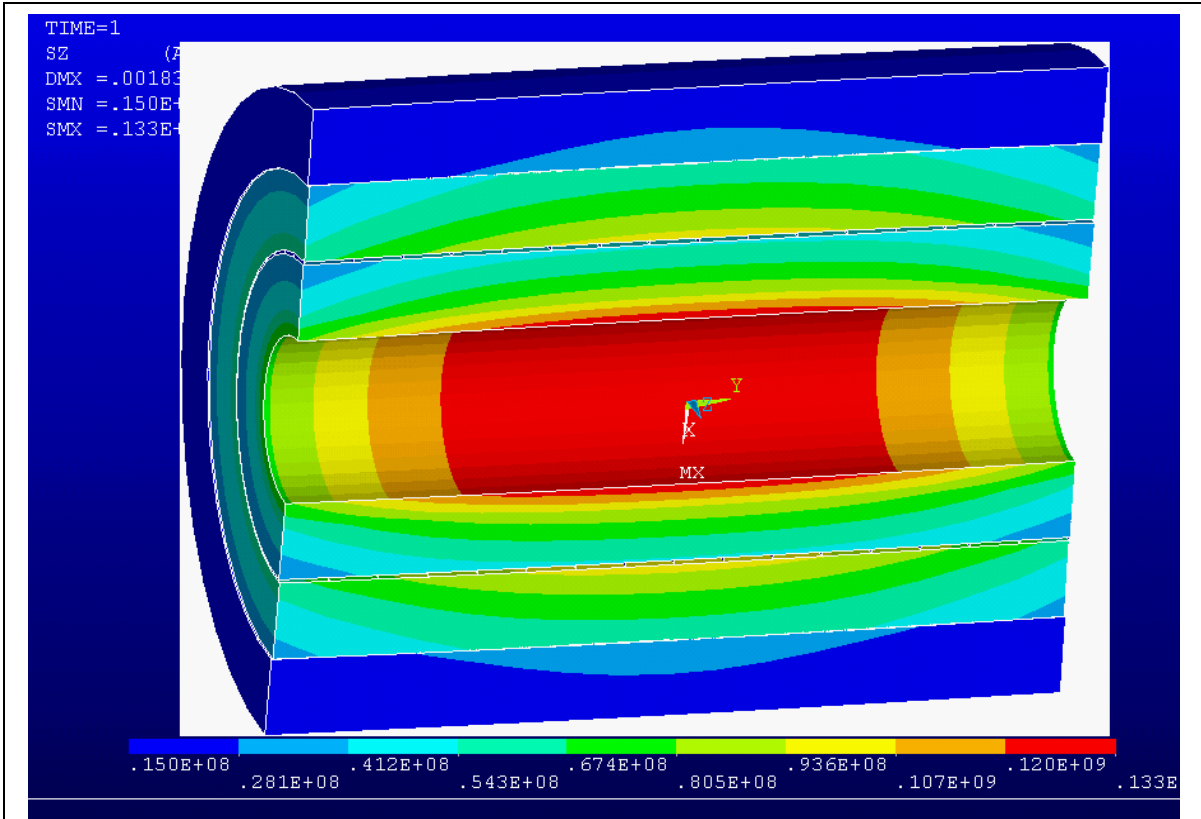


Figure 7.2 Re-run with gap elements the hoop stress went down to 133 MPa

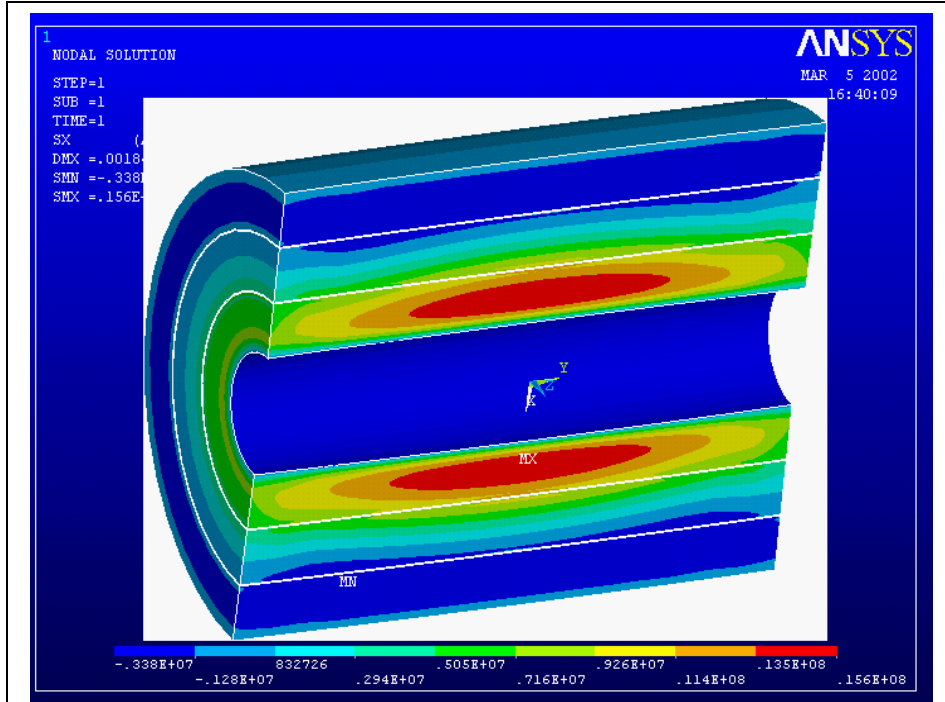


Figure 7.3 Radial tension stress, all coils fully energized. There is about an MPa of tension at the boundary between the first and second module. To avoid damage to the channel ligaments, a parting plane will be incorporated in the channel detail. This needs to occur in the ligament to retain thermal connection with the coolant in the channel.

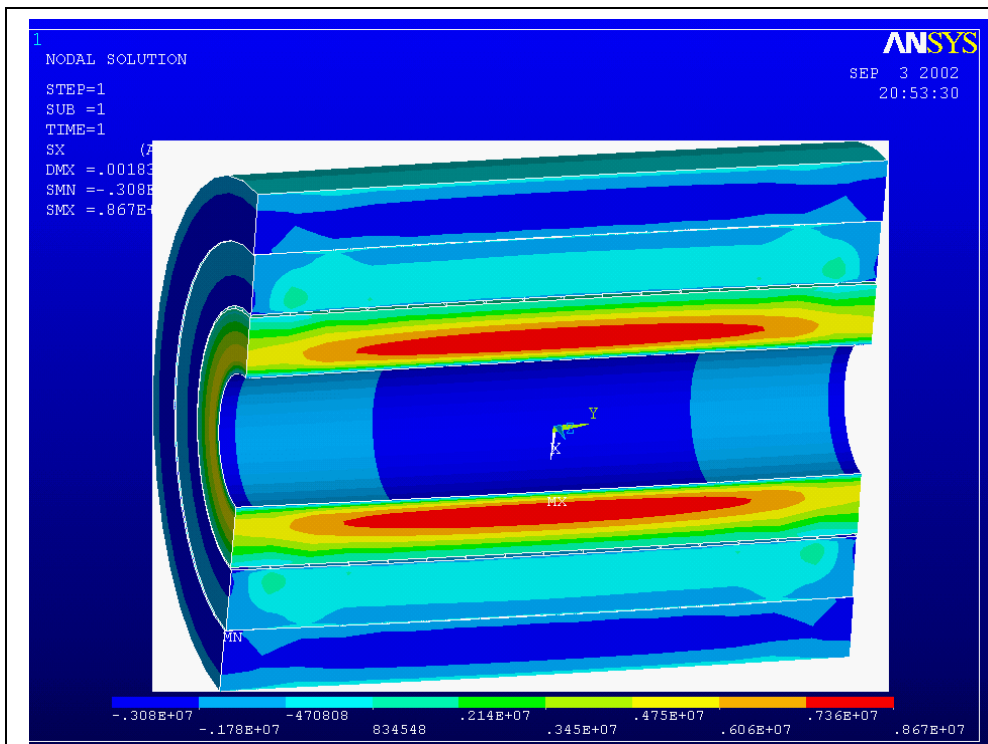


Figure 7.4 Radial Tension in the winding of the first segment is reduced by about A half with gap elements modeling the interactions of the three coil Segments.

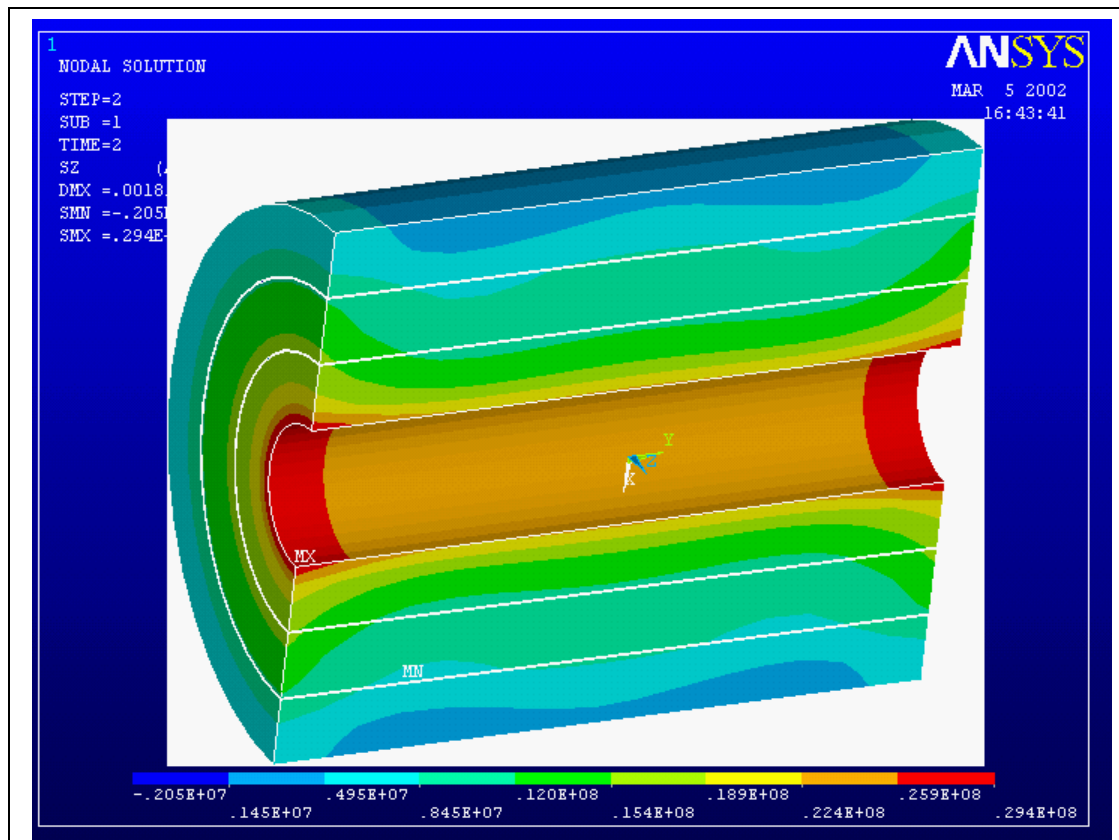


Figure 7.5 Hoop Stress with only the inner two segments energized.

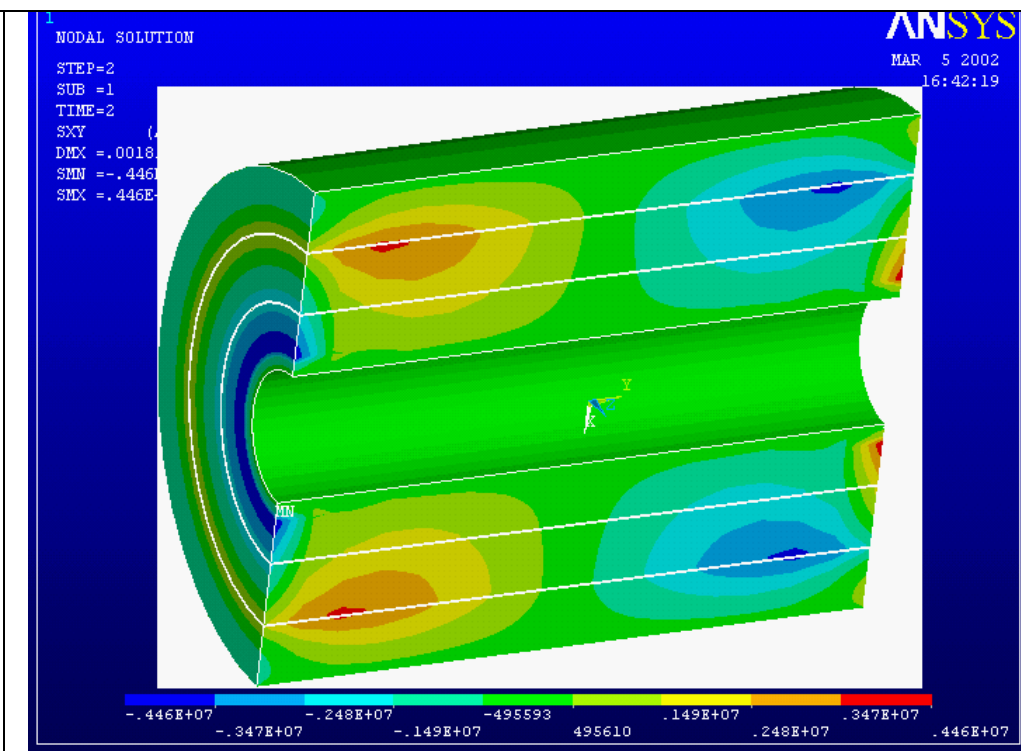


Figure 7.6 Smear radial-axial shear stress with the inner two segments energized. This is a peak at the interface between the second and third modules. It must be carried across the thin ligaments between the channels, or relieved via a slip plane.

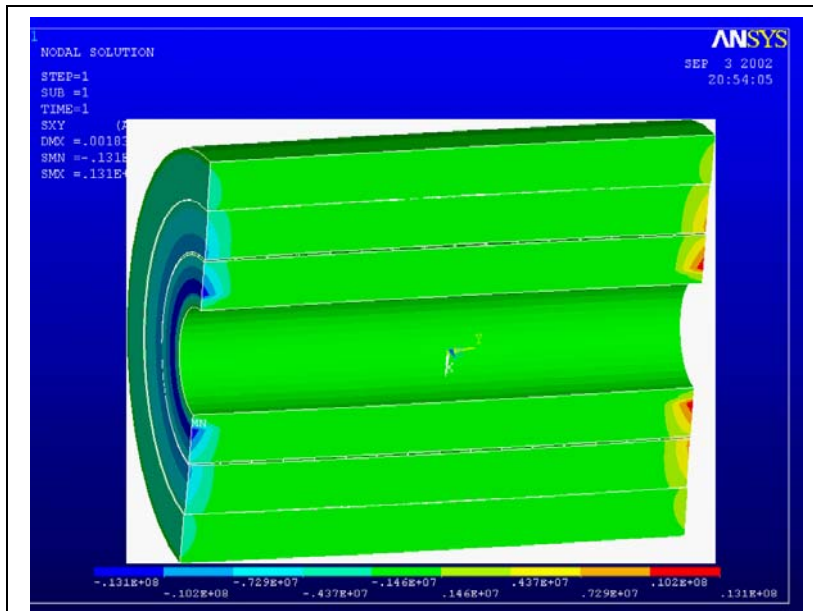


Figure 7.7 With gaps modeling the interfaces between segments, only inner segment shears remain.

Optional Purchase of only the Inner Two Segments

It may be desirable to build only the initial two inner segments and add the outer segment at a later time. The coil was analyzed with the outer segment removed, and the same current density in the other two segments. The max stress for this case is 85.3 MPa, which is a bit more than with the outer segment in place, but less than for the fully energized three segment coil. In all cases the stresses are lower than the expected allowable for the conductor. It is expected that the degree of cold work can be relaxed from the full hard condition. The final choice of the degree of cold work for the

conductor will be determined during detailed design.

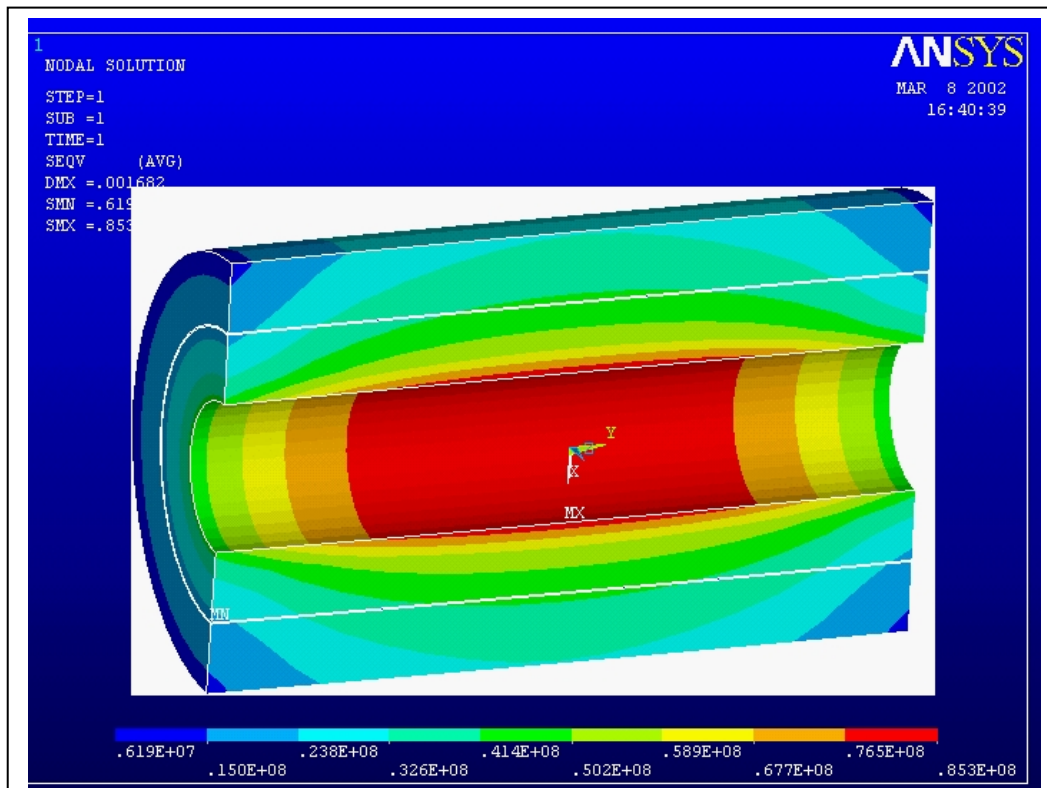
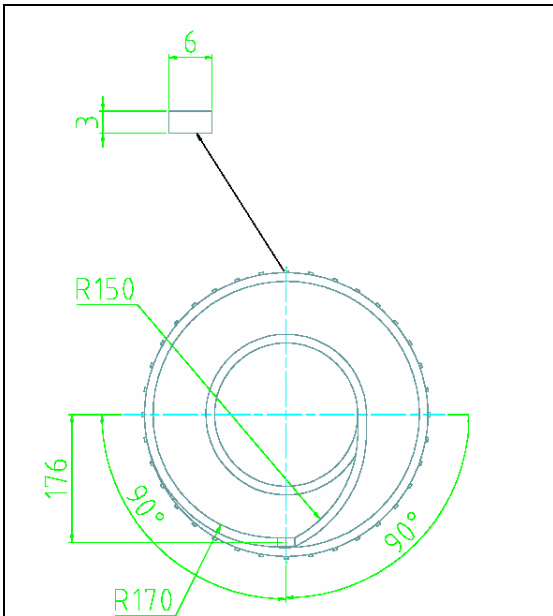
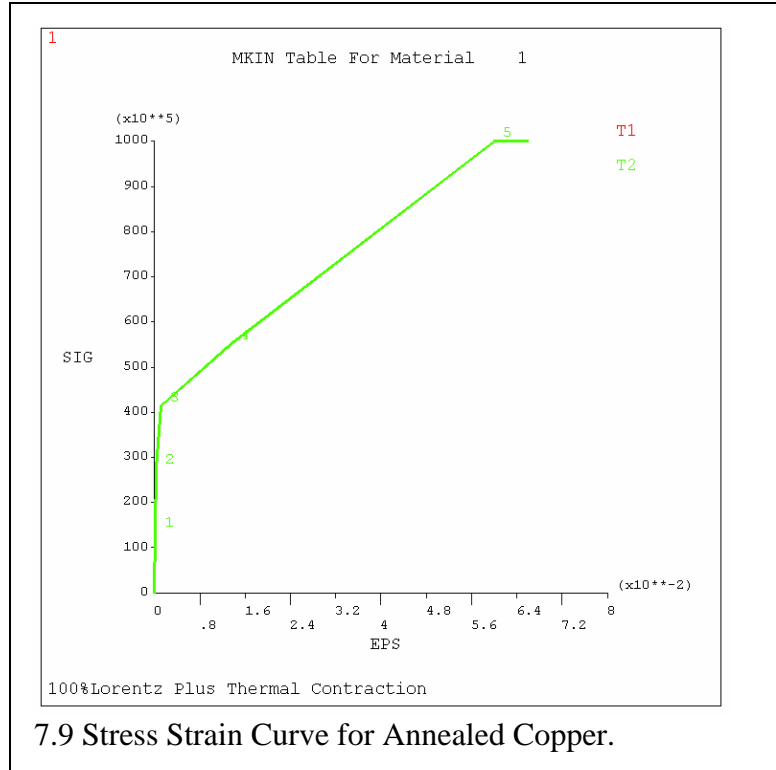


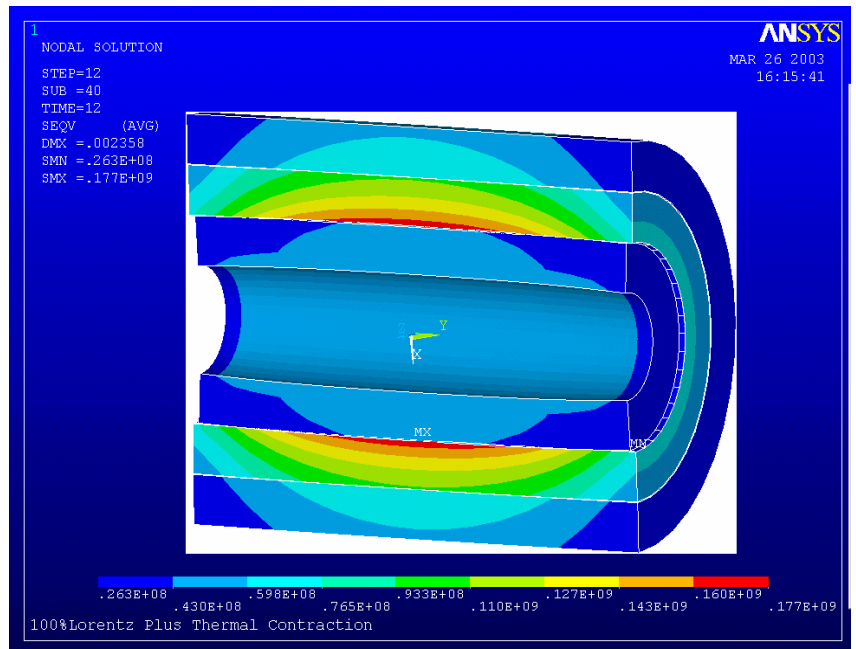
Figure 7.8

Elastic Plastic Analysis

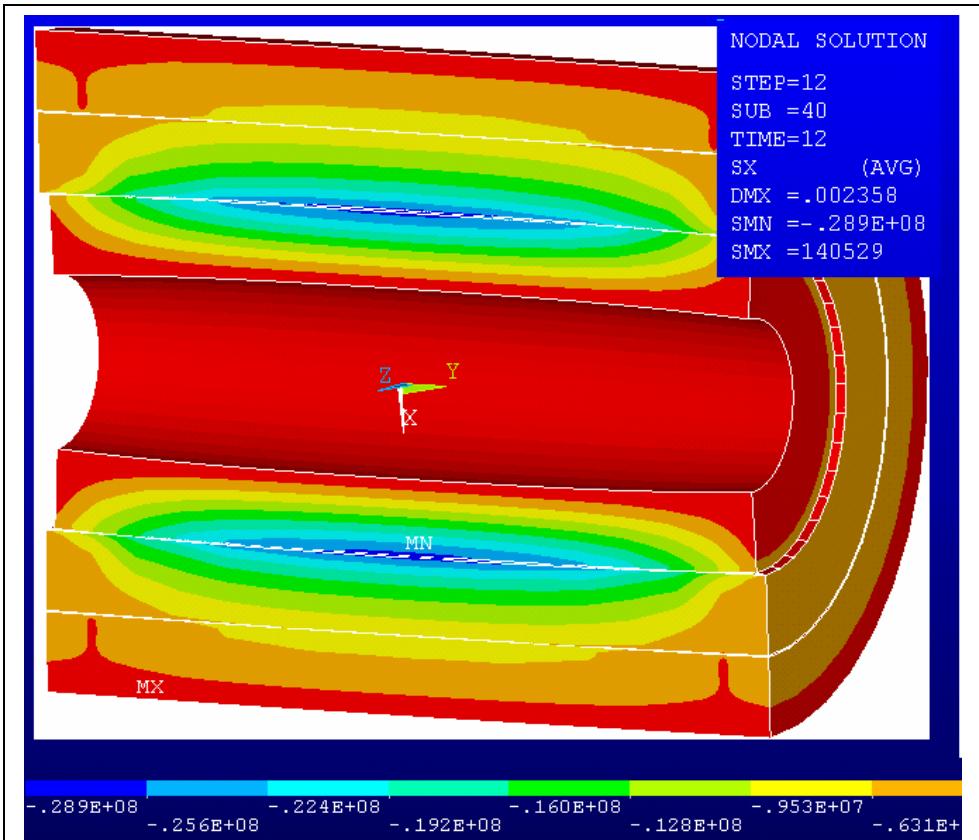
There is a concern that the tight inner radius of the coil will be difficult to wind with full hard copper. Fiberglass tape and Kapton tape cutting/crushing are possible. Relaxing the cold work specification, while still meeting the normal allowables is one approach, but may not ease the winding process sufficiently. A “fall-back” approach is to wind the inner module with annealed copper and allow it to yield and “lean” against the second coil segment. This was analyzed using elastic-plastic properties in the inner segment. The results show acceptable stress levels in the outer segments, but the strains in the inner turns may exceed the capabilities of the bonded insulation.



7.10 Radial compressive stress between segments one and two must be supported by the ribs that form the cooling channels. The baseline design has almost no load across this gap. If annealed copper is used in the inner segment, compressive loads across this space could be significant near in the ribs. The radial stress multiplier is $200 \cdot \pi / (24 \cdot 6) = 4.36$



7.11 Von Mises Stress. The Inner Segment is annealed copper, and the middle segment of the coil supports most of the Lorentz Forces. The Von Mises stress at the ID of segment number two is 177 MPa, which would be acceptable, but there is a multiplier on the radial compressive stress that will increase the local Von Mises under the ribs significantly.



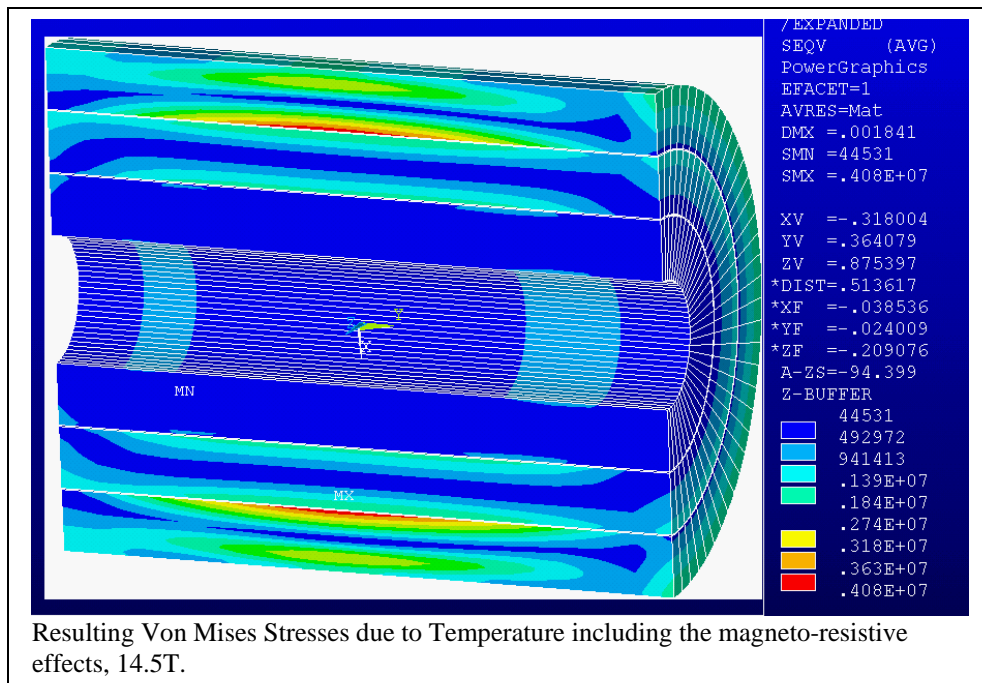
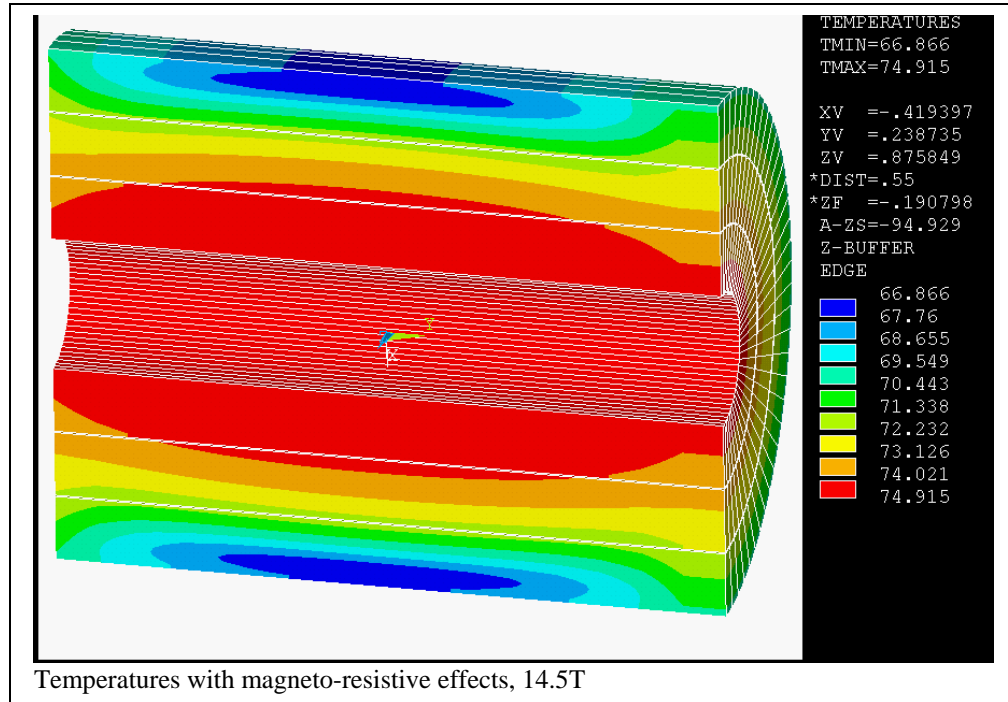
7.12 With an annealed inner segment, The radial multiplier due to the rib projected area is 4.36, and the compressive stress under a rib is $28.9 \times 4.36 = 126$ MPa, acceptable by itself, but unacceptable when added to the hoop tension in the second coil segment.



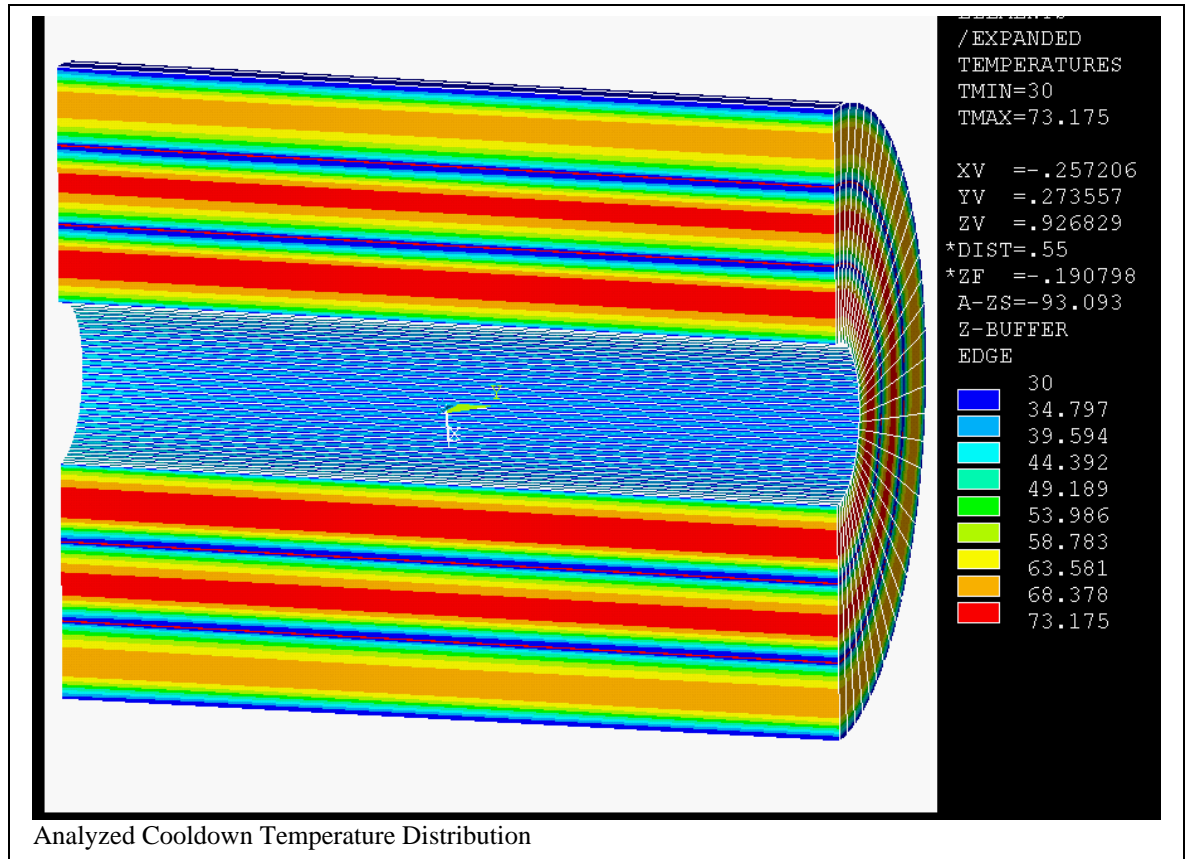
Tabs to maintain torsional registration

8.0 Coil Thermal Stresses.

Temperatures were calculated for the 15 sec ramp-up, and 2 sec flat top and a 7 sec ramp down. The NIST Kohler plot and fitted equation was used for the magneto-resistance. In my calculations, the temperatures were low compared with Bob's. This may have been a result of using the average data from the NIST Kohler plot that the upper bound. To make some progress on the stress calculations the time scale was stretched to come closer to Bob's temperature distribution. The stresses from this analysis are small, less than 5 MPa, The cooldown stresses will require more work. These are discussed on the next page.

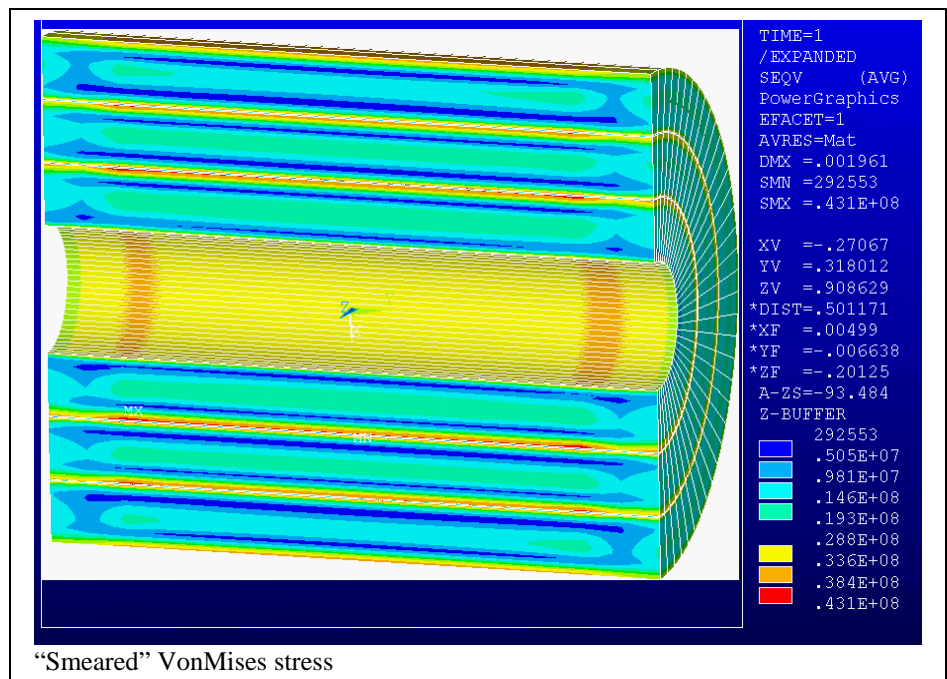


Cooldown Stresses



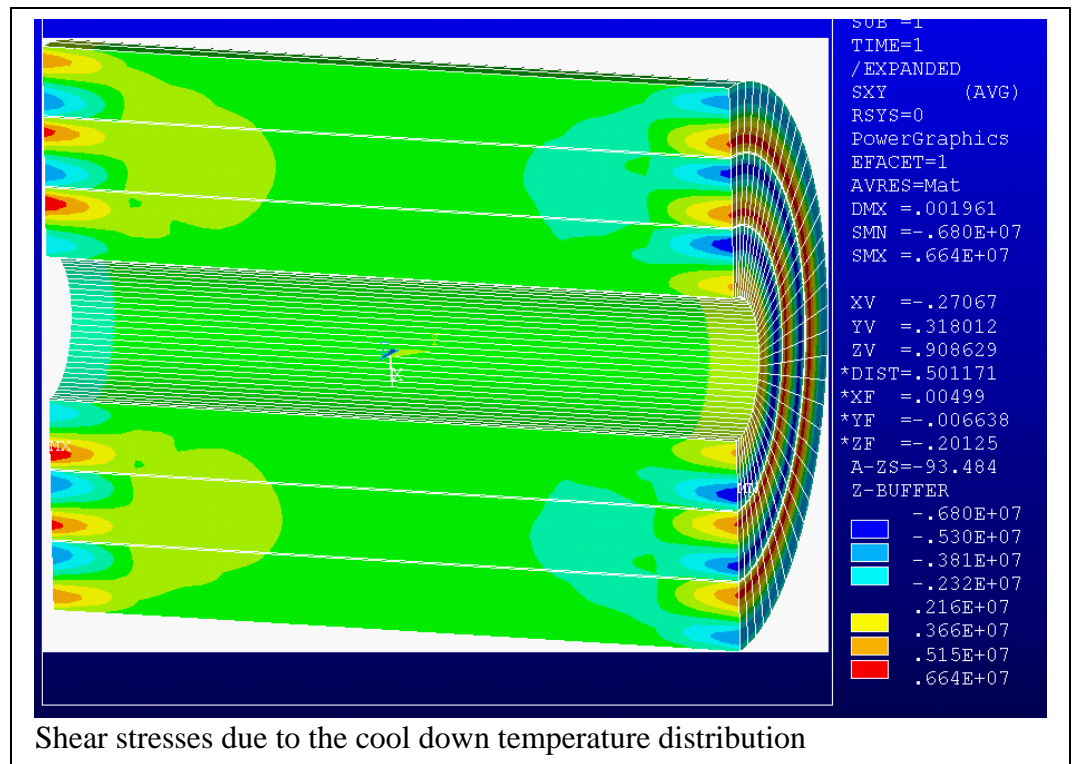
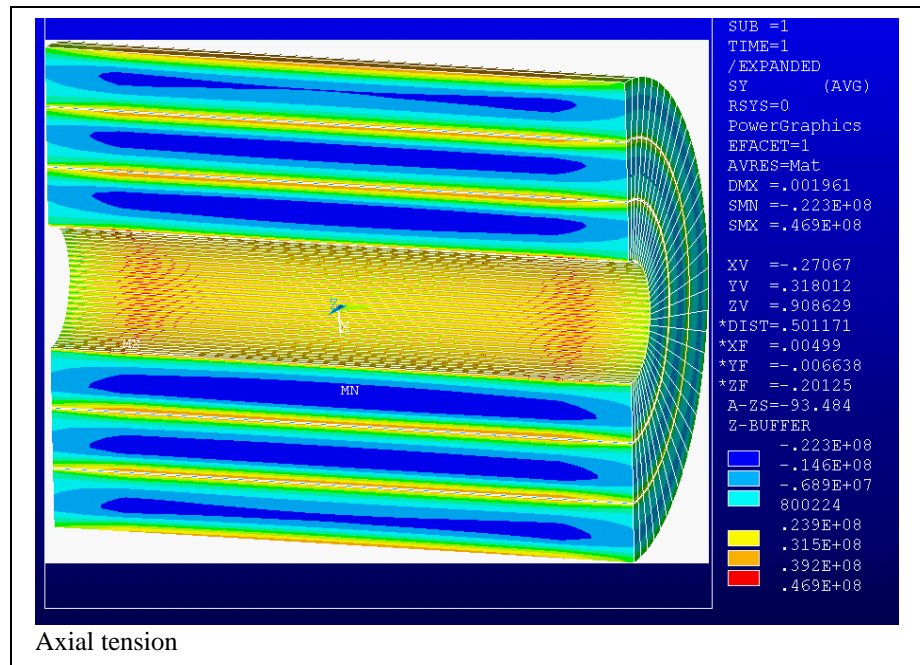
The channels were held at 30K and the temperature distribution was obtained by averaging nodal temperatures with the final temp distribution from the heat-up calculations. This is not rigorous, and is essentially assumed, but it is representative of temperature distribution, and will serve to provide guidance for further analysis and design.

The Von Mises stress is relatively modest, at 43MPa



The axial tension near the channels is approaching 50 MPa, beyond the design capacity of epoxy bonded systems. Some provision will have to be made to either throttle the cooling gas to limit the channel temperature or design to allow the bond failure.

The shear stresses that peak at 7MPa are within the usual allowables for insulation systems, for which design allowables are in the range of 15 to 30 MPa (with no aid from compression)



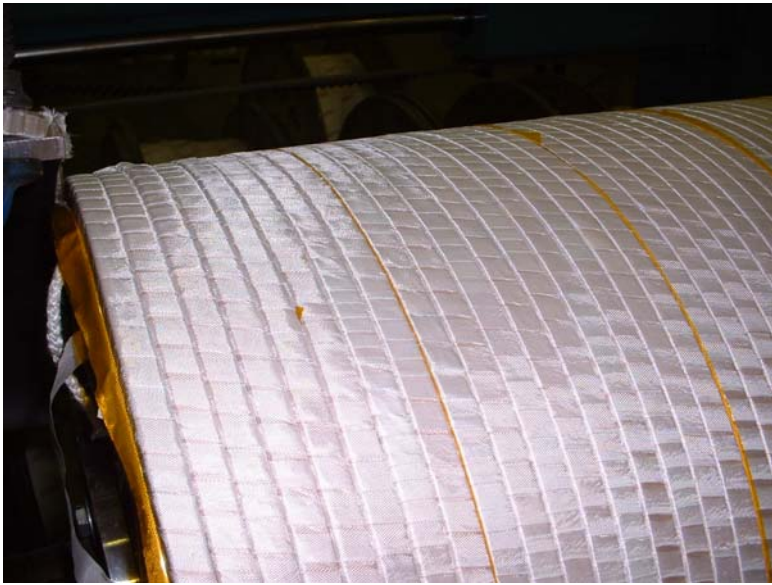
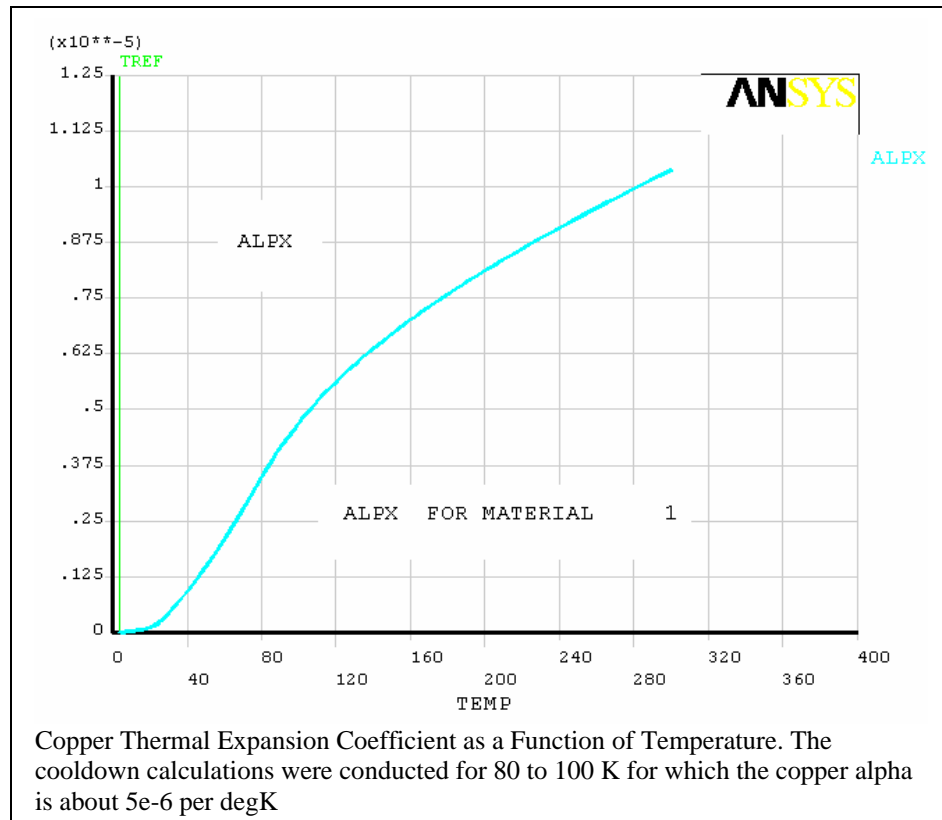
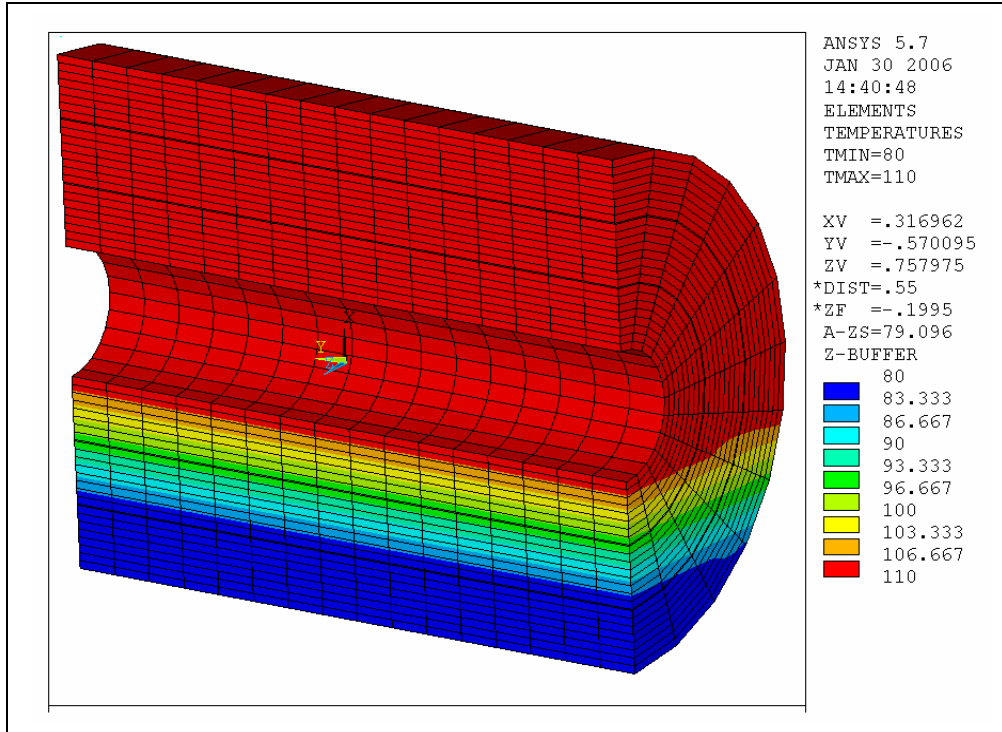


Photo taken by Dave Rakos at Everson 09-08-04. Kapton layer spaced at every eighth turn relieves axial tension in the layers near the cooling channels.



Kapton arc sections inserted between every eighth turn on those layers that face the cooling channels

Cooldown Stress, Global Thermal Differential.



1

NODAL SOLUTION

STEP=5

SUB =1

TIME=5

SY (AVG)

RSYS=0

DMX = .707E-04

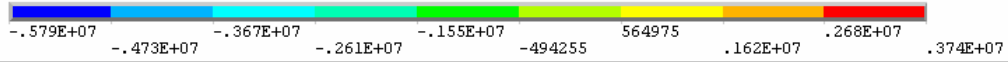
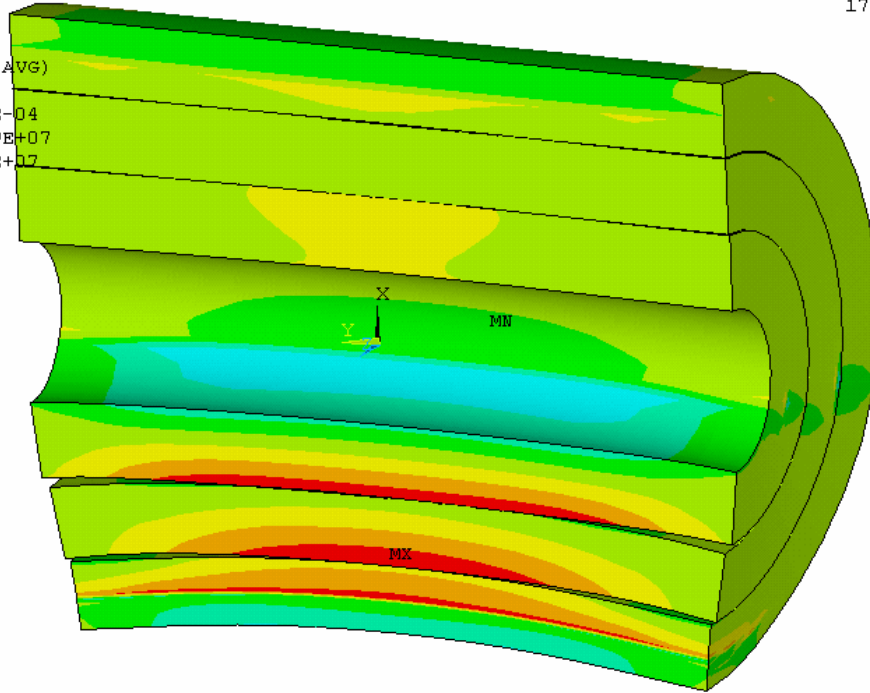
SMN = -.579E+07

SMX = .374E+07

ANSYS

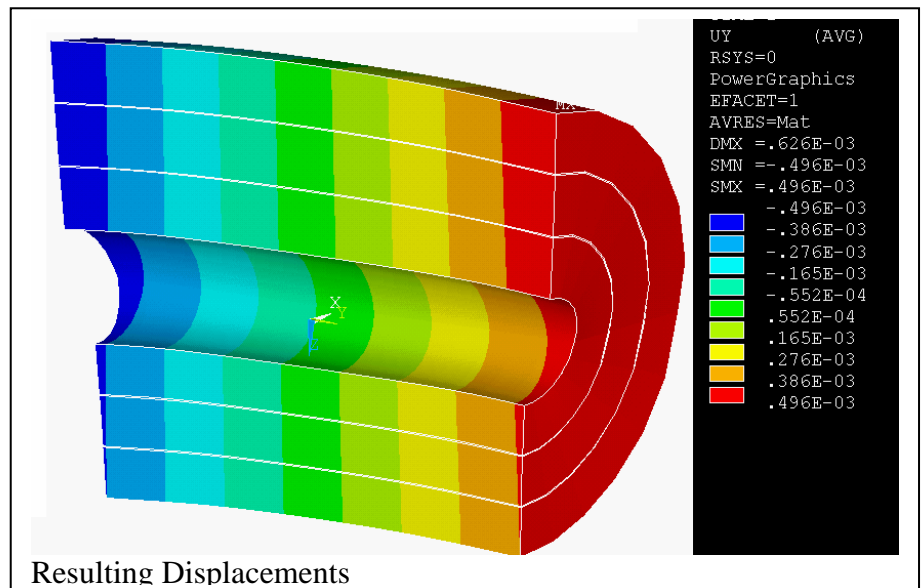
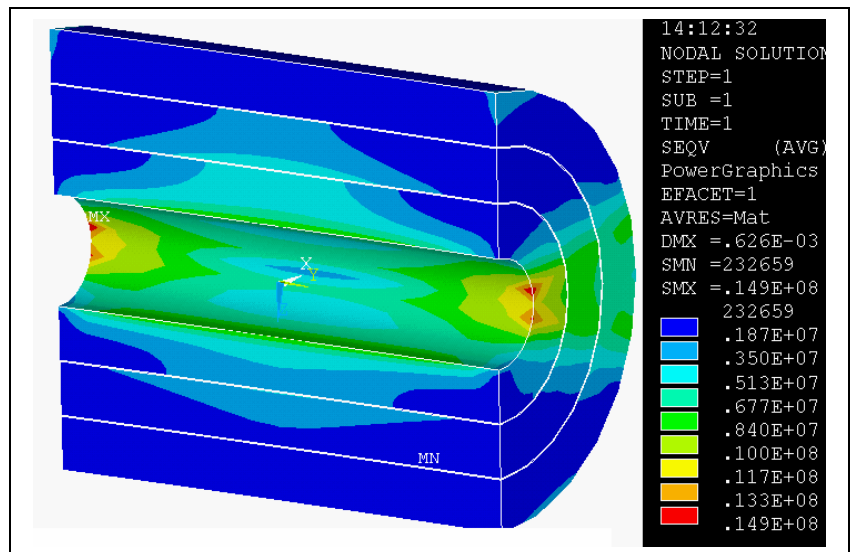
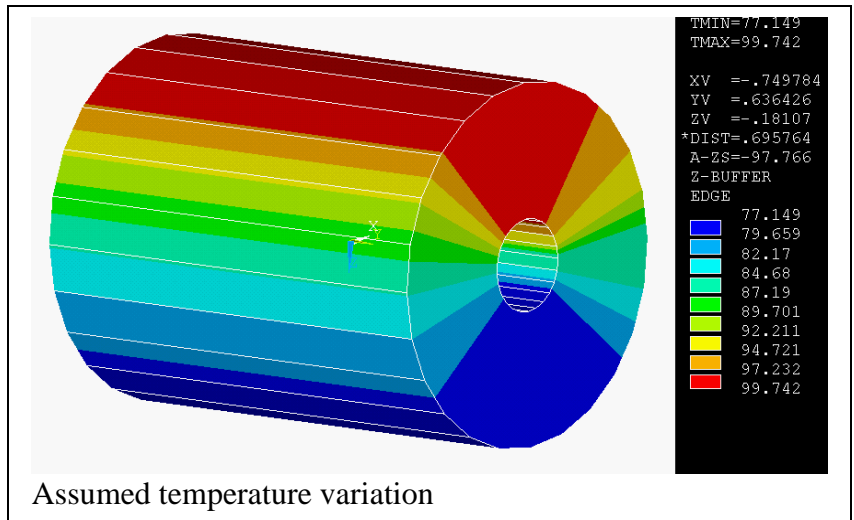
JAN 30 2006

17:25:10



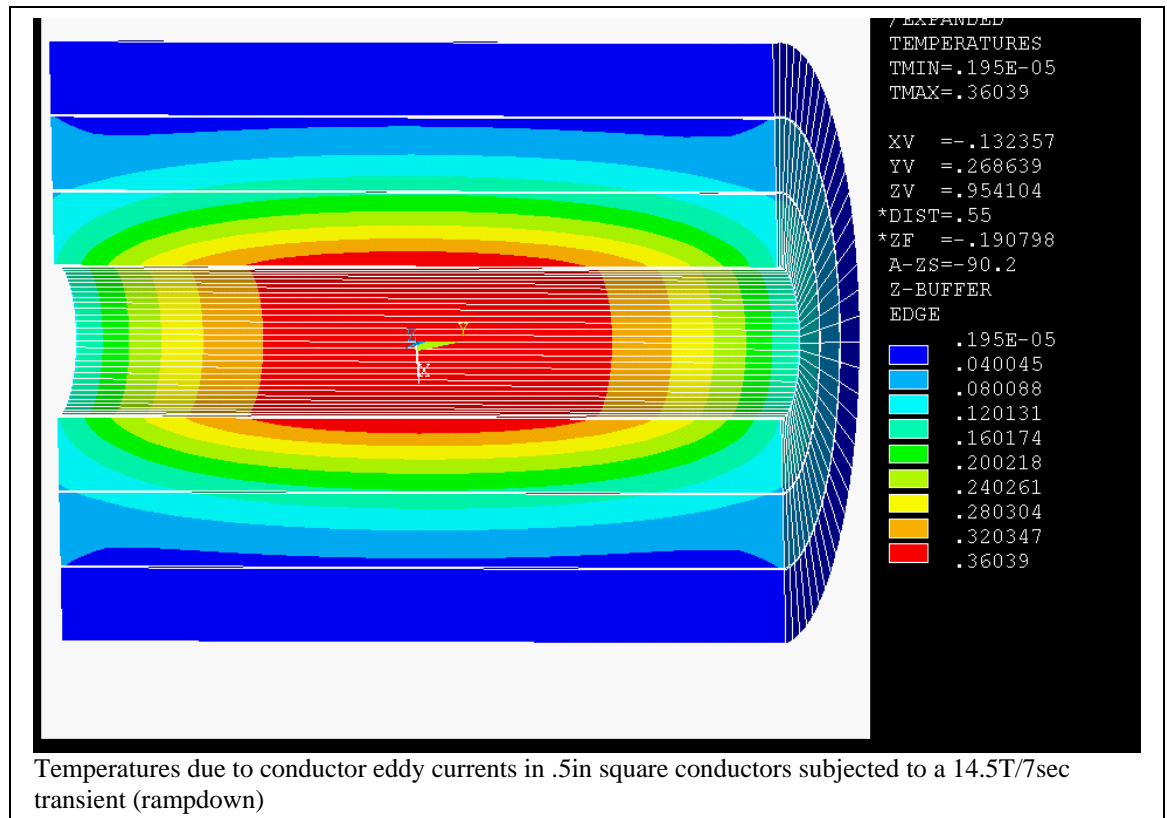
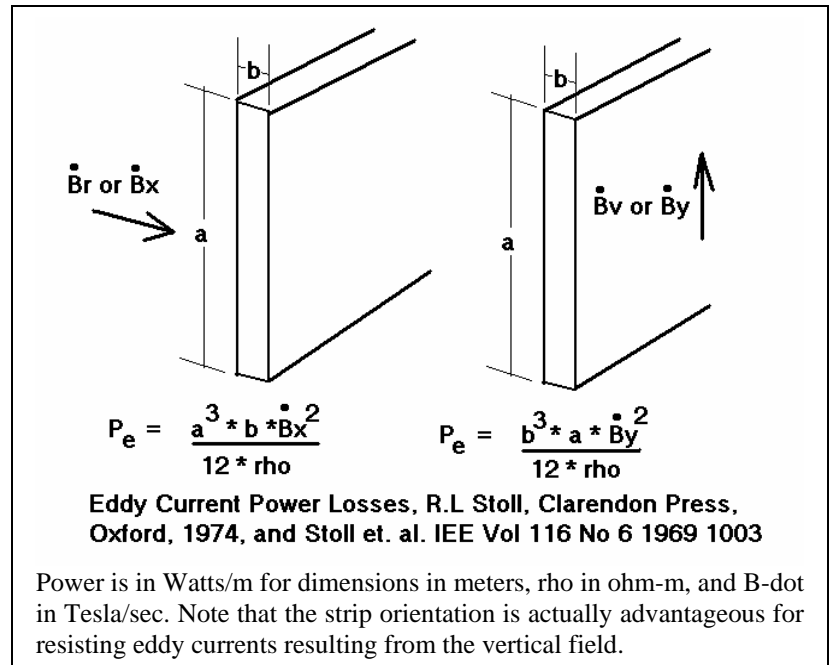
Another Thermal Distribution

If there is stratification of He gas or if LN2 floods the bottom of the cryostat there could be a significant thermal differential between top and bottom of the coil. A 77 to 100 K variation is assumed. The resulting 15 MPa stress is Acceptable



Eddy Current Temperatures – A Non-Problem

Transient fields induce eddy currents in the conductors as well as in the cryostat plates. This has been investigated for a strip wound solenoid used for FIRE, a fusion experiment. Eddy current heating has been evaluated for the BNL pulsed magnet using the same procedure. The conductor cross section is much lower for the BNL conductor, and the eddy current heating is less than one degree. – a non-problem.

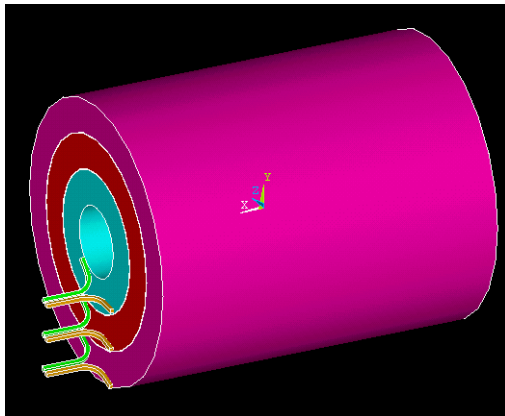


9.0 Break-Outs, Leads, and Penetrations

The choice of modular design favors duplicating the break-out and lead design for all three segments, even though two of the segments are connected in series. The current leads from each coil segment can be accessed from outside the cryostat, and can be re-wired without removal of the cover. In the case where 3 segments are purchased initially, the re-wiring to accommodate the power supply changes is done without opening the cryostat. This feature comes at the expense of the heat leak of the extra terminals connected to room temperature. The break-out concept structurally connects the inner layer break-out with the outer layer break-out. The leads are closely coupled to cancel the net loads on the lead conductors. This is similar to a coax, but because the conductors are side-by-side, and not concentric, there is a small torque. To achieve the interconnection of the leads, they cross the face of the winding. An alternate is to wind an end pancake. The mandrel design may dictate one over the other. At this time only one lead arrangement has been modeled. Bending stresses for combined thermal and Lorentz force loading of 200 MPa can be expected for the cantilevered leads. The analysis model has minimal fiberglass wrap and more extensive interconnection of the leads, and support at the winding pack end will be needed. The lead must penetrate the end plate of the cryostat, and because of the differential thermal motions, and flexure of the end cap under pressure loading, a bellows extension is used. This also has the effect of lengthening the cold length of the lead, reducing the heat gain.

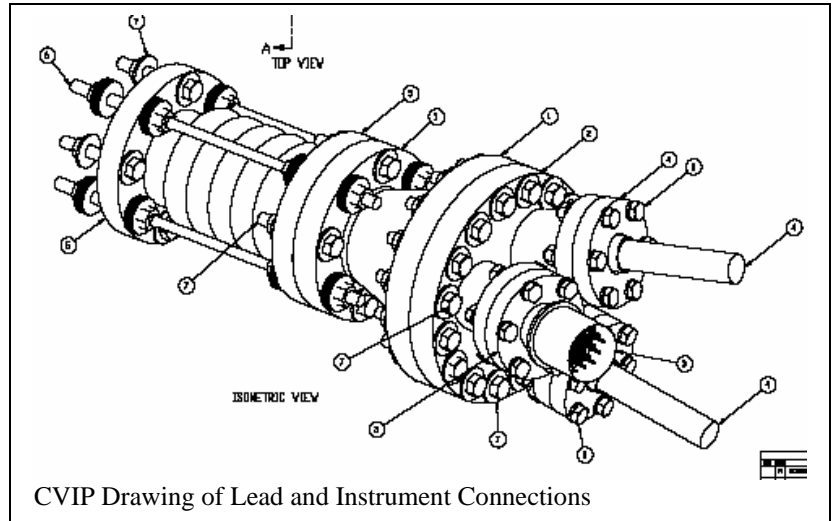
At this time, the analysis includes the Lorentz forces on the leads, and thermal stresses due to the temperature gradient. The displacement induced stress due hoop expansion has not been modeled.

The magnetic model includes the coil as a “smeared” volume, with the detailed break-out models added. The fields and forces in the leads are calculated with 7200 amps in the leads, and the appropriate solenoid end field solution is applied.

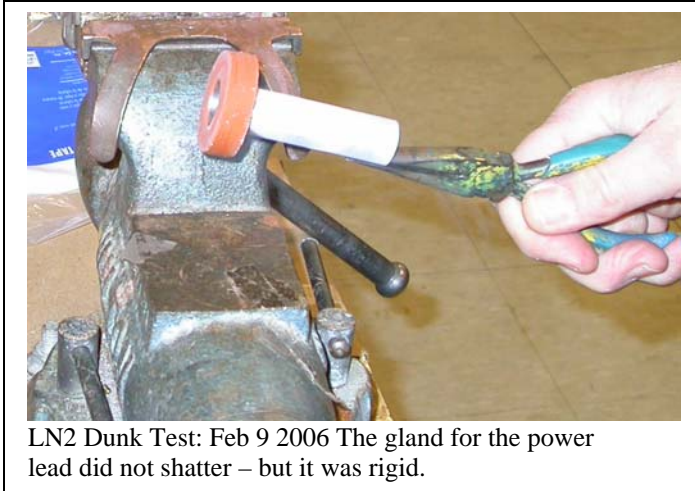


The electromagnetic model.

The fields and forces in the leads are calculated with 7200 amps in the leads, and the appropriate solenoid end field solution is results.



CVIP Drawing of Lead and Instrument Connections



During March tests last year there was a small leak in one of the terminals. Another “washer” of silicon rubber was added. This was compressed at RT and did not leak during March 2007 tests

Joint and Conflat Leakage, March 2007 Tests

- Problems/Observations:
- “Large” LN2 Leaks were evident - but Mercury tests were not impacted.
- One major leak was from the LN2 feed closure valve that was “found” equipment”
- Cobbled together cryogenic system does not represent the CERN magnet deliverables.
- Terminal seals leaked.
- “Quick Fill” effected conflat seal. leaks were detected from the center terminal assembly.



Platenut and Belleville stacks added to the packing nut/gland assembly

A Belleville stack is being added to each of the bolts in the terminal assembly. This should help hold compression on the packing/gland as the assembly cools.



Leaks at the inner conflate of the terminal assembly appear to have been the result of the bellows tie rods not tightening the conflat



Packing nut screw torque:

14 Nm = $14 \times 0.2248 \times 39.37 = 124$ in-lbs or about 10 ft-lbs

Bolt torque = $0.2 \times F \times D$ where F is the bolt tensile force and D is the nominal diameter

I used 1/4-28 screws which are close in size to the metric equivalent Kirk purchased with the platenuts

The bolt force is then $124 / 0.2 / .25 = 2480$ lbs. each

The stress area of the bolts is .03 in² so the stress is: $2480 / .03 = 83$ ksi or 570 MPa.

These must be good screws.

In the photo there are 8 tack welds. They look no more than 1/8 inch square or .015 in².

316 RT weld ultimate is 482 MPa. The tack welds are loaded in shear so the shear ultimate is 240 MPa or 34.8 ksi .

So their capacity is around $34800 \times 8 \times .125^2 = 4350$ lbs. The ultimate and thus the capacity just about doubles at cryogenic temperatures.

For 6 screws this is 725 lbs per screw and the recommended torque is $0.2 \times F \times D = .2 \times 725 \times .25 = 36$ in-lbs = 3 ft-lbs = 4.1 Nm.

I think I was putting on about 5 to 8 ft-lbs with my uncalibrated hand/arm at MIT.

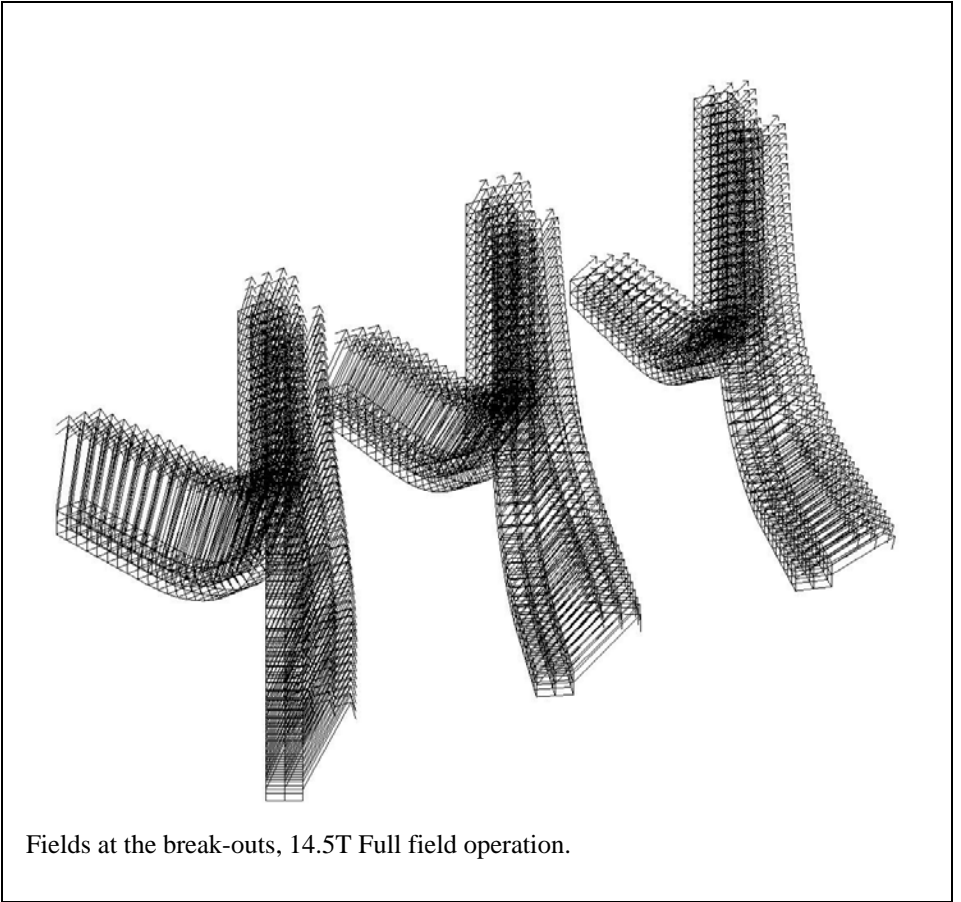
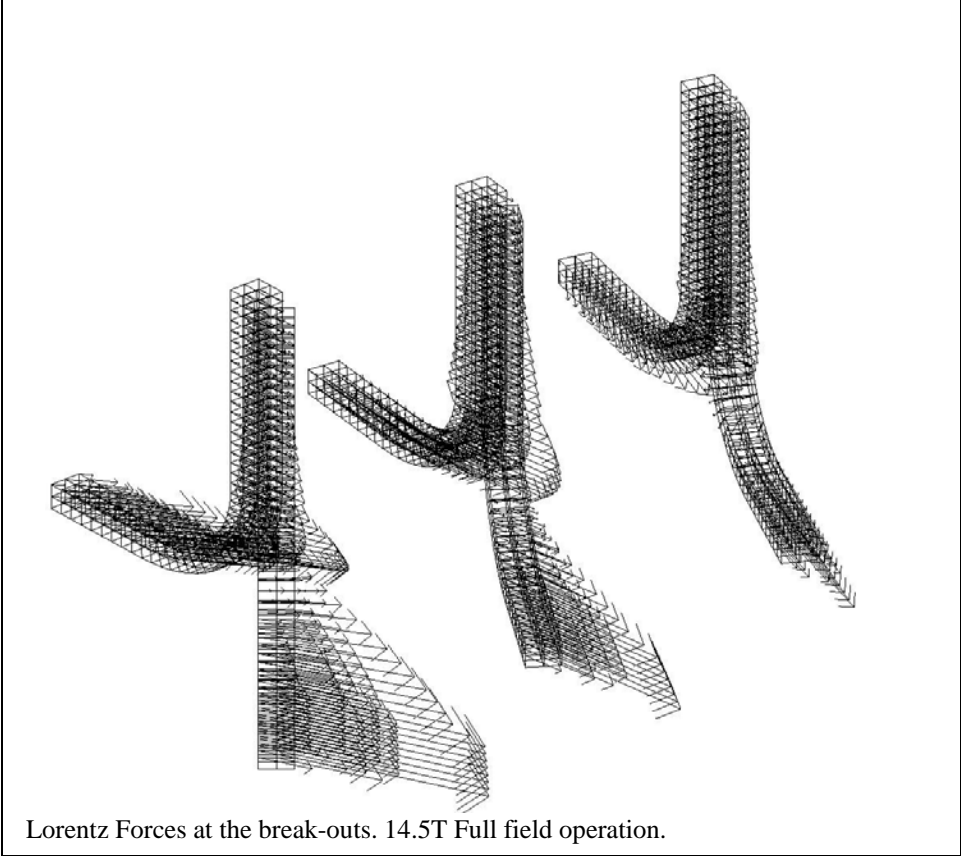
If applied at RT, I think 14 Nm will break the tack welds. I would recommend no more than half this at RT.- Even this may be close to the ultimate. The material properties improve when things get cold. The torque could be increased when the magnet is cold, but I would not risk doubling the torque – you don't want a problem after the magnet is in the tunnel.

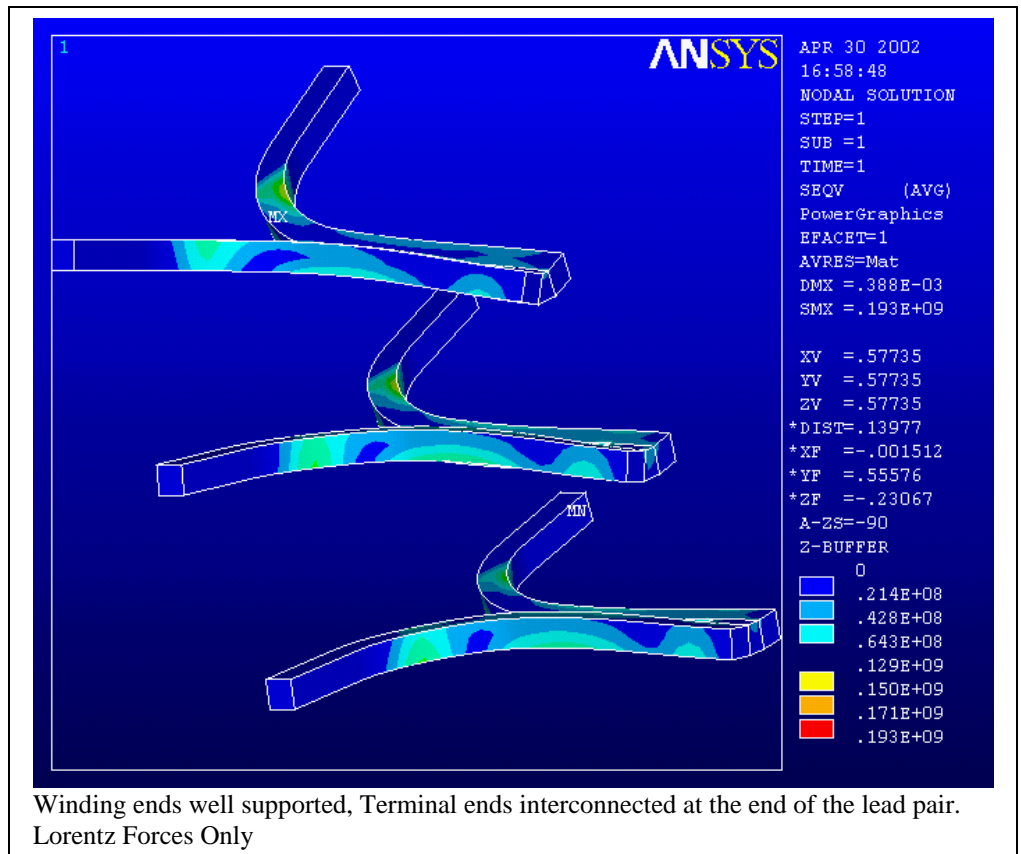
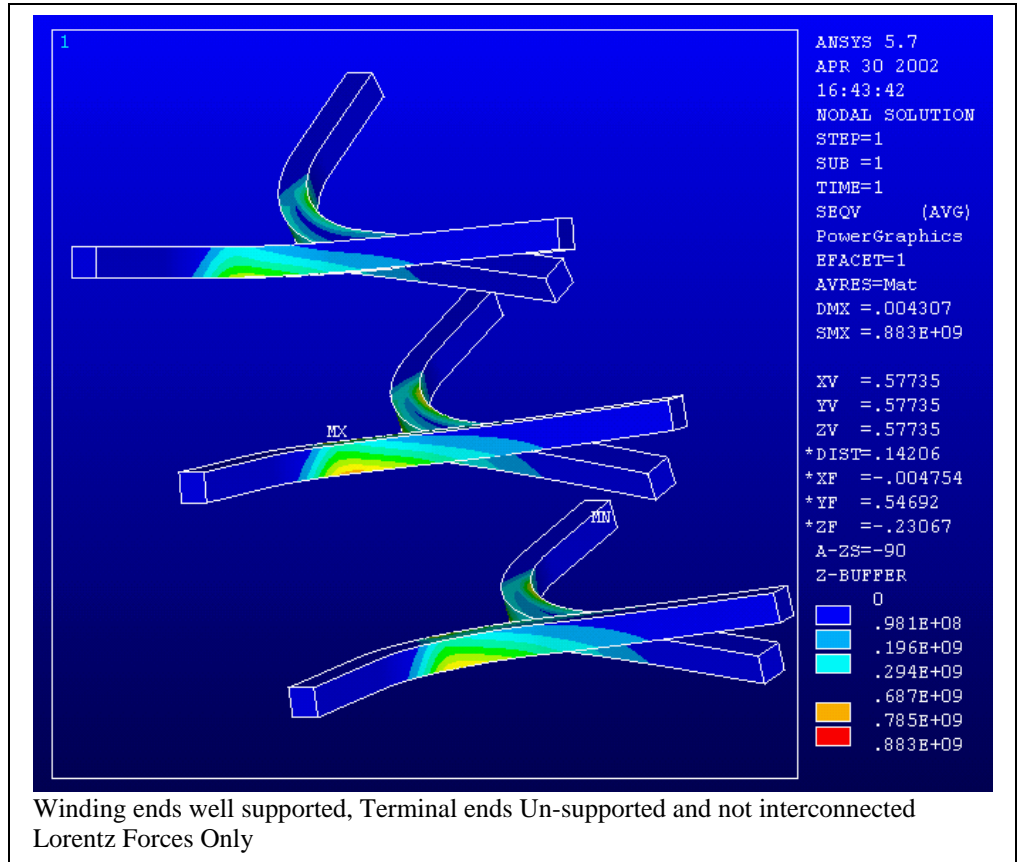
When the magnet was at CERN the glands leaked when flooded with LN₂. The silicon rubber gland was replaced with Teflon material used for valve packing material in cryogenic valves
Here are a couple of web pages for the material:

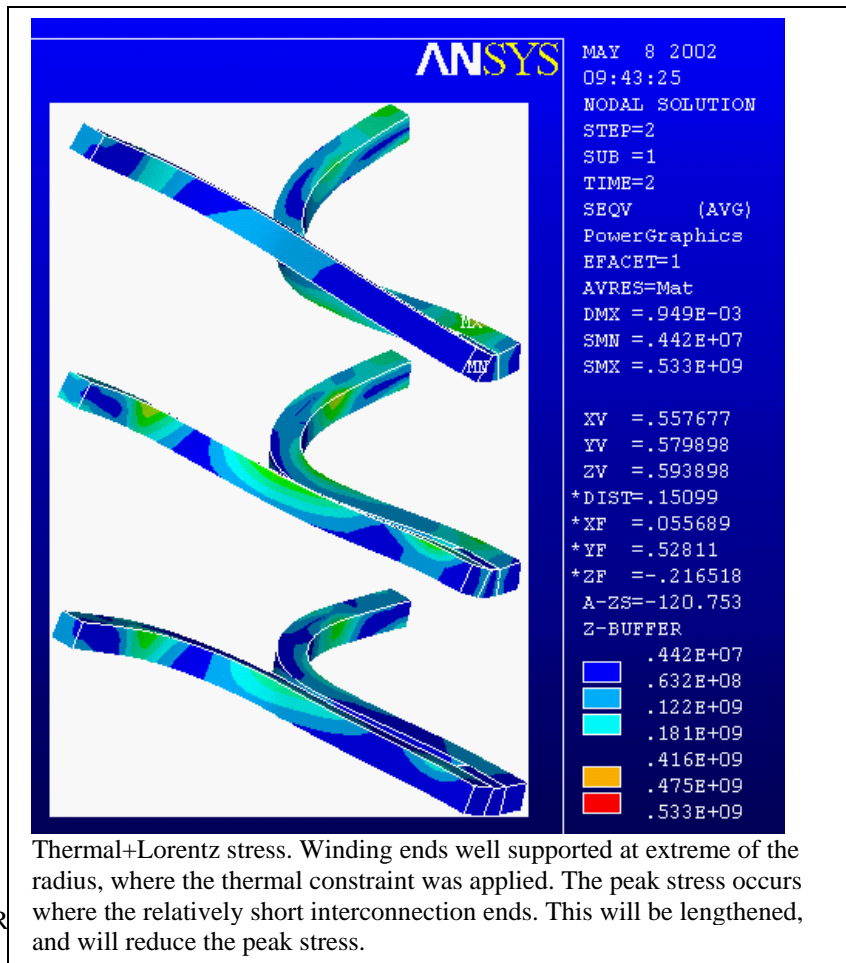
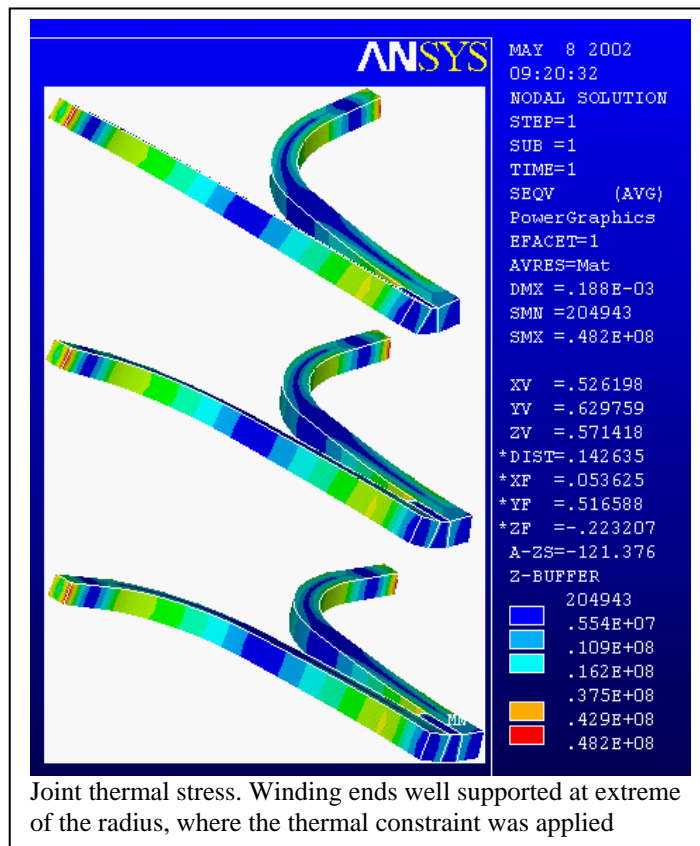
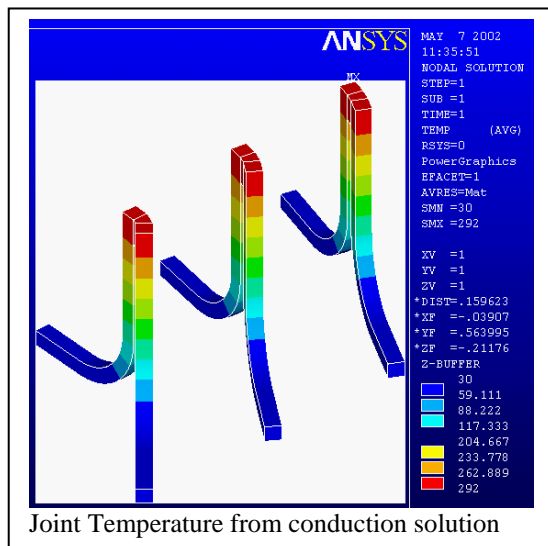
http://www.gore.com/en_xx/products/sealants/packing/gore-tex_valve_stem_packing.html

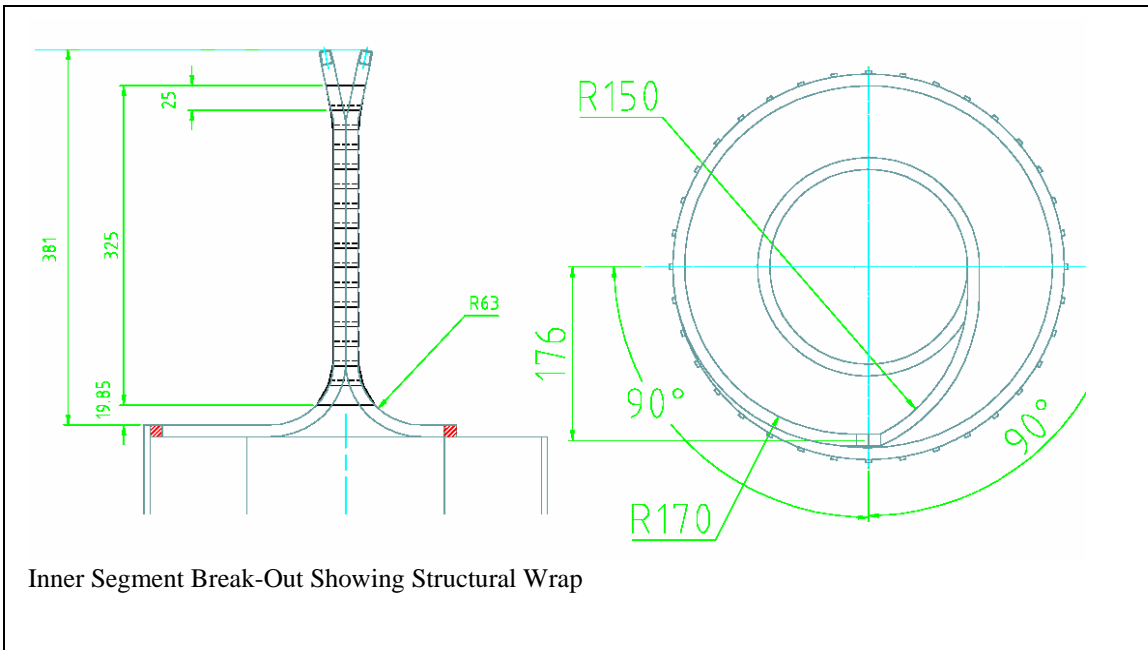
<http://www.kraftindustrialsupply.com/GORE-TEX%20VALVE%20STEM.htm>





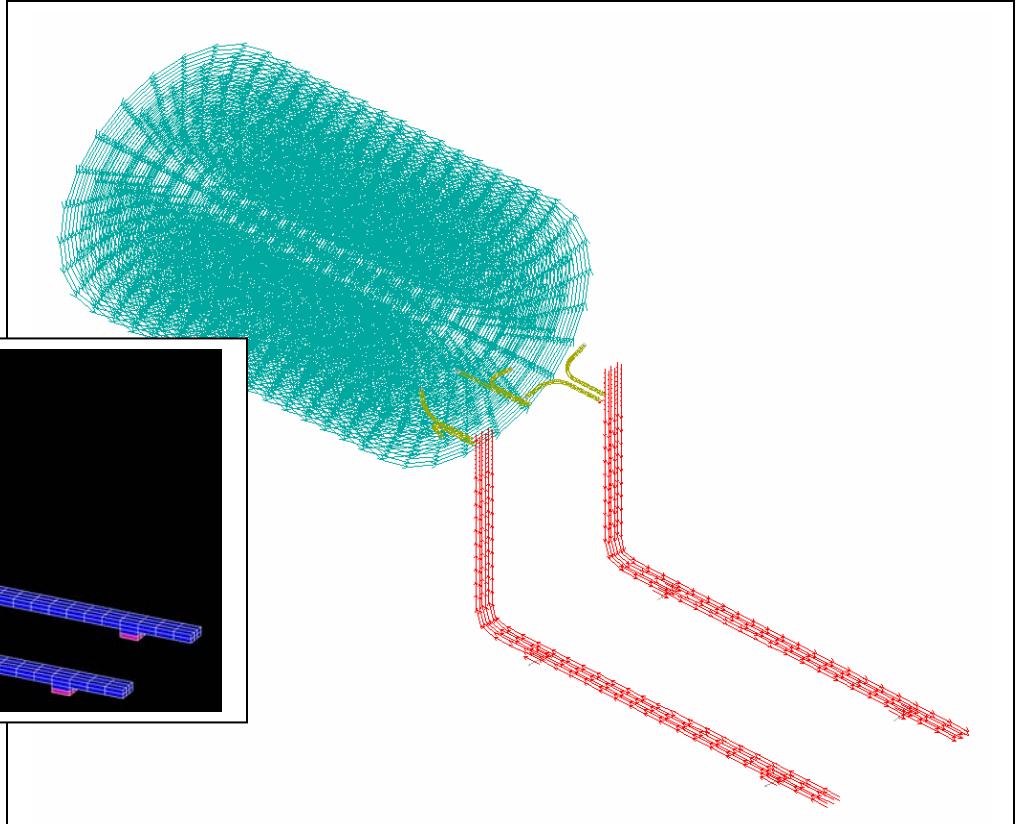
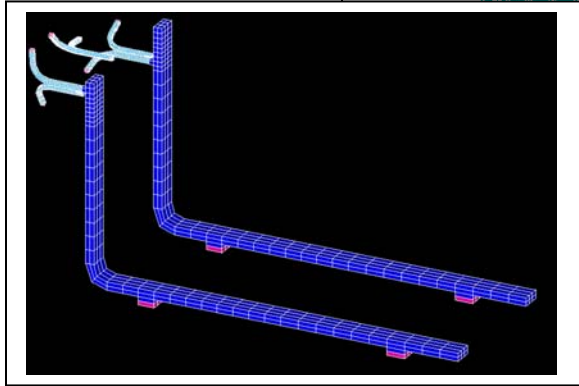






Loads on the Bus Bar Extensions

The leads are modeled as 1 X 3 inch bar/strap.

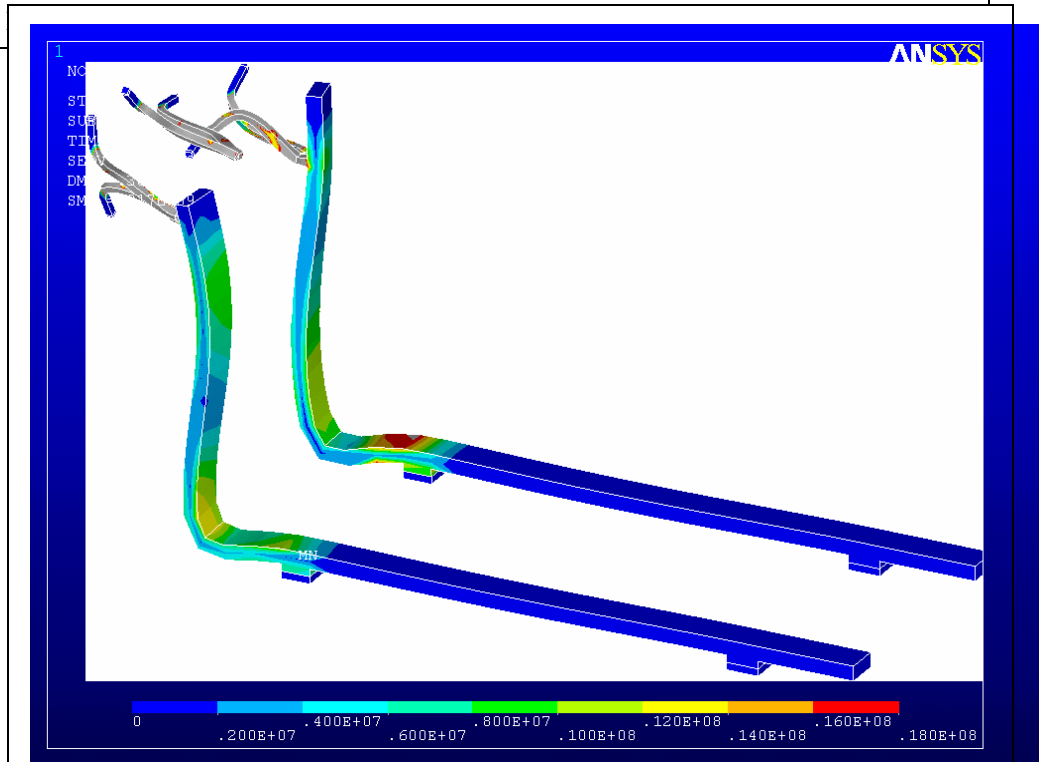


The total reaction for the 2 pads on the rear lead are:

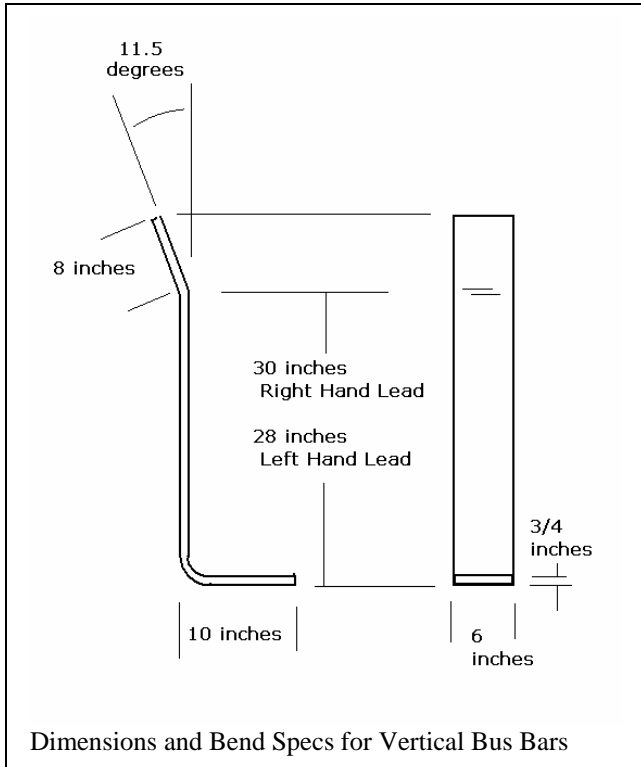
FX 113.73N
FY 1982N
FZ -24.5N

The total reaction for the 2 pads on the front lead are:

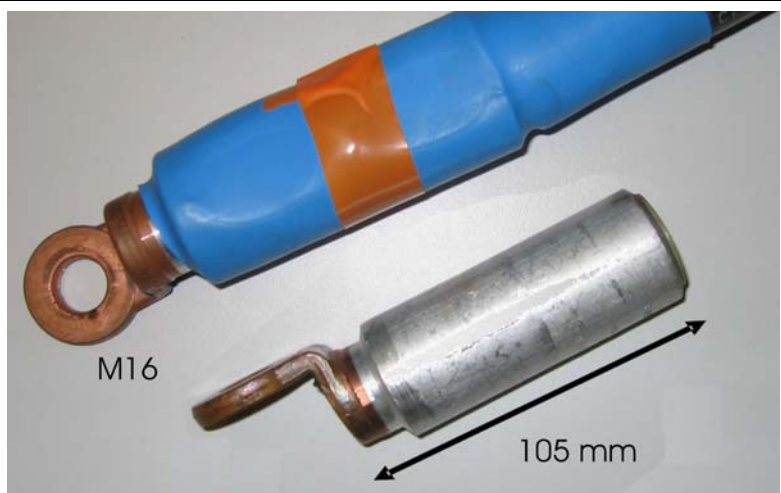
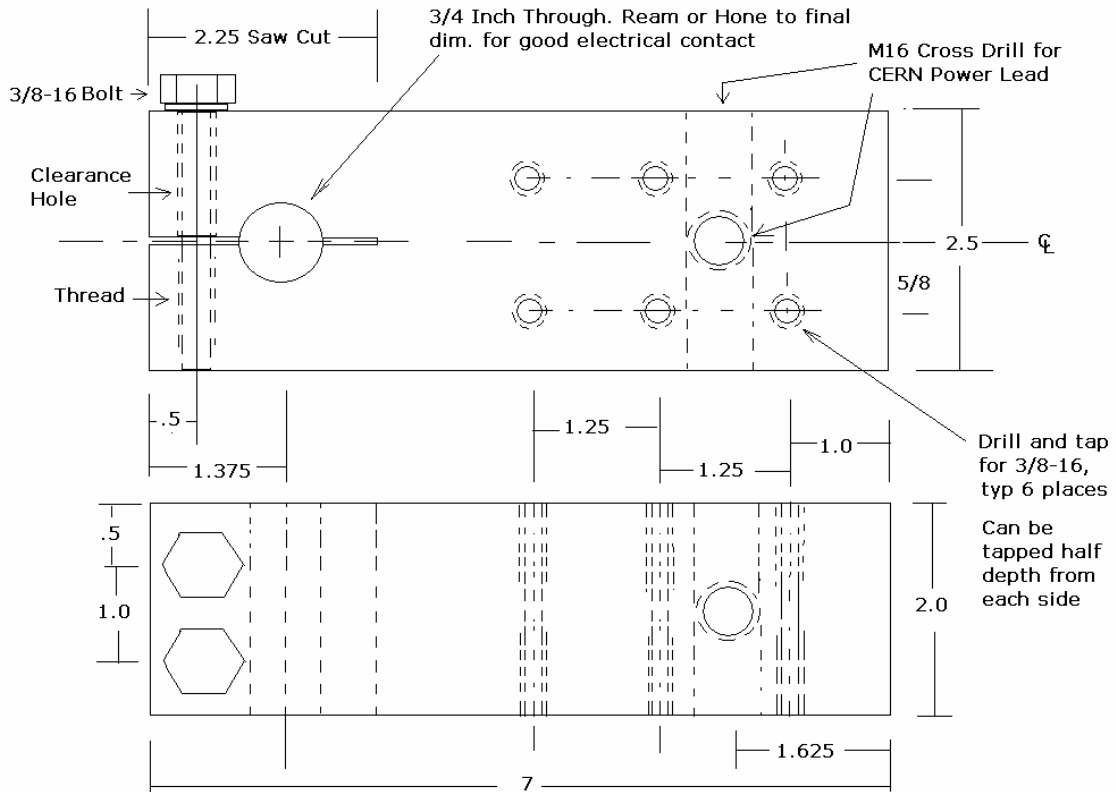
FX 627N
FY 1714N
FZ -17.25N



Strap Stress is only 18 MPa.



BNL/MERIT Magnet Lead to Buss Bar Terminal Block
 (Two Required) , Material: Copper
 Dimensions in Inches - Except M16 Thread

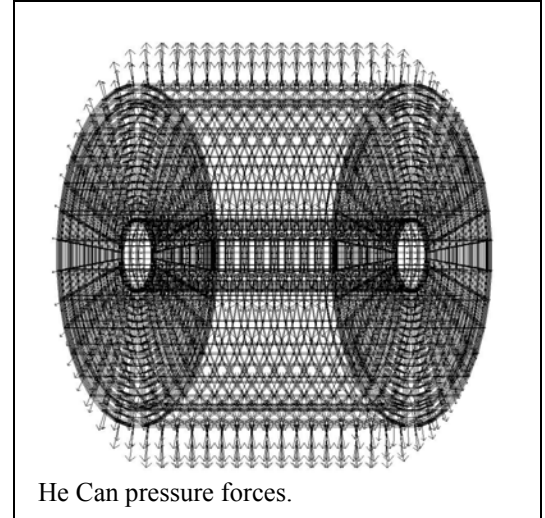


BNL/ME CERN Power Lead Termination

10.0 Helium Cryostat Stress

The normal operating pressure of the cryostat is 15 atm. In one of the considered cryogenic systems, there was a possible pressure excursion during cooldown to 20 atm. This resulted from the expansion of the warm He gas on the downstream side of the magnet. Other systems The relief valves will be set based on a 15 atm design pressure. The cryostat has a flat head at the side where the penetrations are made. This is bolted to the cryostat shell. Mechanical seals are used. The flat head thickness is 2 cm.. The shells are 304 for the Vacuum (outer) shell and 316 for the inner vessel. The outer vessel is 4.75mm thick, 1045mm OD and 1035.5mm ID. The Inner vessel is 12.7mm thick, 964mm OD and 938.6mmID. The Inner Vessel $p*r/t$ stress is $20*14.7*6895*(964+938.6)/4/.0127/1e6= 75.9216\text{MPa}$

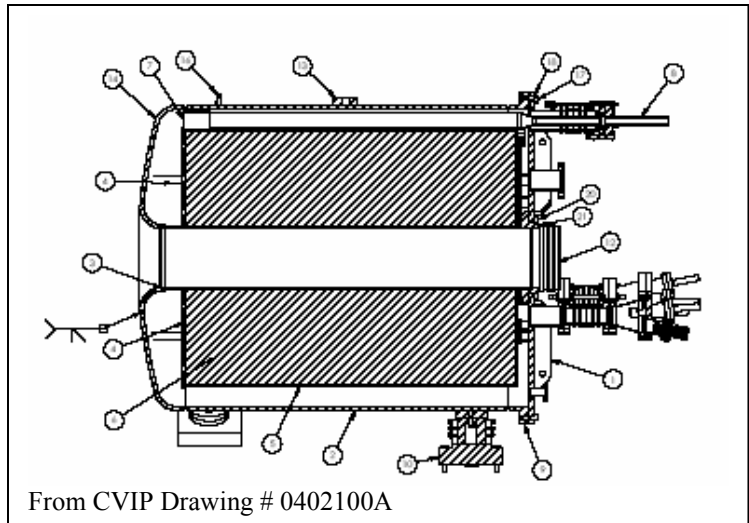
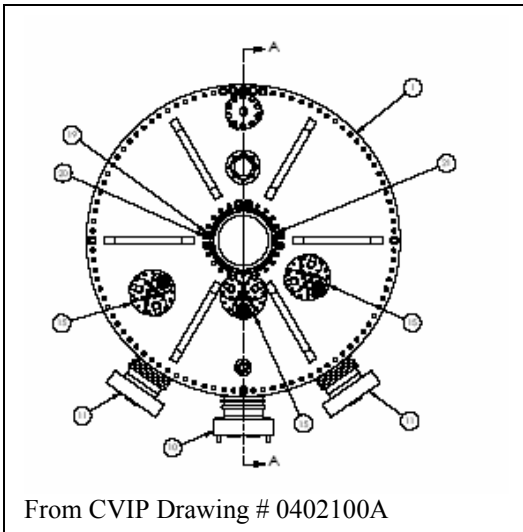
$\frac{1}{4}$ inch plate stock could have been used thickness is 6.35mm and the $p*r/t$ stress is 151 MPa. The 316 SST is recommended. Cold worked 304 SST starts to lose it's ductility below liquid nitrogen temperatures. The $\frac{1}{4}$ inch plate stock meets the membrane stress requirement. . (present finite element analyses are based on 5mm) Flat head thickness is 2 cm, Dished head is .5 inch thick, Vacuum Jacket Dished head is .125 in.

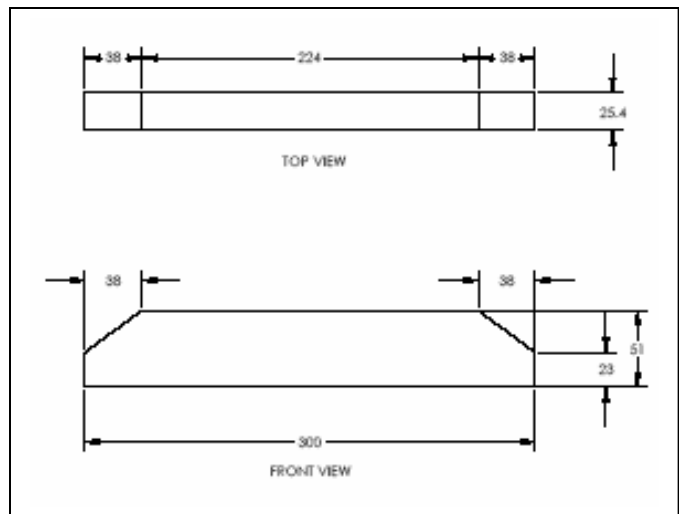
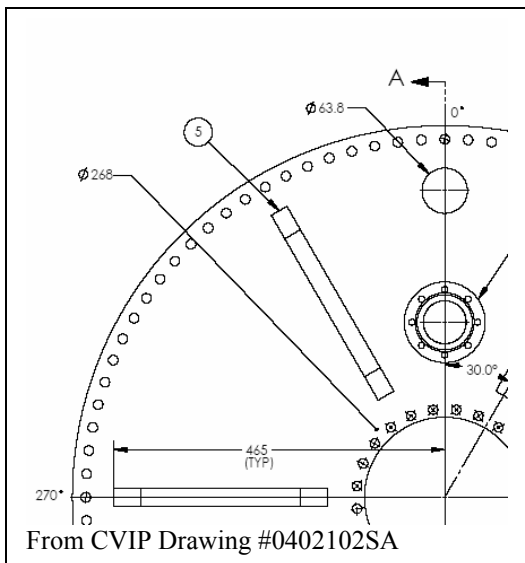
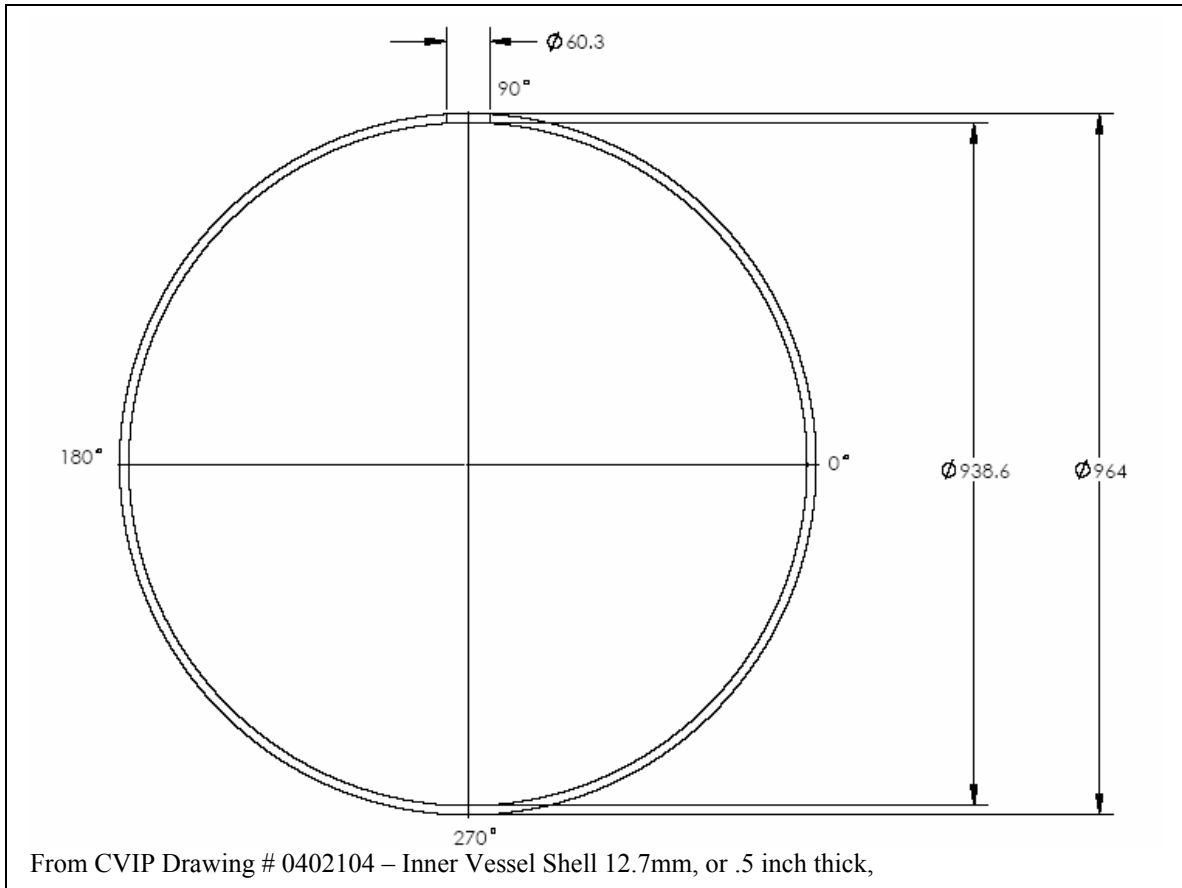


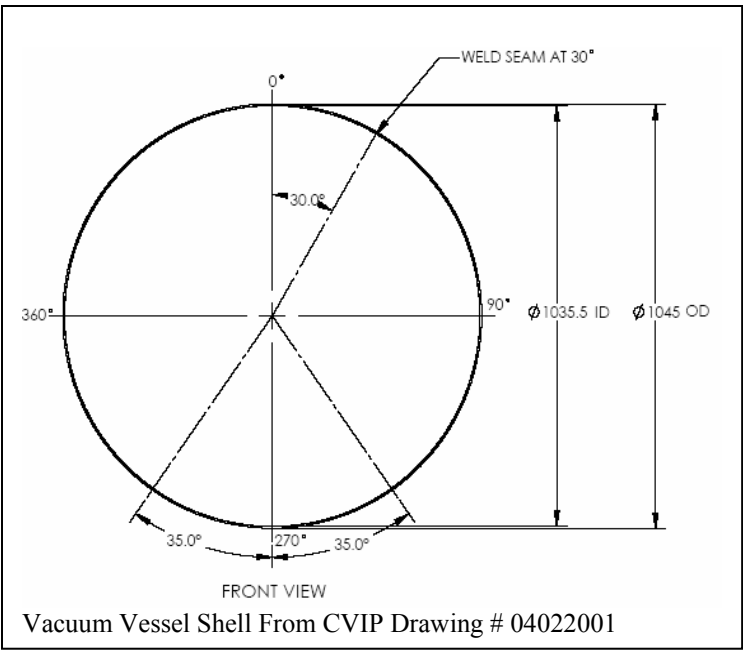
Structure Room Temperature (292 K) Maximum Allowable Stresses, S_m = lesser of 1/3 ultimate or 2/3 yield, and bending allowable= $1.5*S_m$

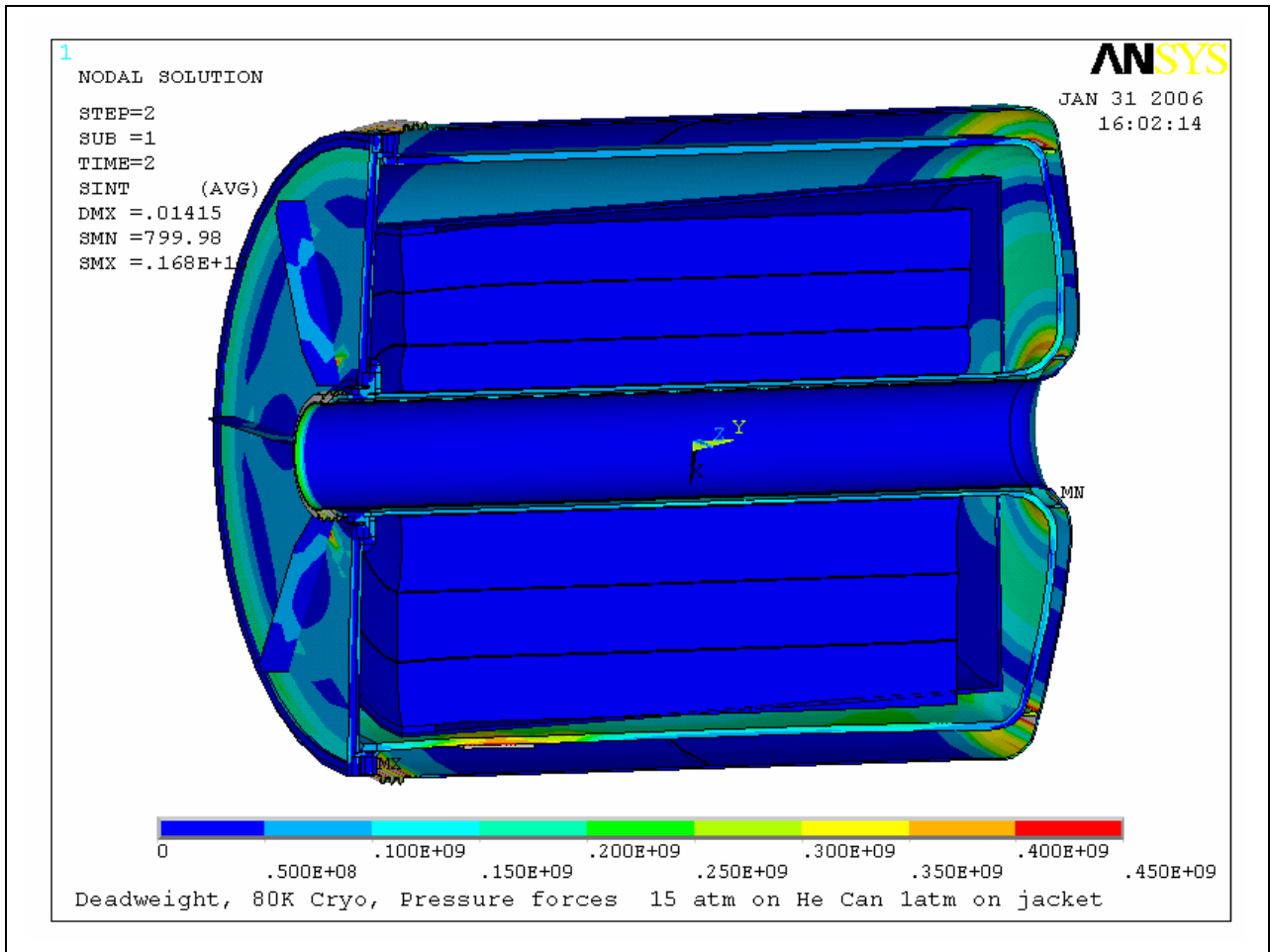
Material	S_m	$1.5S_m$ – bending
316 LN SST	183Mpa (26.6 ksi)	275Mpa, (40ksi)
316 LN SST weld	160MPa(23.2ksi)	241MPa(35ksi)

Without stiffeners on the flat head, local (corner) stresses are high – 700 MPa. After the addition of six 1cm thick stiffener ribs , the peak stress went down to 448 MPa. This occurred in the ribs, and the weld and head stress is much improved, but is still above 300 MPa further reductions are needed to meet the weld allowables, although the plate allowables summarized above are for room temperature, and are much below the cryogenic capability of 304 or 316.. Increasing the depth of the ribs from the 6cm modeled, is indicated.





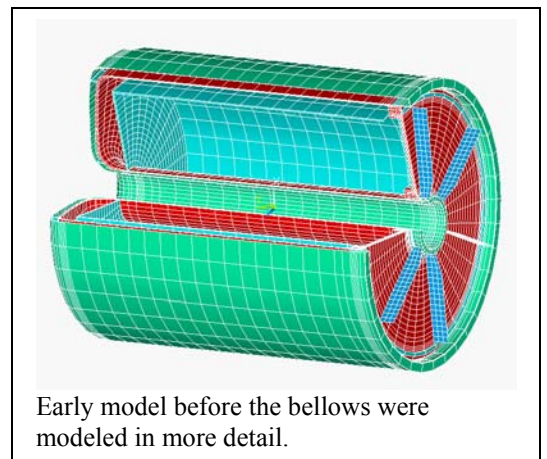




Pressure Test

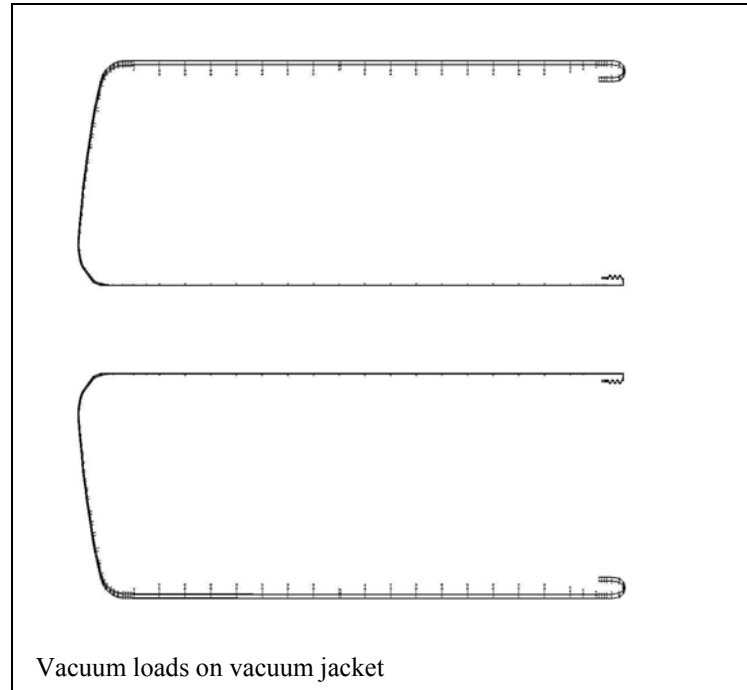
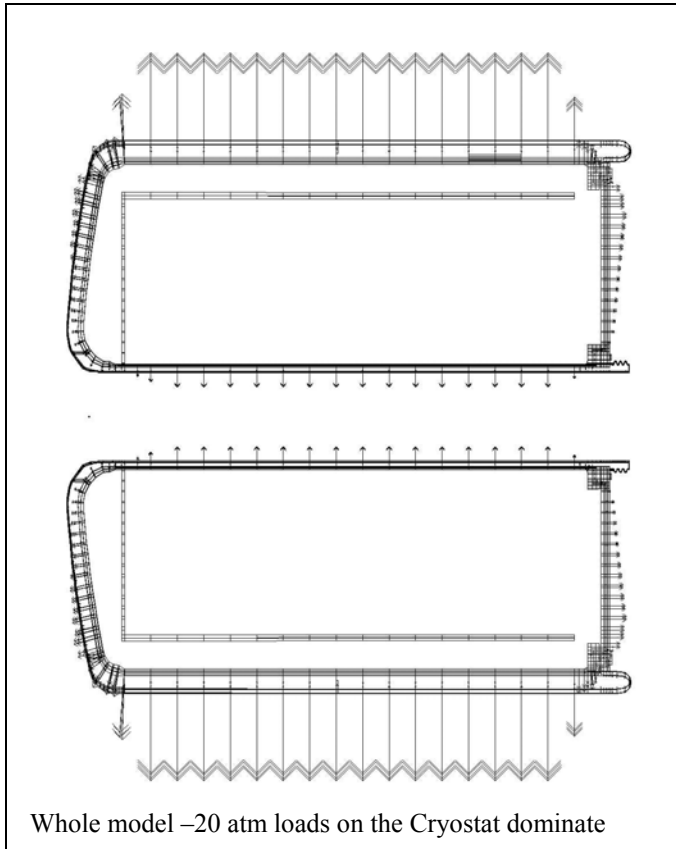
From a 11-21-05 email from David Nguyen:

“We performed pressure test last Saturday. The actual pressure was 245psi hold for 25 minutes. The inner flange did not leak at all. We only found a small leak at the top of the outer flange. We released the pressure, tight four bolts, and brought the pressure back up to 245psi again.”



10.1 Vessel Simulation Model

Forces, Pressure times element area



10.2 Shell Stresses – Simplified Analysis

Note that the ASME VIII sizing equations (UG27 and UG28) don't apply to the annular geometry. The inner bore tube of the cryostat is loaded in hoop compression and axial tension. It also supports the weight of the magnet. The axial stress contribution to the stress intensity is different than a normal vessel. In the outer shell the axial stress is reduced from a usual vessel stress. The inner shell is loaded in compression and the axial tension adds significantly to the stress intensity.

Cold Vessel Shell Stresses

IR	OR	t	Pressure atm	pressure MPa	P*t/t Stress	E*	Hoop σ/E	Axial σ	Tresca	Sm	F.S.
.09424	.1	.00476	-21	-2.128	-44.7	.7	-64	78.2	142	183MPa	1.29
.472	.482	.01	21	2.128	101.5	.7	145		145	183Mpa	1.26
.482	.4693	.0127	15	.103425	56.2	.7	80.2		80.2	183MPa	2.27

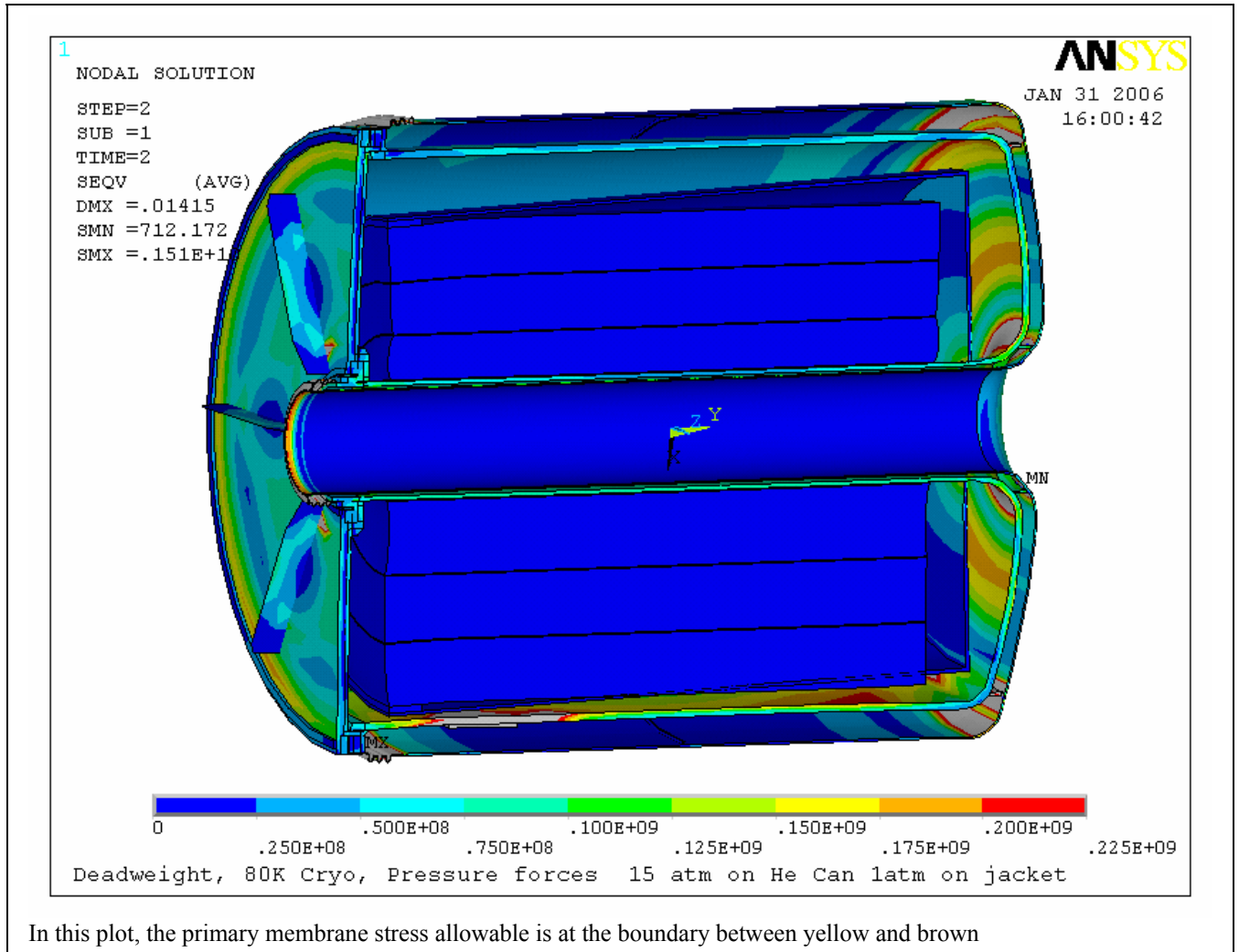
*Weld efficiency, from Table UW-12, For Longitudinal butt welds, welded from both sides, and ground smooth, with no radiographic examination, the weld efficiency is 70% .

The simplified stress analysis of the vacuum jacket shell stresses are considered in section 11.

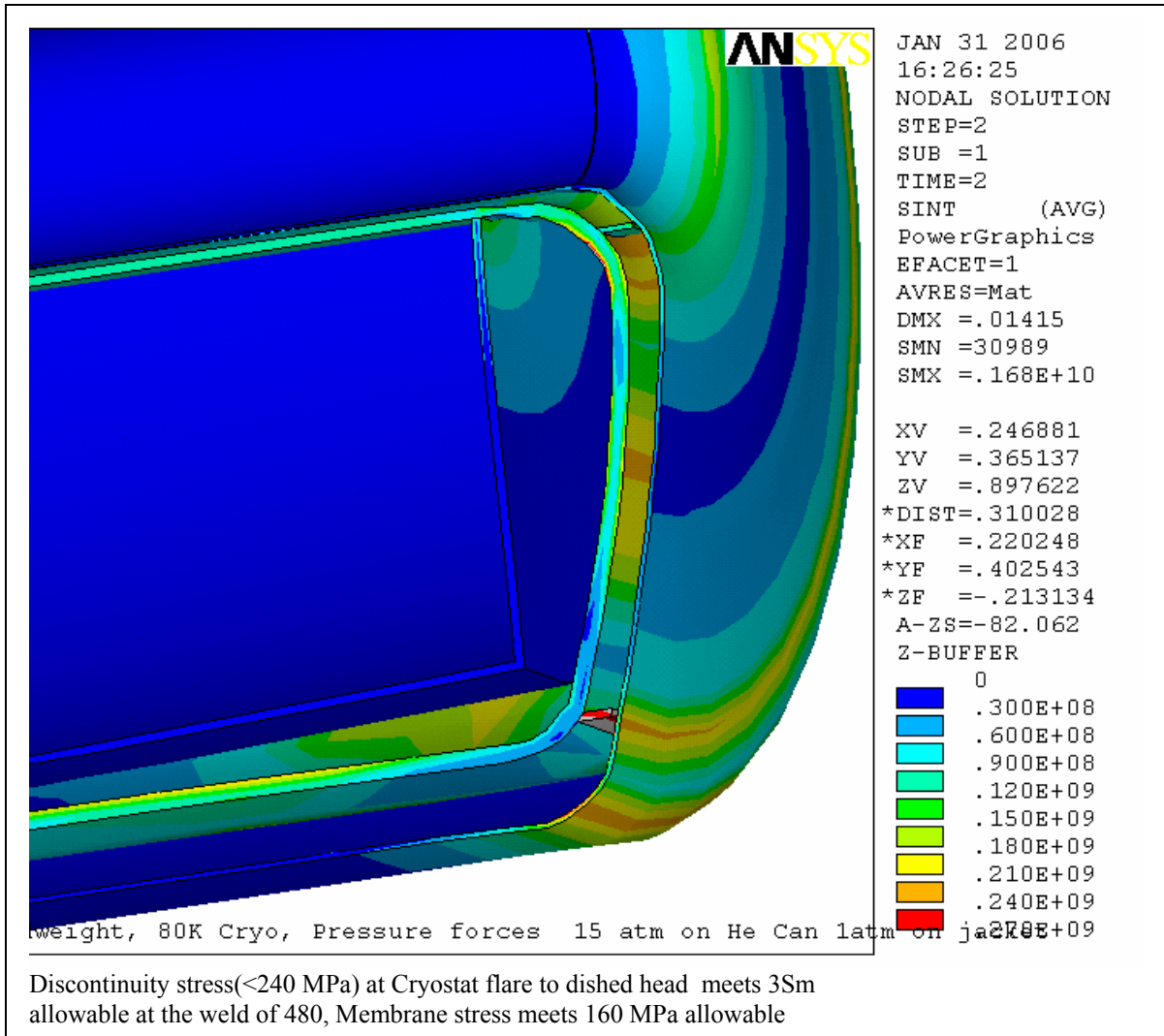
10.3 Simulation Model Results

Primary Membrane Stress

All Cryostat and Vacuum Jacket Stresses (with the exception of the bellows details) satisfy the primary membrane stress

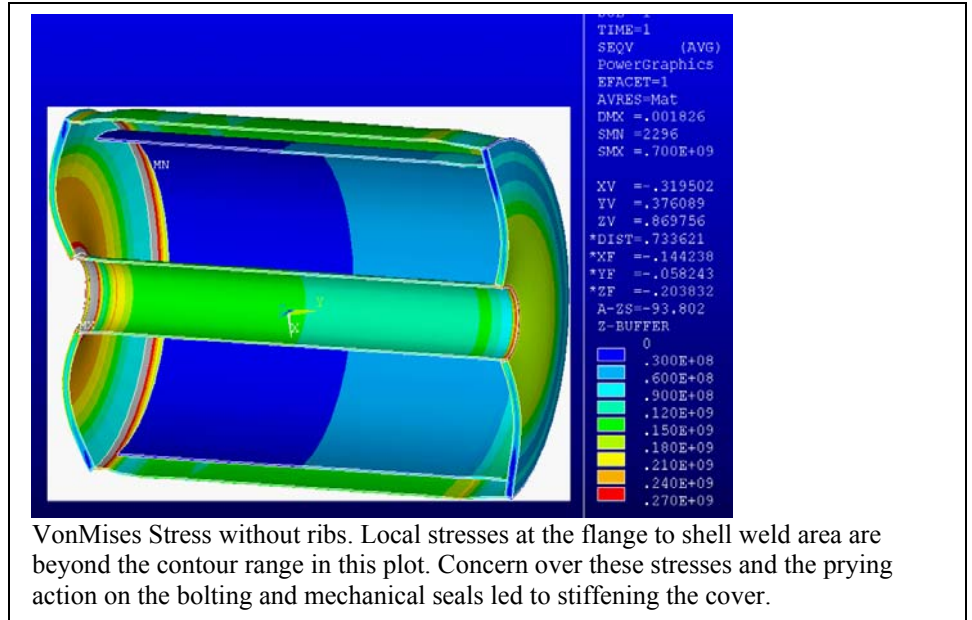


Discontinuity Stresses

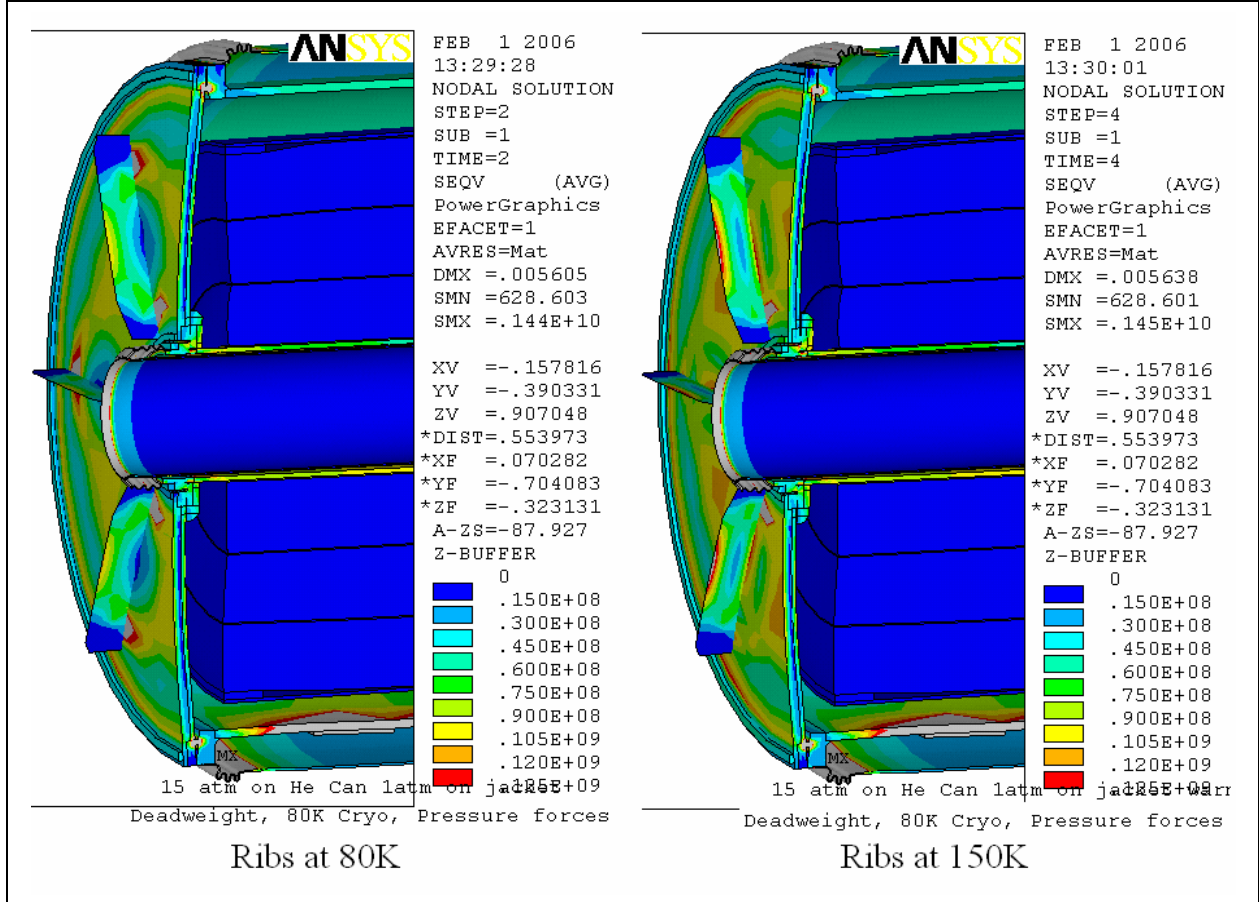


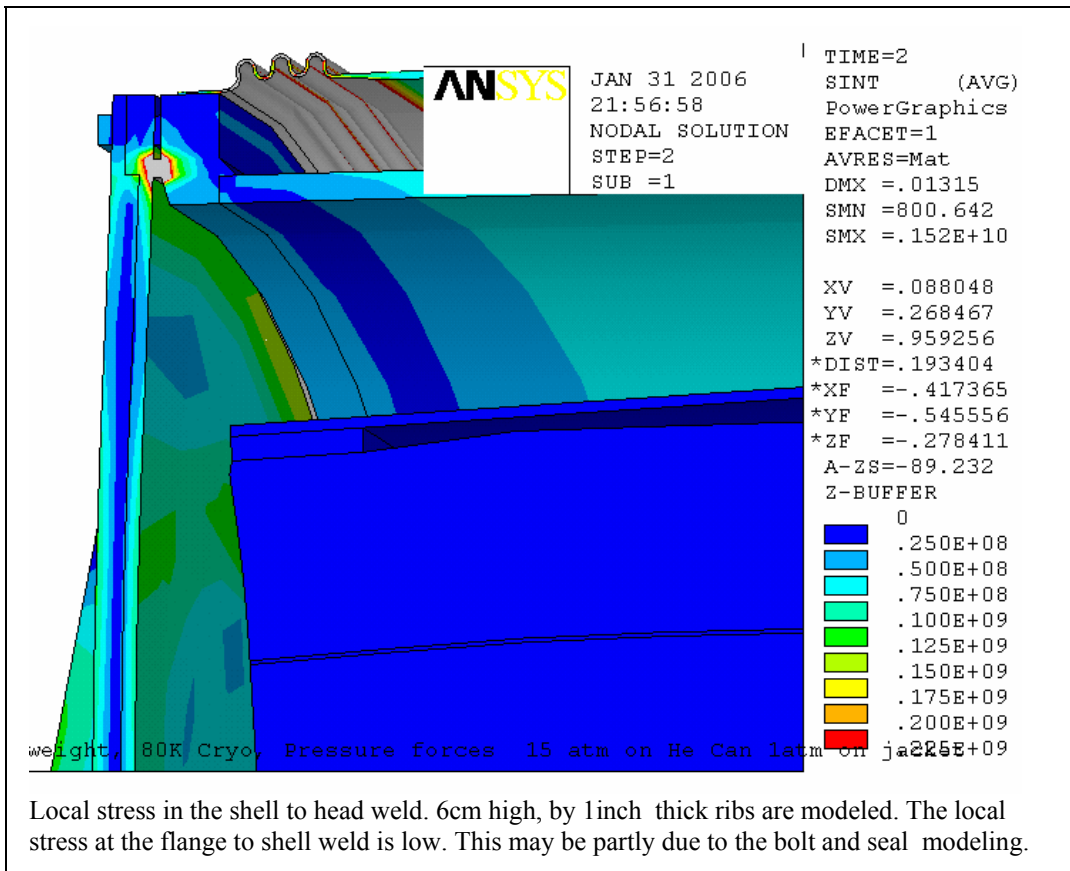
Cryostat/Helium Can Stress – Rib Stiffeners and Head Closure Details

Without stiffeners on the flat head, local corner weld stresses were high - 700 MPa. After the addition of six 1cm thick stiffener ribs, the peak stress went down to 448 MPa. This occurred in the ribs, and the weld and head stresses were much improved, but are still above 300 MPa. Further reductions were needed to meet the weld allowables, although the plate allowables quoted above are for room temperature, and are much below the cryogenic capability of 304 or 316. 1 inch thick ribs have been specified, partly to reduce the cover displacements that might effect the mechanical seals.



The ribs may protrude beyond insulation applied to the cover. If they are at a different temperature, this could impose some stresses on the ribs and closure details. This effect does not appear to be of concern.



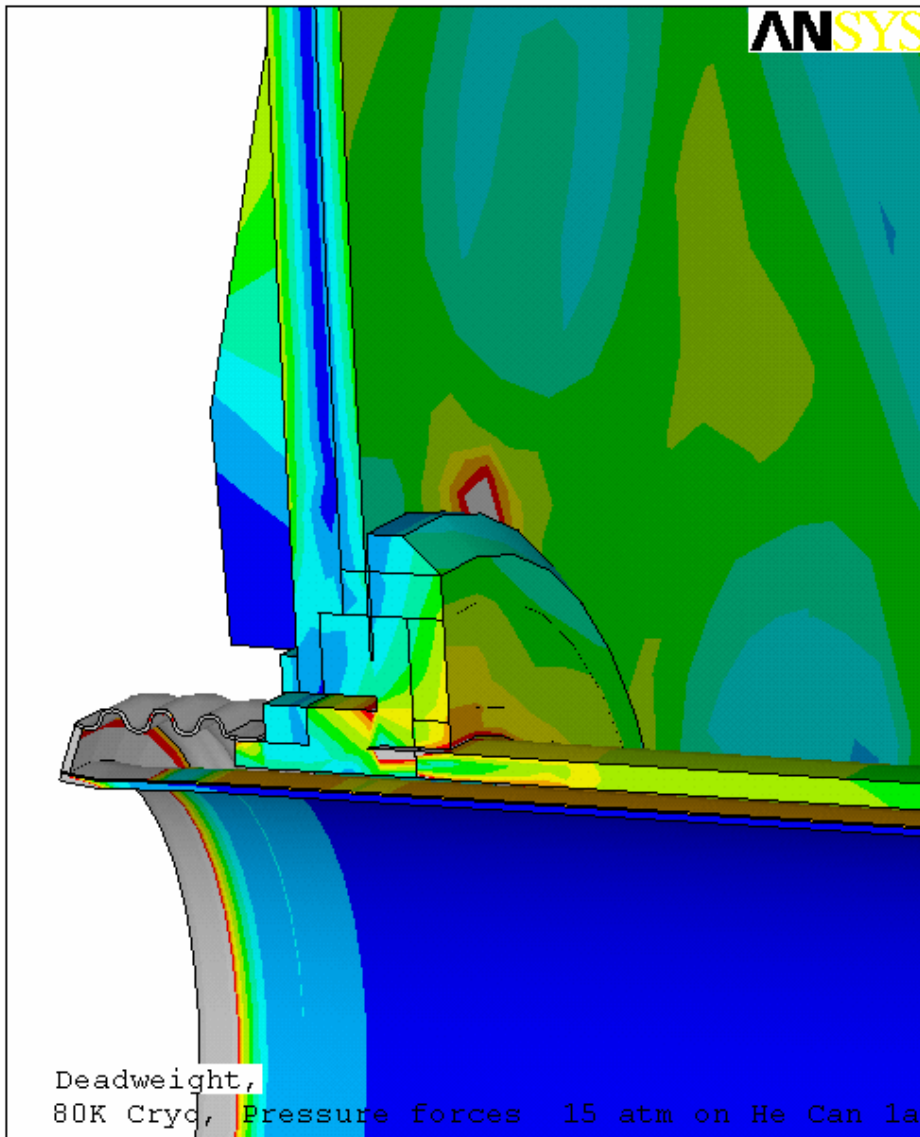
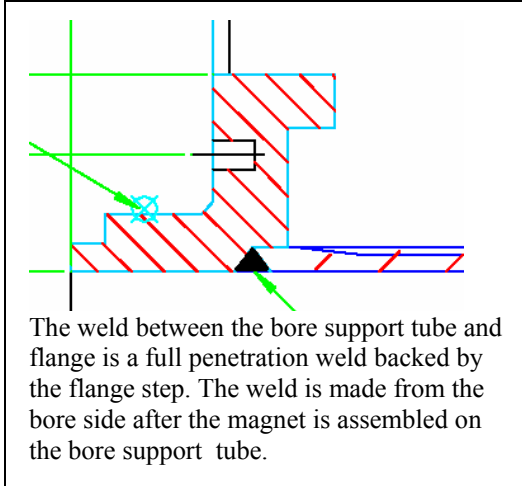


Local Flange to Shell Connection

Bending stress allowable is 241MPa (room temperature weld allowable in bending) OD flange stress is at the allowable.

ID flange has a F.S. of 1.5

Bellows details will be specified to sustain the 3 to 4 mm displacement and one atmosphere.



JAN 31 2006
 16:07:57
 NODAL SOLUTION
 STEP=2
 SUB =1
 TIME=2
 SINT (AVG)
 PowerGraphics
 EFACET=1
 AVRES=Mat
 DMX =.01415
 SMN =30989
 SMX =.168E+10

XV =-.28137
 YV =.357095
 ZV =.890682
 *DIST=.142218
 *XF =-.090077
 *YF =-.631488
 *ZF =-.130942
 A-ZS=-94.113
 Z-BUFFER

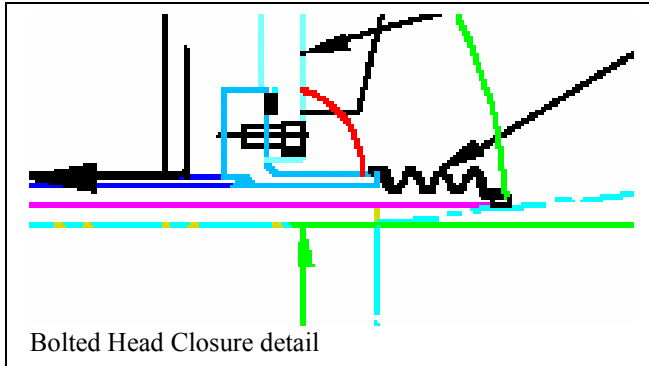
0
.150E+08
.300E+08
.450E+08
.600E+08
.750E+08
.900E+08
.105E+09
.120E+09
.135E+09

Deadweight,
 80K Cryo,

Pressure Forces 15 atm on He Can 1atm on ja

Local stress in the shell to head weld. 6cm high, by 1 inch thick ribs are modeled. The “leading edge weld is not modeled well but this will be a location of stress concentration. The contours for this plot were picked to eliminate the high stresses in the bellows and rib stiffener end, and show the stresses in the support tube where the flange end rotation will induce bending. Here the membrane stress is well below the weld membrane limit of 160 MPa. And bending allowable of 240 MPa

Cryostat Bolting Calculations



Bolts are used at an inner radius, and an outer radius of the cover flange of the Helium/LN2 cryostat. The cover area is large, and when the cryostat is pressurized to the nominal 15 atm for Helium gas operation, the axial load supported by the cover is quite large, approximately 250,000 lbs, which is reacted by an inner and outer bolt circle totaling 120 ½ inch bolts. A Basic program is used to calculate stresses in these cryostat cover bolts. First a reasonable method of apportioning the pressure load to the inner and outer array of bolts is needed. In the Basic program, moment equilibrium is applied to a strip section, and a mean radius of the pressure load is calculated. The inner and outer reaction loads are calculated from the contributory areas. Because the axial pressure load path is redundant, the finite element model may give a better idea of how much axial load is picked up in the bore tube and how much appears in the outer shell. This showed the Basic calculation to be conservative by 50%. 316 Stainless Steel properties improve at low temperatures, but due to the anticipated pressure test at room temperature, RT bolt and flange properties are assumed. This adds more conservatism for normal operating conditions. .

High strength 316 Stainless Steel bolts are used for the cover bolting. These have been specified as ASTM A193 Grade B8M - Class 2 - Type 316. For bolts ¾" and under, the ultimate strength, f_u is = 110,000 psi. and the yield strength, f_y = 95,000 psi. Thread shear failure is a possible concern, given that the bolts are stronger than the flange- This is only true at room temperature. At cryogenic temperatures, both the bolting and flange materials have a high yield stress. Direct tension on the bolt stress area is the limiting stress. The Basis program yield a factor of safety of 1.12 against the RT allowable for the cold worked 316 bolts chosen for the design. Because of space constraints, and uncertainty in the bolt tightening needed to seat the mechanical seal, the thread engagement will be less than the 1.5 diameter rule of thumb. In the calculation, one bolt diameter is assumed for the ½ inch bolts and 8mm is used for the M12 bolts based on the as assembled/as delivered thread engagement. This a factor of safety for thread shear on the inner bolt circle of 1.1617 against the RT allowable of annealed 316. This margin

Bolt Analysis

```

clear

!room temp bolt properties ASTM A193 B8M
class 2
let fy=95000
let fu=110000
let sshear=.4*275.8e6/6895 !316 RT annealed
let dpres=15 !atm
let dpres=15 !atm
print "Design Pressure=";dpres;" atm"
let sm1=2*95000/3
let sm2=110000/3
let sm=sm1
print "Allowable Stress=";sm
if sm2<sm1 then let sm=sm2
!let sm=.6*fy !AISC)
print "Bolt Ultimate Strength=";fu
print "Bolt Yield Strength=";fy
let numin= 24
let numout=6*16
print "Number of Inner Bolts:";numin
print "Number of Outer Bolts:";numout
let p=dpres*14.7
let ro=(1.007)/2
let ri=(.248)/2
let ro=(.964+1.007)/4
let ri=(.291+.248)/4
let rm=((ro^3-ri^3)/3)/((ro^2-ri^2)/2)
let f=p*pi*(rm^2-ri^2)*39.37^2
let ba=.375^2*pi/4
let ba=.5^2*pi/4
let bsa=.5^2*pi !pull out shear area
let bsa=pi*.5*.45*3/4 !pull out shear area
according to FED-Std-h28
let ba=.1416 ! 1/2 inch coarse thread
stress area
print "Bolt Tensile Area=";ba;" Bolt
Thread Shear Area=";bsa
print "Tensile Load on inner Cyl:";f;"lbs"
print "Tensile Load on inner
Cyl:";f/.2248;"N"

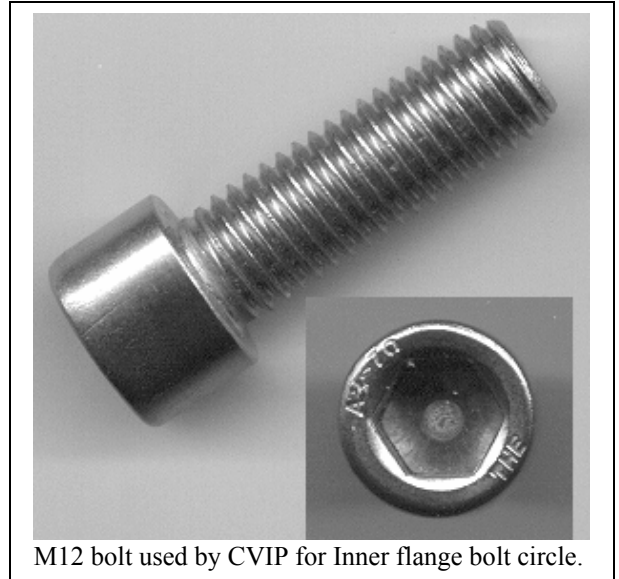
print "Inner Bolt Tensile
Stress";f/ba/numin
print "Inner Bolt Pull Out Shear
Stress";f/bsa/numin
print "Inner Bolt Tensile Factor Of
Safety";sm/(f/ba/numin)
print "Inner Bolt Shear Factor Of
Safety";sshear/(f/bsa/numin)
print "Inner Cylinder Stress Based on Bolt
Loading";f/.2248/(pi*.184*.004763)/1e6;"MPa"
"
let f=p*pi*(.5^2-rm^2)*39.37^2

let ba=.375^2*pi/4
let bsa=.5^2*pi !pull out shear area
let ba=.0773 ! 3/8 inch coarse thread
stress area
let ba=.1416 ! 1/2 inch coarse thread
stress area
print "Tensile Load on outer Cyl:";f;"lbs"
print "Tensile Load on outer
Cyl:";f/.2248;"N"
print "Outer Bolt Tensile
Stress";f/ba/numout
print "Outer Bolt Pull Out
Stress";f/bsa/numout
print "Outer Bolt Factor Of
Safety";sm/(f/ba/numout)
print "Outer Bolt Shear Factor of
Safety=";sshear/(f/bsa/numout)
print "Outer Cylinder Stress Based on Bolt
Loading";f/.2248/(pi*.986*.01)/1e6;"MPa"
end
    
```

will go up based on cryogenic properties of the 316 plate and bolt. Use of studs instead of bolts would eliminate the uncertainty in the engagement, and can be considered, once the dimensions and seating requirements of the seal are certain. In the Basic program, the ASME tensile allowable of the lesser of 2/3 yield or 1/3 ultimate, is used. If the AISC allowable of $0.6 \cdot f_y$ is used the factor of safety goes up.

Design Pressure= 15 atm with M12 Bolts

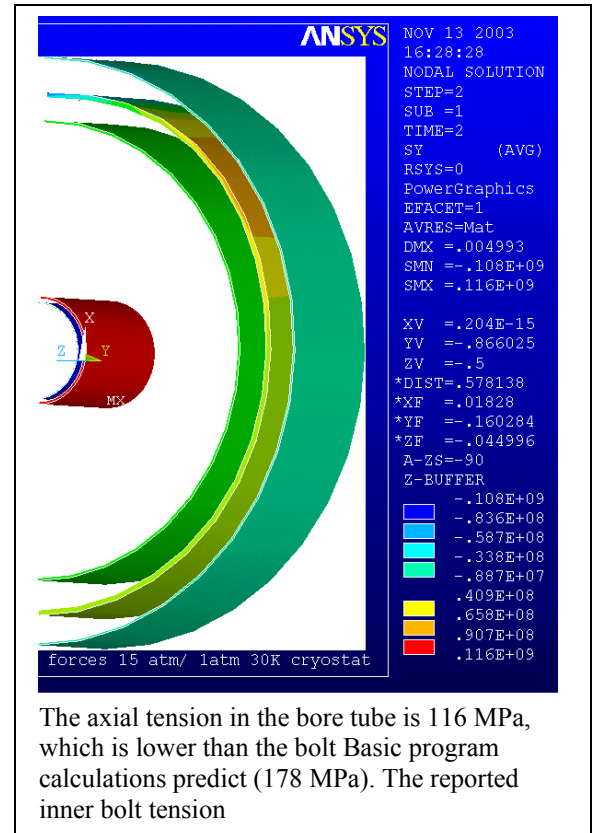
Allowable Stress= 57000
Bolt Ultimate Strength= 110000
Bolt Yield Strength= 95000
Number of Inner Bolts: 24
Number of Outer Bolts: 96
Bolt Tensile Area= .13033826 Bolt
Thread Shear Area= .50092224
Tensile Load on inner Cyl: 110378.99 lbs
Tensile Load on inner Cyl: 491009.75 N
Inner Bolt Tensile Stress 35286.067
Inner Bolt Pull Out Shear Stress
9181.3146
Inner Bolt Tensile Factor Of Safety
1.0391259
Inner Bolt Shear Factor Of Safety 1.1617
Inner Cylinder Stress Based on Bolt
Loading 178.33716 MPa
Tensile Load on outer Cyl: 138553.87 lbs
Tensile Load on outer Cyl: 616342.83 N
Outer Bolt Tensile Stress 10192.581
Outer Bolt Pull Out Stress 2881.2246
Outer Bolt Factor Of Safety 3.5973878
Outer Bolt Shear Factor of Safety= 5.5531943
Outer Cylinder Stress Based on Bolt Loading 20.309319 MPa



Design Pressure= 15 atm with 1/2 inch bolts

Allowable Bolt Stress= 57000
Bolt Ultimate Strength= 110000
Bolt Yield Strength= 95000
Number of Inner Bolts: 24
Number of Outer Bolts: 96
Bolt Tensile Area= .1416 Bolt Thread Shear Area= .53014376
Tensile Load on inner Cyl: 110378.99 lbs
Tensile Load on inner Cyl: 491009.75 N
Inner Bolt Tensile Stress 32479.694
Inner Bolt Pull Out Shear Stress 8675.2406
Inner Bolt Tensile Factor Of Safety 1.1289105
Inner Bolt Shear Factor Of Safety 1.8443293
Inner Cylinder Stress Based on Bolt Loading 178.33716 MPa
Tensile Load on outer Cyl: 138553.87 lbs
Tensile Load on outer Cyl: 616342.83 N
Outer Bolt Tensile Stress 10192.581
Outer Bolt Pull Out Stress 1837.6278
Outer Bolt Factor Of Safety 3.5973878
Outer Bolt Shear Factor of Safety= 8.7068776
Outer Cylinder Stress Based on Bolt Loading 20.309319 MPa

Pressure= 20 atm
 Allowable Bolt Stress= 57000
 Bolt Ultimate Strength= 110000
 Bolt Yield Strength= 95000
 Number of Inner Bolts: 24
 Number of Outer Bolts: 96
 Bolt Tensile Area= .1416 Bolt Thread
 Shear Area= .53014376
 Tensile Load on inner Cyl: 147171.99 lbs
 Tensile Load on inner Cyl: 654679.67 N
 Inner Bolt Tensile Stress 43306.259
 Inner Bolt Pull Out Shear Stress 11566.987
 Inner Bolt Tensile Factor Of Safety
 .84668286
 Inner Bolt Shear Factor Of Safety 1.3832469
 Inner Cylinder Stress Based on Bolt Loading
 237.78288 MPa
 Tensile Load on outer Cyl: 184738.49 lbs
 Tensile Load on outer Cyl: 821790.44 N
 Outer Bolt Tensile Stress 13590.108
 Outer Bolt Pull Out Stress 2450.1703
 Outer Bolt Factor Of Safety 2.6980409
 Outer Bolt Shear Factor of Safety=
 6.5301582
 Outer Cylinder Stress Based on Bolt Loading
 27.079091 MPa
 Ok. echo off



Size Designation	Nominal (Major) Diameter, D _n	Nominal Shank Area, A _n	Coarse Pitch Threads						Fine Pitch Threads					
			Threads per inch, n	Pitch (mm per thread), p	Pitch Diameter, d _p	Minor Diameter, A _s	M, UNRC Tensile Stress Area, A _{ts}	MJ, UNJC Tensile Stress Area, A _{ts}	Threads per inch, n	Pitch (mm per thread), p	Pitch Diameter, d _p	Minor Diameter, A _s	M, UNRF Tensile Stress Area, A _{ts}	MJ, UNJF Tensile Stress Area, A _{ts}
M12	12.000	113.10		1.7500	10.863	76.247	84.267	92.687		1.2500	11.188	86.037	92.072	98.311
0.5000	12.700	126.68	13	1.9538	11.431	81.103	91.548	102.63	20	1.2700	11.875	95.903	103.20	110.76

The M12 bolts have a stress area of 92.687 mm²

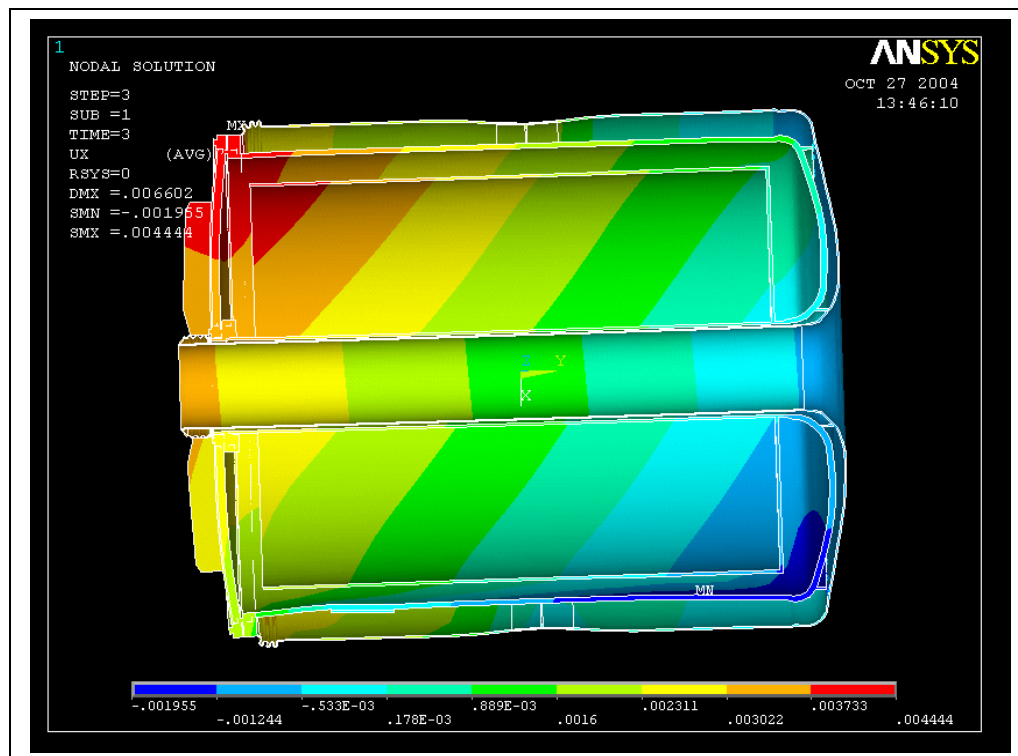
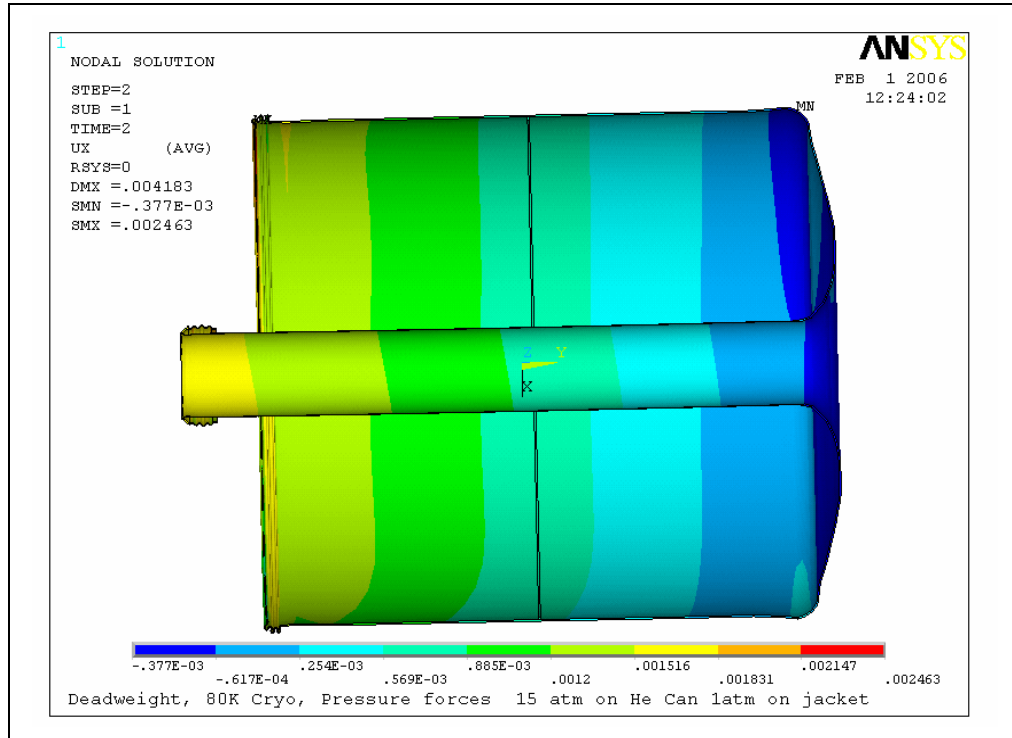
The M12 bolts still meet the stress allowables for 15 atm. There is some uncertainty in how much load appears on the inner bolt circle vs. the outer bolt circle. Finite element calculations and calculations based on the pressure centroid, both indicate acceptable bolt stresses at 15 atm.

Cryostat Inner Bore Tube, or Spline Tube

The OD of the vacuum jacket bore tube is 168.3 (CVIP #0402200SA) The ID of the spline tube is 184.0mm(CVIP# 040210001). This gives $184 - 168.3 = 15.7$ mm clearance. In an August 9th 2005 email David Nguyen indicated that CVIP purchased a pipe with an ID of 17.175" Which is about 182.245 mm. A small amount of clearance will be lost if the pipe ID is not machined to 184 mm. The only issue for this clearance is that the tubes need to be concentric enough so that there is no thermal short. The spline tube is the main structural element that supports the magnet, so we can't afford for it to be undersize in terms of thickness. If the ID was 178.3 mm that would give us 5 mm on the radius for the gap, which would be tough to maintain concentric, but would be OK if you can keep them from touching. As I remember we discussed with Chau Lac adding some G-10 stand-offs or small buttons to help maintain the concentricity. That would allow a $(184 - 178.3) / 2 = 2.35$ mm tolerance range on the inner radius. As of December 2005, the G-10 buttons or stand-offs were not added.

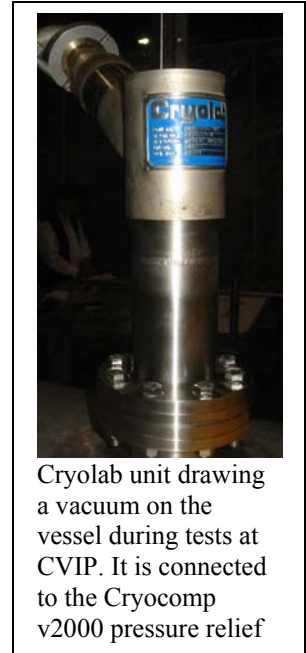
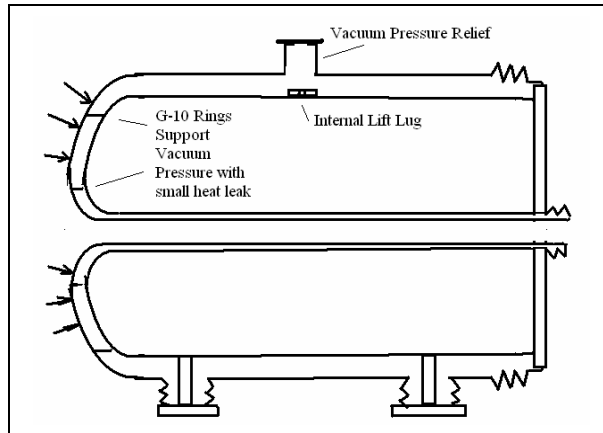
10.4 Cooldown Displacements

The latest (Feb 1 2006) Model shows the tilting expected as a result of cooldown. This is a consequence of the three point support. Displacements in the latest model are lower than previous analyses due to updates of shell thicknesses. In the following plots, X is positive down. The bellows end of the bore tube moves downward 1.8 mm and the dished head end moves upward by .4mm. In the previous analysis the bellows end of the bore tube moved down 3.7mm and the dished head end moved upward about 1.2 mm. This will be measured during the tests at MIT.



11.0 Vacuum Jacket

The Vacuum jacket provides thermal insulation around the bulk of the surface area of the cold vessel, leaving the flanged cover end to be insulated by foam. It's design pressure is a vacuum or 1 atm. external pressure. To allow for possible pressurization, a relief valve is provided at the top. Since the inner cryogenic vessel is fully welded and tested to 16.66 atm., there is no credible pressure that would develop from leaks of the inner vessel. The metal seals and lead gland nuts do not leak to the vacuum boundary. In-leakage through cracks in the bellows are conceivable, and condensation may form in the vacuum boundary that could develop a pressure when the inner vessel is warmed. The vacuum boundary pressure relief system is intended to preclude any significant pressure differential. The Vacuum Vessel is 4.75mm thick. It has been structurally tested under vacuum at CVIP in December of 2005. Vacuum vessel stresses are quantified in section 10 in a global modeling of the cold vessel and vacuum jacket together. Shell buckling is considered in section 11.3. A simplified stress analysis of the shell is presented in the following table.

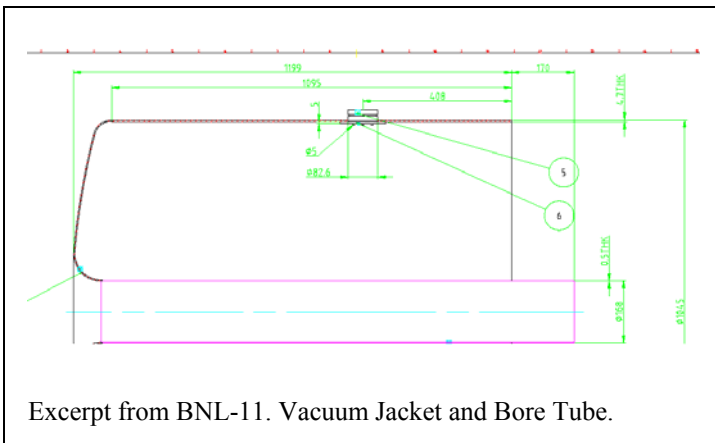


Cryolab unit drawing a vacuum on the vessel during tests at CVIP. It is connected to the Cryocomp v2000 pressure relief

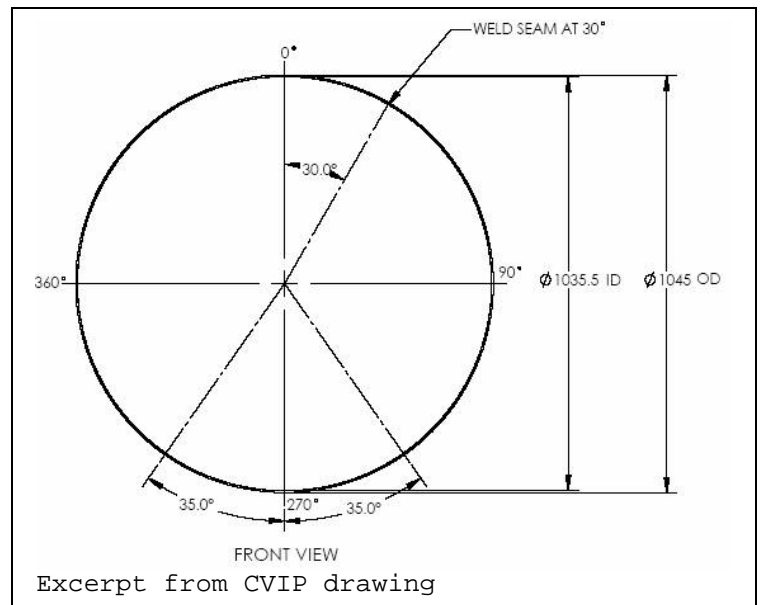
	IR mm	OR mm	T mm	Pressure atm	pressure MPa	P*r/t Stress MPa	E*	Hoop σ/E MPa	Axial σ	Tresca	Sm	F.S.
	517.75	522.5	4.75mm	1.0	.101	11.1	.7	15.8	5.5	16.6	156**	9.4

*Weld efficiency, from Table UW-12, For Longitudinal butt welds, welded from both sides, and ground smooth, with no radiographic examination, the weld efficiency is 70% .

** Table 4.8 Room Temperature Annealed 304 Bar – Rolled plate used in the shell should be higher

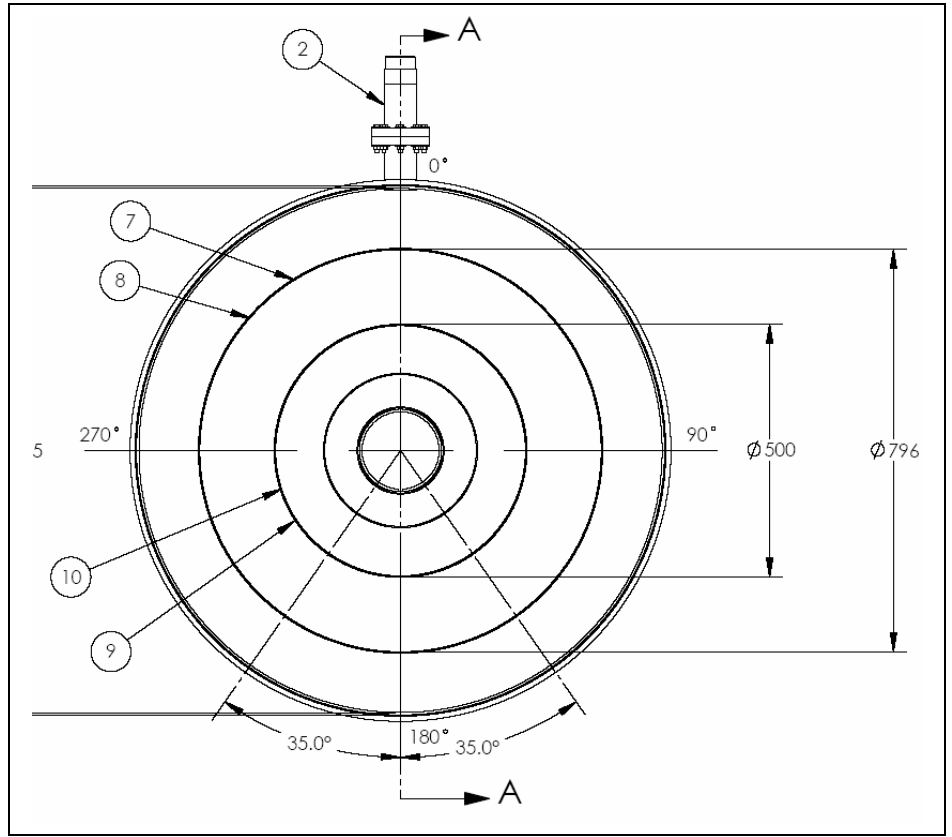
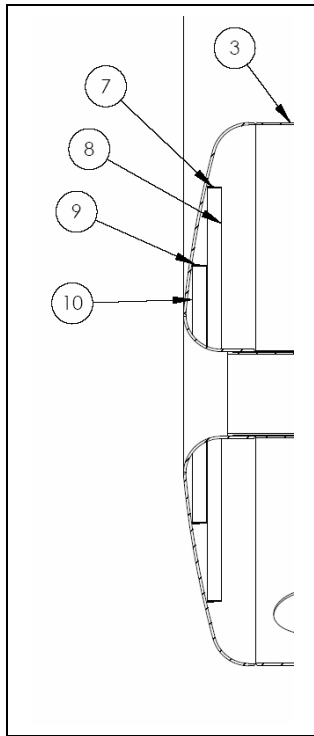


Excerpt from BNL-11. Vacuum Jacket and Bore Tube.



Excerpt from CVIP drawing

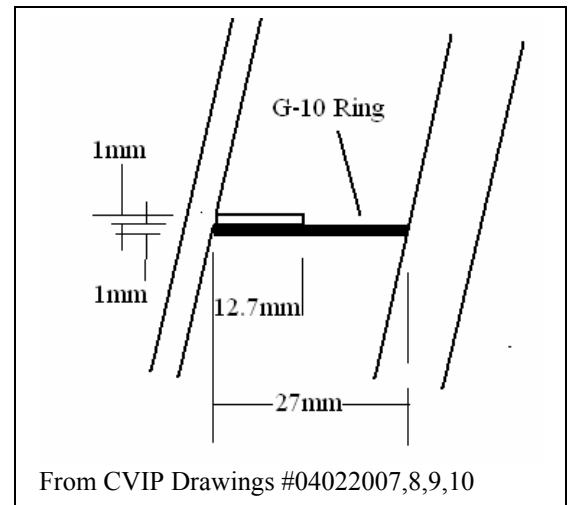
11.2 G-10 Rings



The vacuum pressure load on the dished head at the end of the vacuum jacket is $(1.045 \times 39.37)^2 \times \pi / 4 \times 14.7 = 19542$ lbs, or 86931 N. The compressive stress on the rings if they are evenly loaded is $86931 / (.796 \times \pi \times .001 + .5 \times \pi \times .001) = 21$ MPa. The room temperature tensile strength is about 300 MPa, For the high density strip used for the hoop, the compressive strength should be similar. This would allow $21/300 = .071$ or 7% of the two rings to be reliably in contact to support the vacuum load.

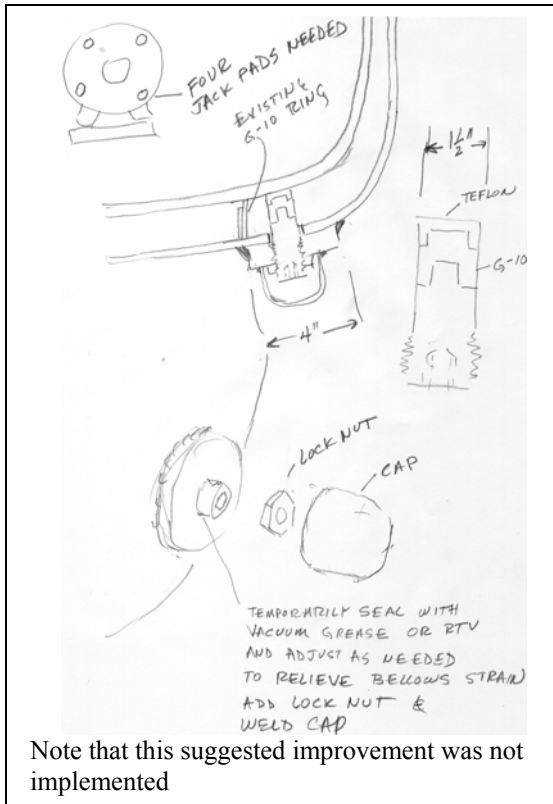
Net Support Bellows Load

The bellows at the support feet produce an uncompensated load. They are 5 inches in diameter. The two angled feet would produce a total vertical load of $5^2 \times \pi / 4 \times 14.7 = 577$ lbs. each. The two angled feet would transfer their loads to the G-10 ring. The centered support foot would transfer its load to shear in the larger inner and outer bellows. The g-10 rings are loaded to 19,542 lbs and would have a friction capacity of $.3 \times 19542 = 5862.6$ lbs., ten times the net uncompensated vertical load.

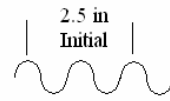


Vacuum Jacket Behavior under Pressure and Possible G-10 Ring Repair (If Needed)

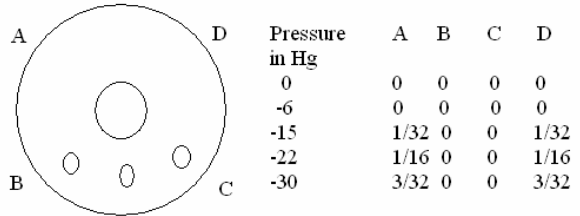
The vacuum system was successfully evacuated with no structural failure. This was witnessed by Kirk McDonald. During the retention test, the central support at the terminal end experienced a leak in the bellows that was attributed to a poorly manufactured bellows from the bellows supplier. There was a flaw in the collar weld. An attempt was made to repair the bellows, and during the next evacuation, all the bellows appeared to move excessively, and the repaired bellows leaked. A new bellows was ordered. It was installed the week after Christmas and the shell was marked and measured prior to evacuation. During the next evacuation the shell, bellows and bore tube displacements were monitored. From the measurements, the inner G-10 ring is probably not loaded. There is indication of flexure of the head translating to movement of the bore. Tapping the dished head and noting whether the head “rang” or responded with a “thud” gave a less rigorous indication of contact. This confirmed that the inner ring was not in contact. Based on this method, the outer ring contact is at three regions of 12 to 18 inches in length.



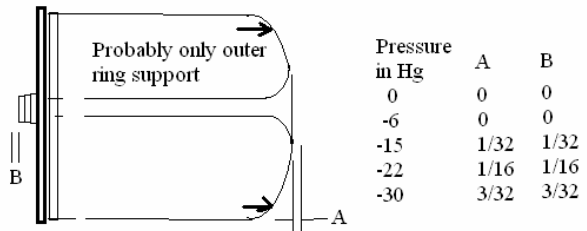
Note that this suggested improvement was not implemented



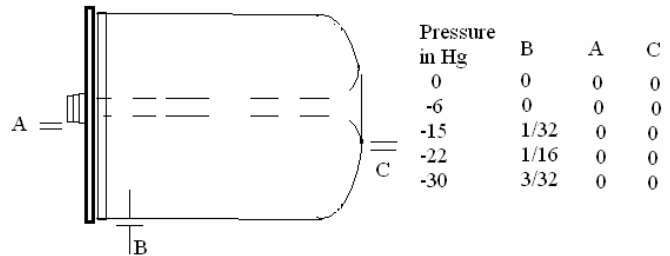
Large Bellows Axial Contraction Starting at a marked 2.5 inches



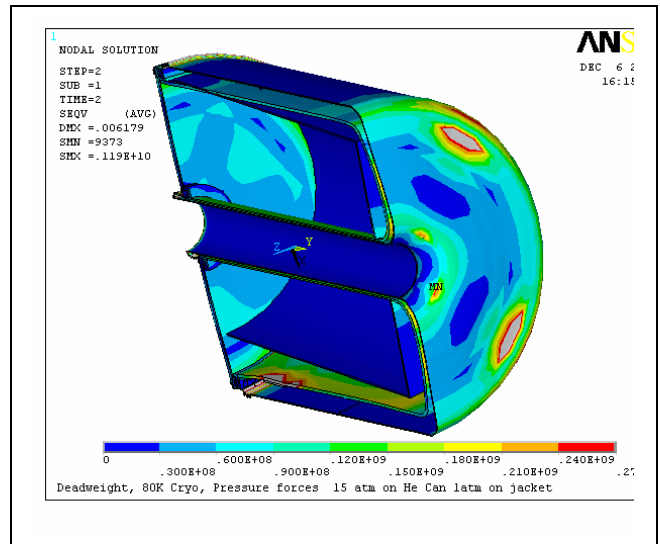
The Inner Bore moves Towards the Terminal End, Bottoms out to the G-10 Ring, and then Flexes the Inner Part of the Dished Head



The Vacuum Jacket Moves Downward at B (But not A) as the outer G-10 Ring Seats

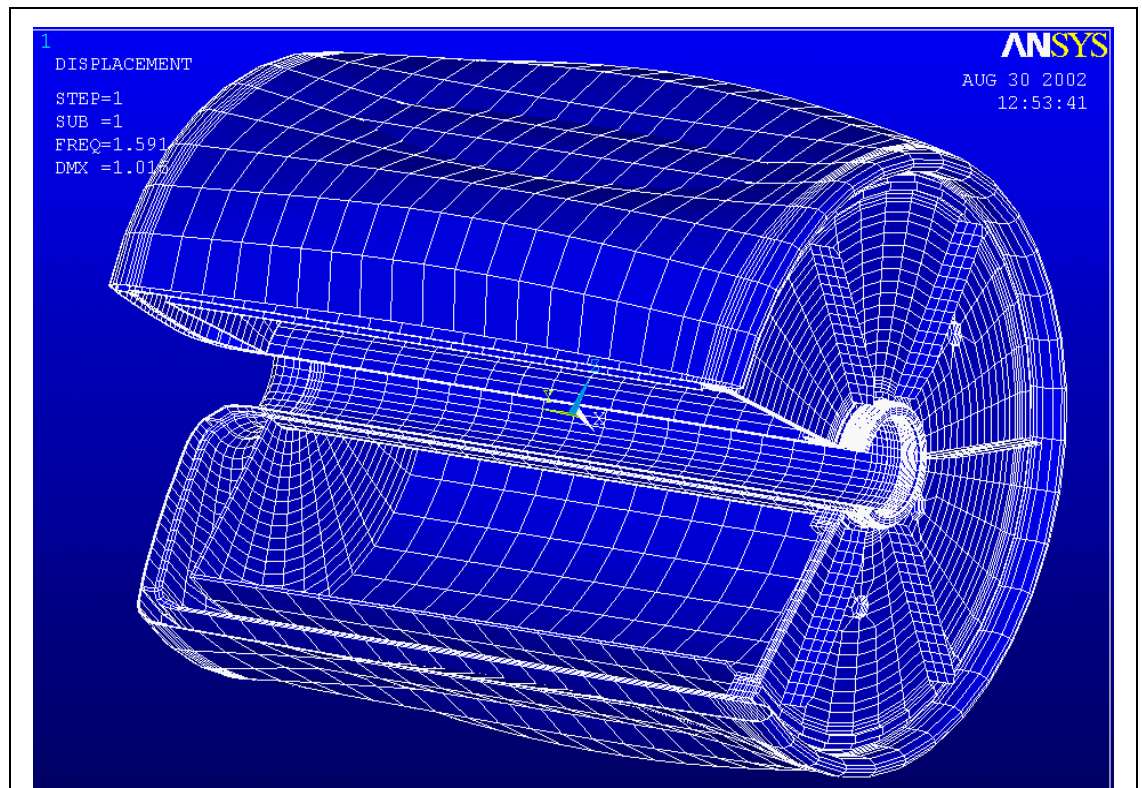


Displacements under Vacuum Pressure, Taken at CVIP
December 29th 2005

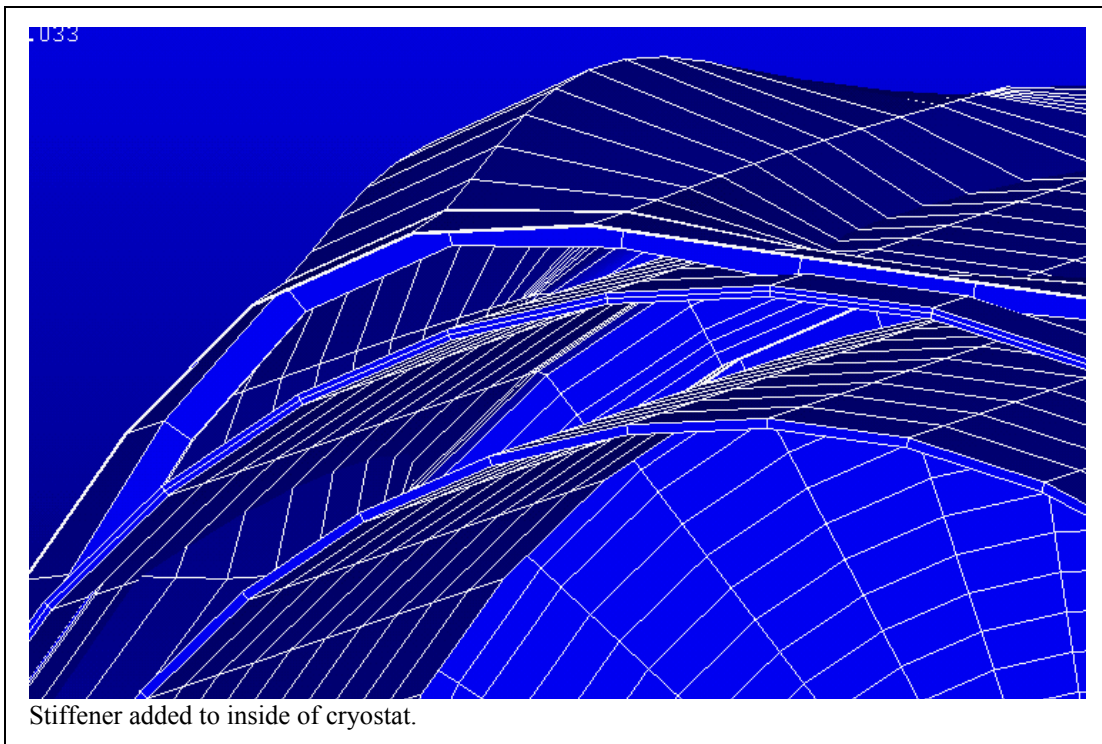
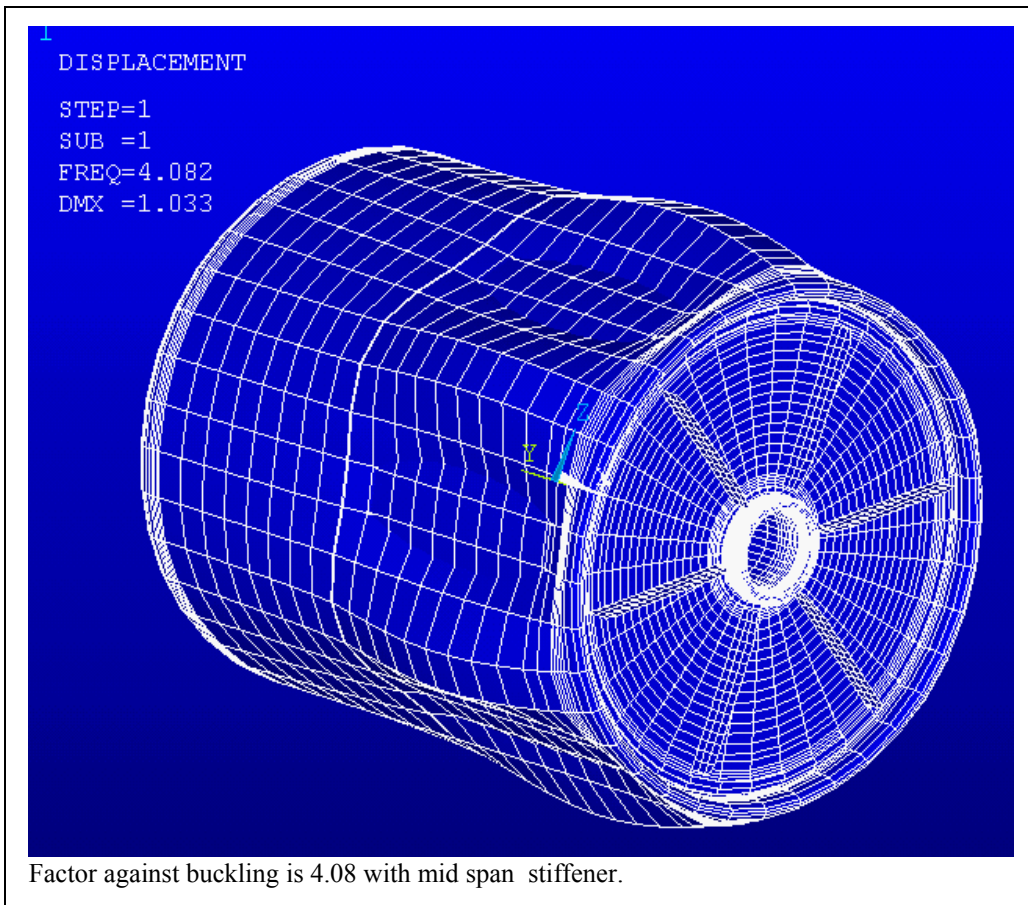


11.3 Vacuum Jacket Linear Elastic Buckling (Eigenvalue Method)

A 1mm thick vacuum jacket was analyzed. The 1mm thick vacuum jacket only has a margin of 1.5 against buckling. A factor of 5 is needed. Stiffeners were added in the analysis, and this was found to produce adequate buckling behavior. BNL drawing, BNL-011 shows 5mm walls for the vacuum jacket. The actual thickness supplied by CVIP is 4.75 mm. This should be more than sufficient, and the stiffener will not be required.

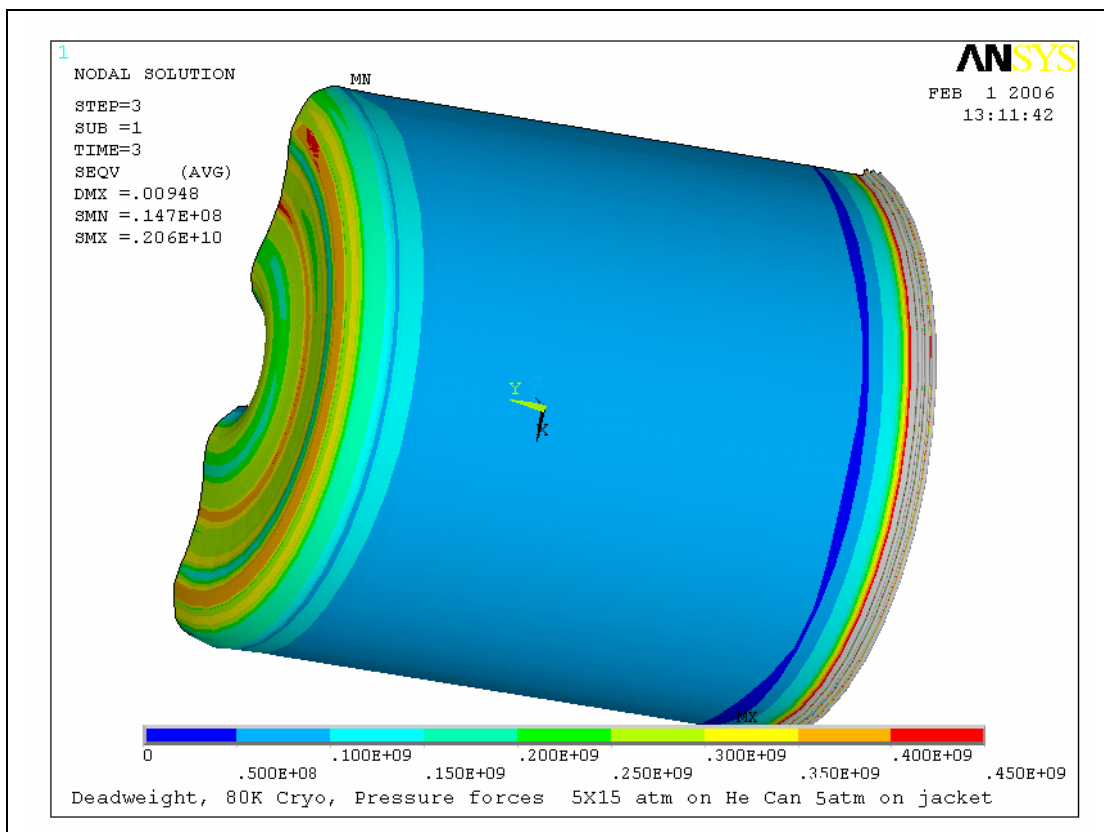
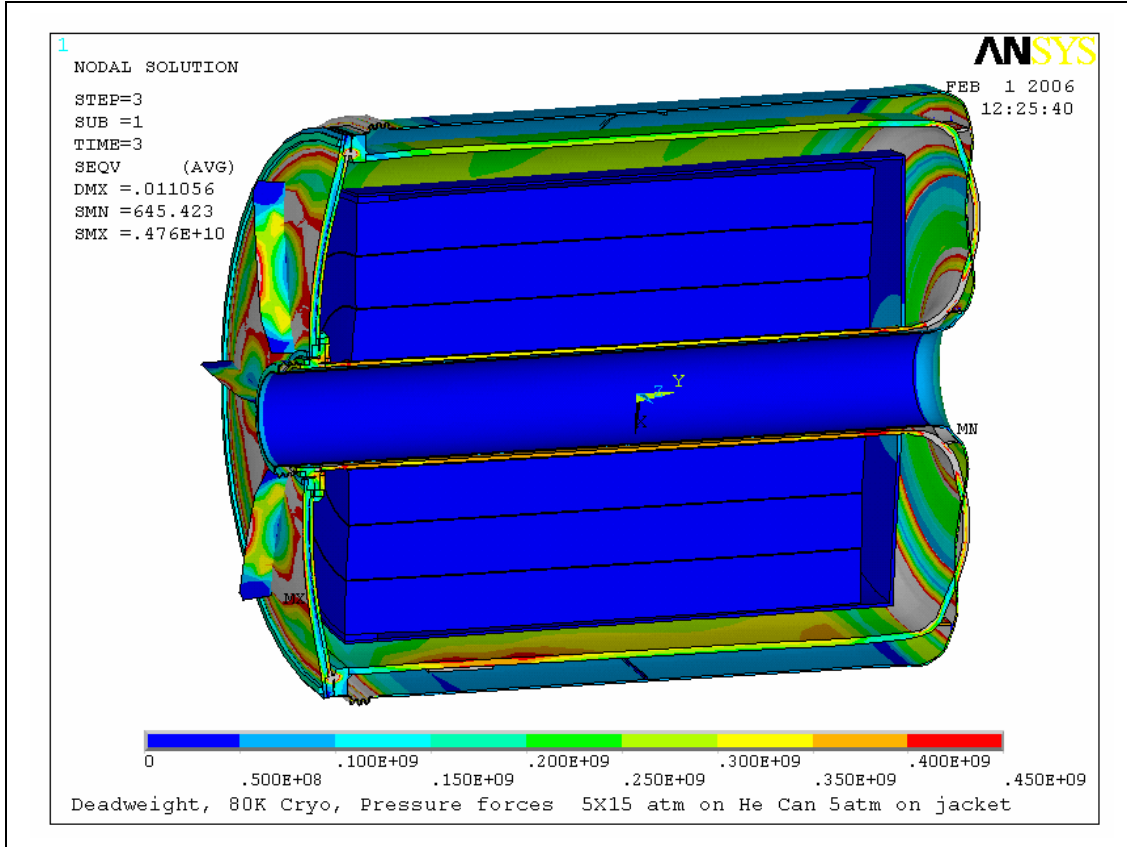


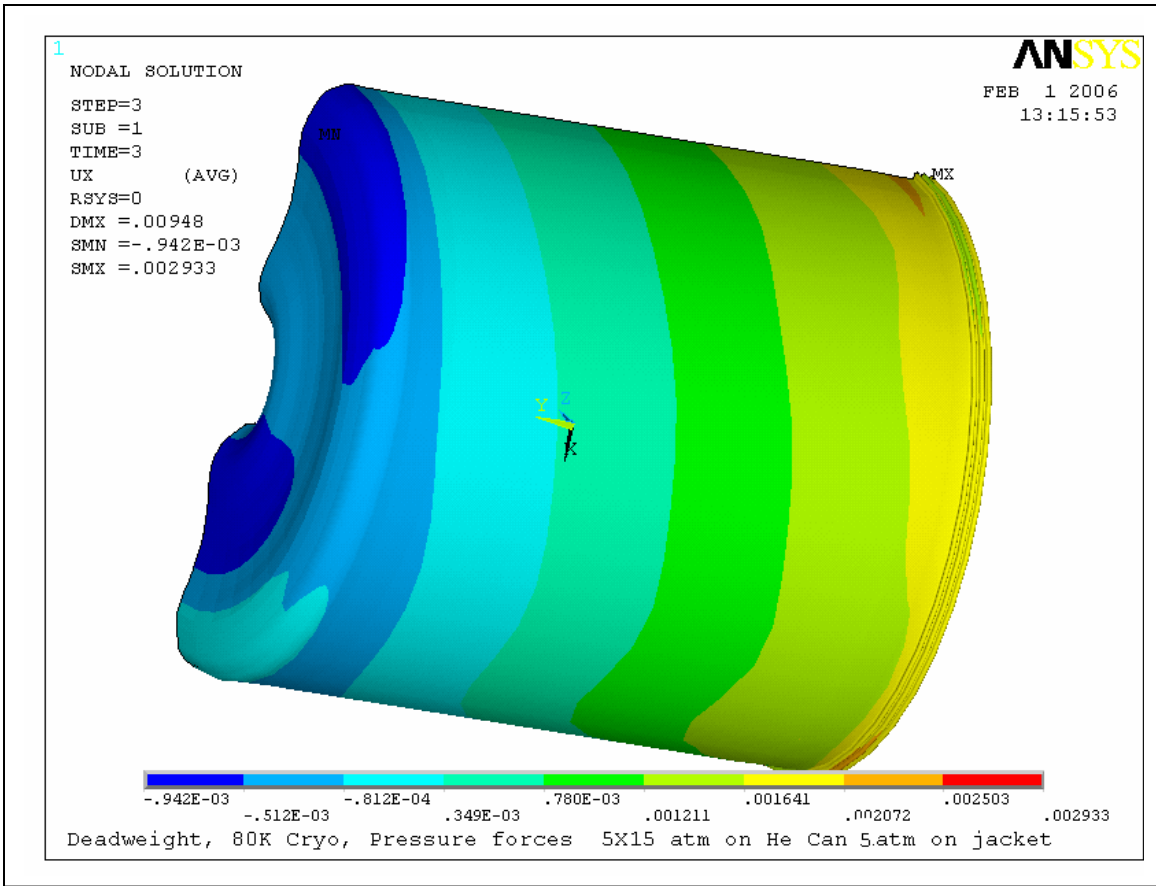
Eigenvalue Buckling Analysis, Load vector is 15 atm on cryostat and 1 atm on vacuum jacket.
Analysis includes
Thermal strains and pressure loads.



11.4 Vacuum Jacket Elastic Buckling (Large Displacement Method)

The vessel simulation was run with a large displacement solution and five times the pressure loading. The solution was bounded and this is an indication that elastic collapse of the shells is not possible.



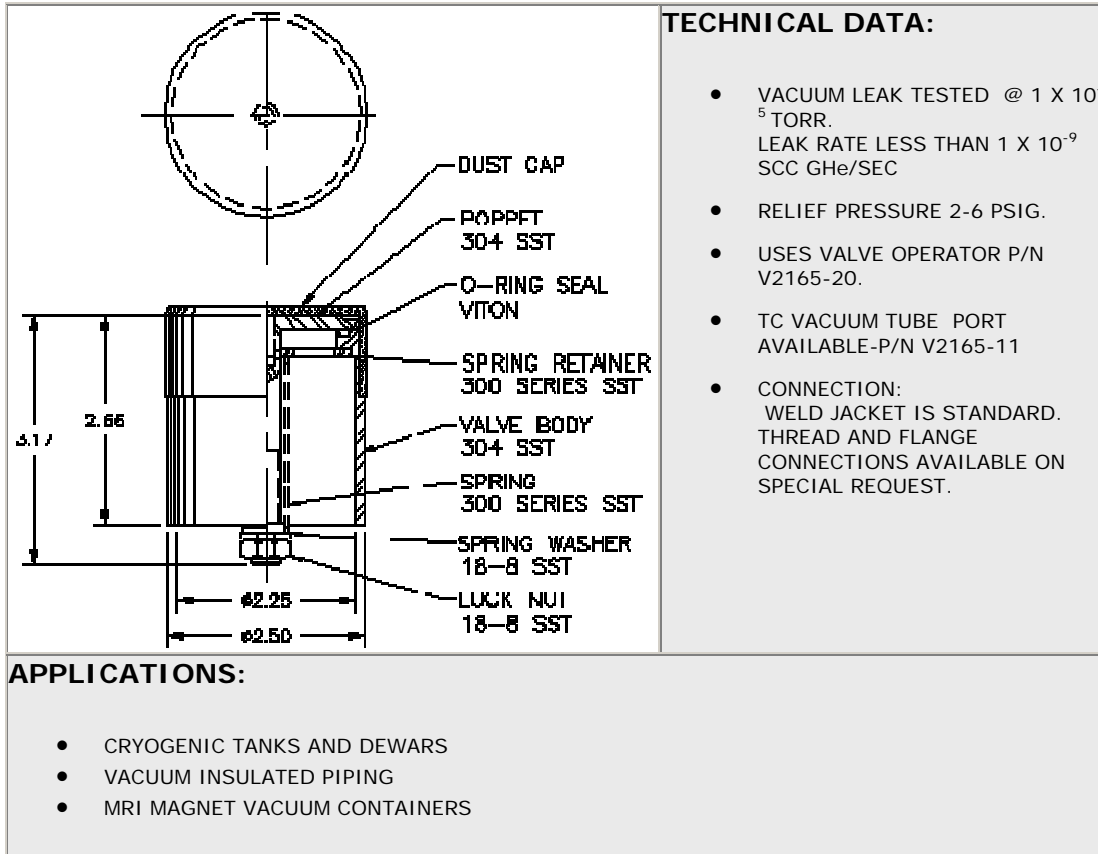


V2000 VACUUM SEAL-OFF AND OPERATOR SERIES

V2000 SERIES VACUUM SEAL-OFF VALVE

The V2000 vacuum seal-off valve is a high conductance, compact and highly reliable means of providing an evacuation entry into any large vacuum system. The valve incorporates a spring loaded poppet which provides relief in the event of an over pressure, protecting an inner vessel or neck seal. The poppet and spring assembly is installed after the body is welded to the vacuum space.

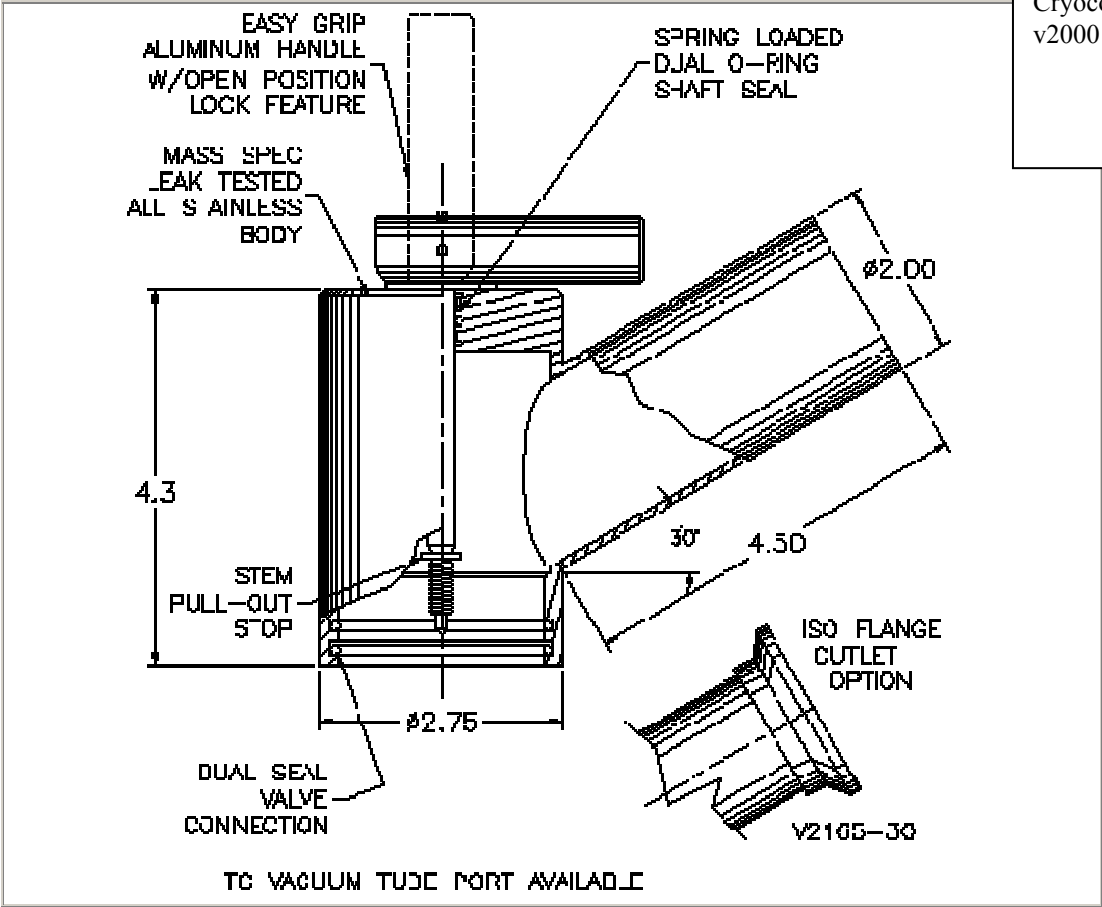
The V2000 is used with a valve operator to connect to a vacuum pump and is removed after the evacuation process is finished. The valve operator stem engages the poppet and the handle is toggled up to pull the poppet open and begin the evacuation. After the evacuation is complete, the handle is toggled down and the poppet re-seats providing a vacuum tight seal. The stem is then disengaged from the poppet and the valve operator is removed to complete the operation.



V2000 SERIES VALVE OPERATOR

The V2000 valve operator is the connection between the V2000 seal-off valve and the vacuum pump. This tool removes and replaces the seal plug during the pump down operation using a tight seal at the body junction and a spring loaded, redundant seal on the shaft. TC vacuum tube ports and ISO flange outlets are available as options.

Cryocomp
v2000



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Thermocouple type or TC Vacuum Gauge supplied by CVIP. Only good below 2 Torr

12.0 Bellows Design

There are four areas where bellows are employed to allow thermal differential contraction, and to add thermal conduction length. All of these will be specified in terms of required operating pressure, and lateral and axial displacements. The stress analyses presented in this section are intended to be an “existence proof” for the bellows space allocation. The CVIP consulted with the bellows manufacturer for appropriate convolution details, and the manufacturer provided qualifications of the bellows designs in accordance with the EJMA standards.

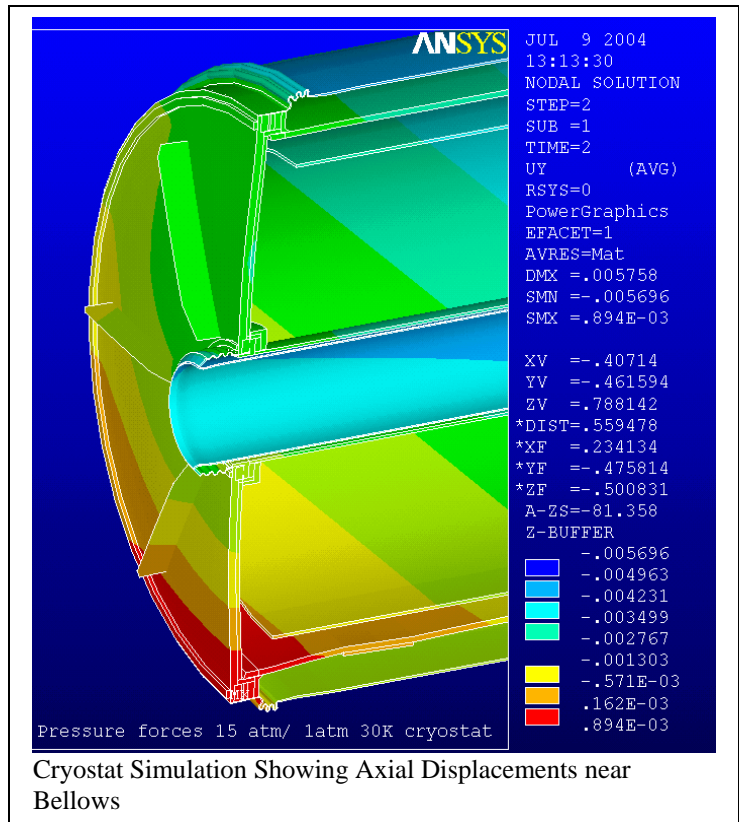
Table of Bellows Specifications Revision 2:

Item #	Location	Operating Pressure	Axial Displacement	Lateral Displacement	Axial temp Gradient	Material
BNL-002 #27	End of Bore Jacket	Vacuum	3mm	1 mm (Radial)	30K-292K	316 SST
BNL-002 #24	Cover end of Vacuum Jacket	Vacuum	3mm	1 mm(Radial)	30K-292K	316 SST
BNL-002 #30	Leads (Normal Operation)	15 atm	0.7 mm	.25mm	30K-292K	316 SST
BNL-002 #30	Leads (extreme Thermal Differential)	15 atm	3mm	1mm	30K-292K	316 SST
BNL-002 #40	Gravity Support Pads	Vacuum	~2mm	2.2mm	Only 292K	304 or 316 SST
BNL-002 #21	He/LN Outlet	15 atm	~2mm	2mm	~0 K	316 SST

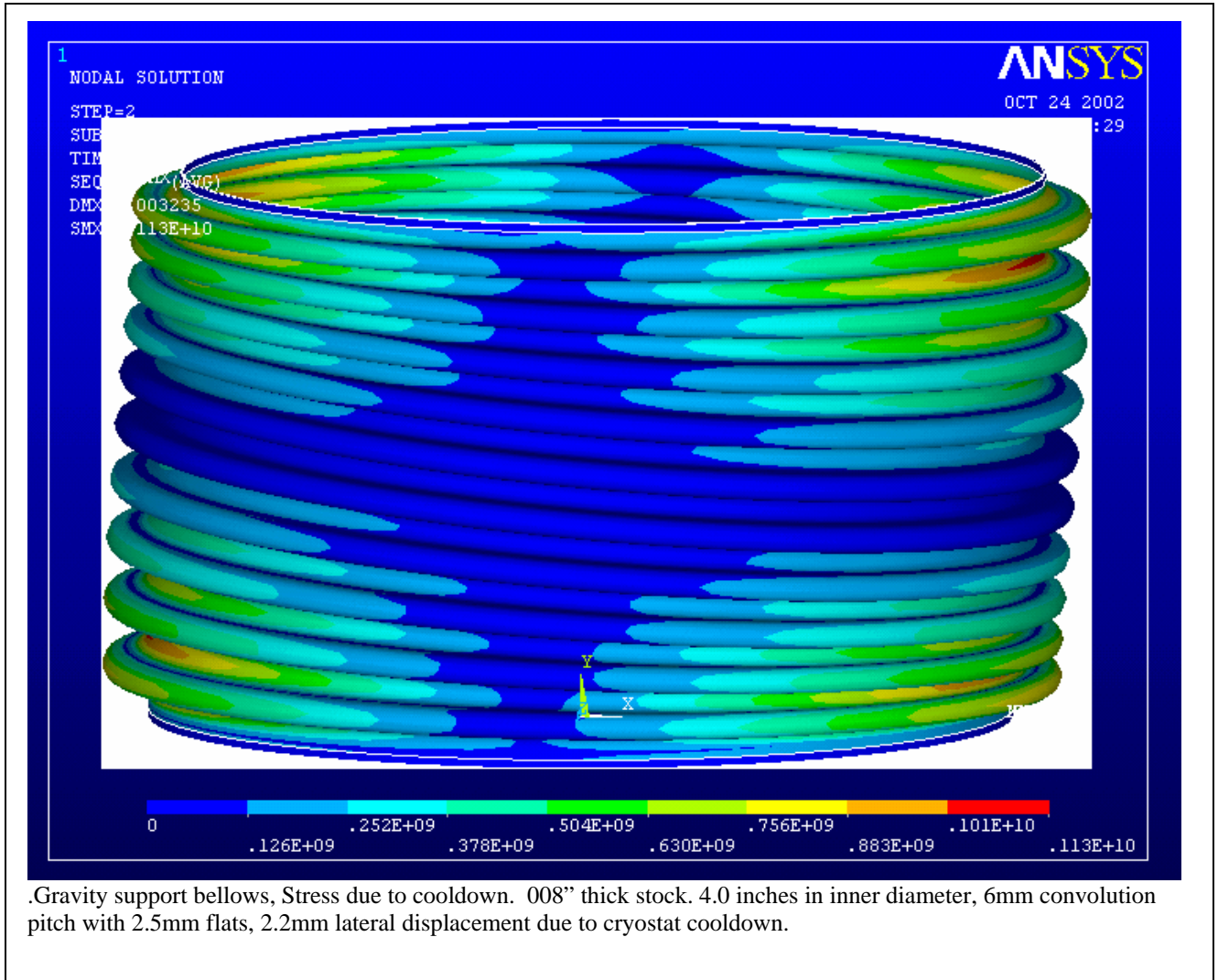
All bellows have a critical pressure at which they become unstable. Instability can occur in either of two modes, column instability (or squirm), or inplane deformation of the convolution side wall.

Squirm is the phenomena whereby the centerline of a straight bellows develops a sideways or lateral bow.

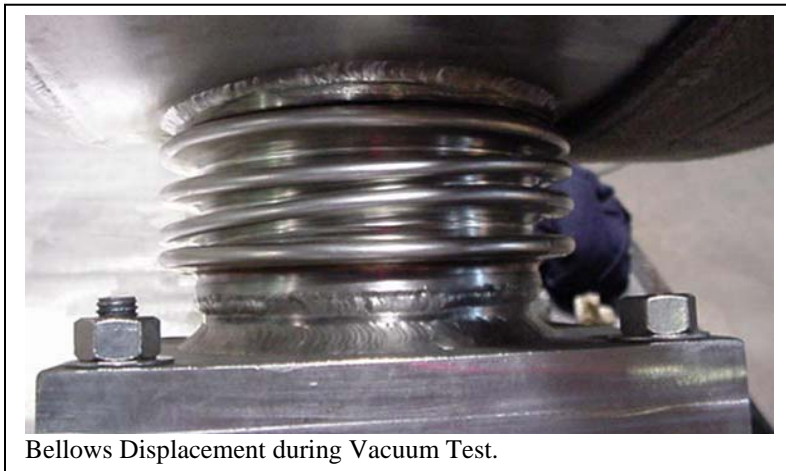
The critical pressure at which this instability occurs is a direct function of the diameter and spring rate, and an inverse function of the length. If the bellows is bent, or angulated, the centerline can begin to move away from the center of curvature. In each case, the effective length of the bellows increases, lowering the material available to withstand the pressure, thereby increasing the hoop stresses. As the length increases, the tendency to squirm increases and the stresses become higher and higher until catastrophic failure occurs. A simple way to visualize this phenomena is to remember that the bellows is a



cylinder of given volume. Internal pressure tries to increase a vessel's volume. Since a bellows is flexible in the axial direction, it can increase its volume by increasing the length of its centerline. With the ends fixed, it does so by simulating the appearance of a buckling column. In the BNL Pulsed Magnet, bellows "squirm" is stabilized with tension rods for the lead and LHe/LN2 bellows.



Bellows materials are specified as 316 for sub- LN2 temperature service, and 304 for RT service. Room temperature yields of highly cold worked materials should be in the range of 500 MPa for 316 and 1000 MPa for 304. All the bellows must see their pressure service at room temperature. The gravity support bellows worst loading is due to a lateral displacement due to cryostat contraction.



Operating conditions		
Internal pressure	Psi	Full vacuum
Temperature	Deg. F	60
Axial movement	in	+0.08 / -0.08
Lateral movement	in	.1
Angular rotation	deg	0
Cycle life		1000
Spring rate	lbs/in	<1000 or advise
Leak rate	atm-cc/sec	1x10 ⁻⁹
Outside pressure		ambient
Small weld end OD	in	.5
Small weld end thickness	in	.065
Small weld end length	in	.5
Large weld end OD	in	5.75
Large weld end thickness	in	.375
Large weld end length	in	.625
Over all length	in	3
Convolutions ID	in	min 4.87
Materials		304 stainless steel

Notes:

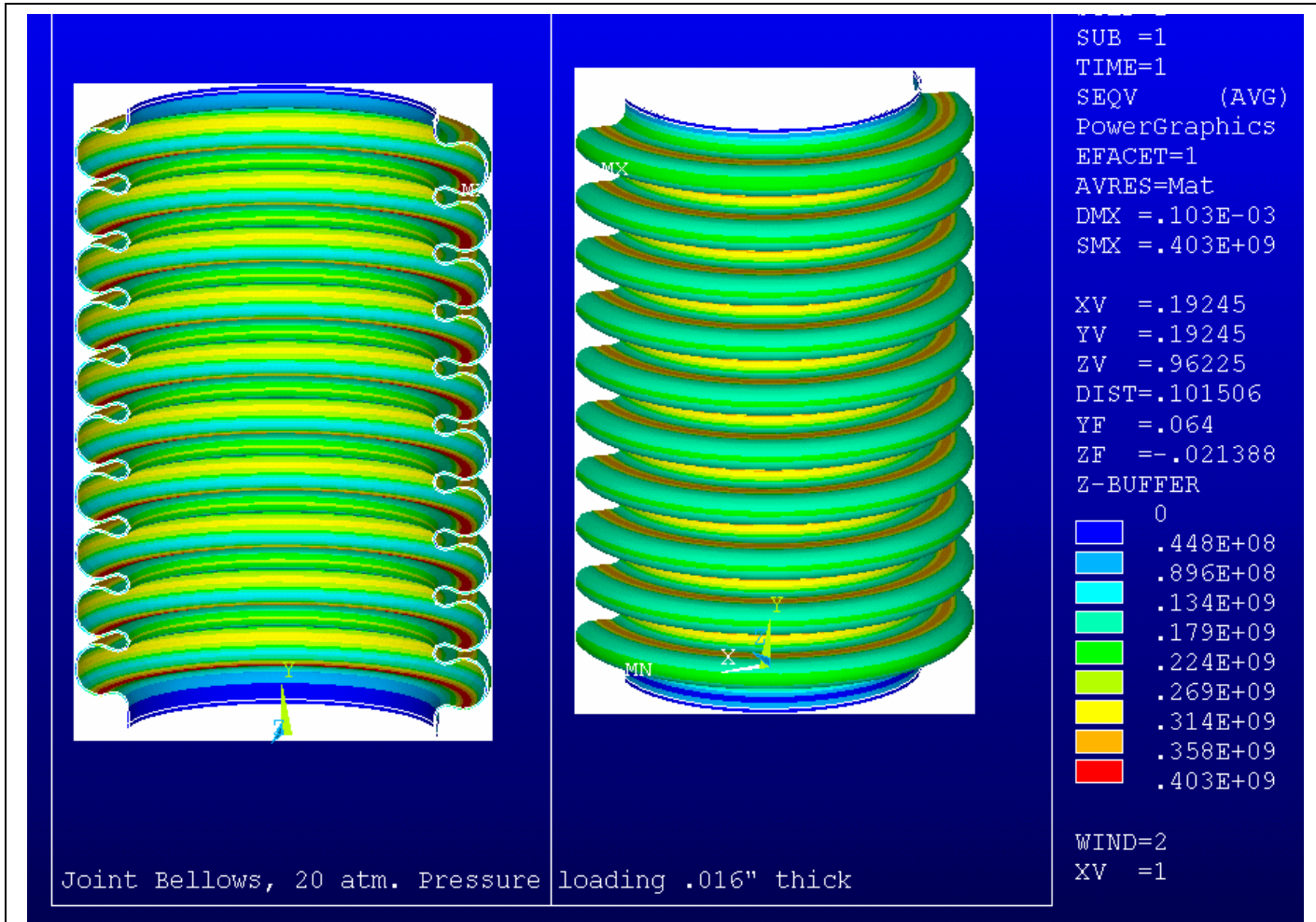
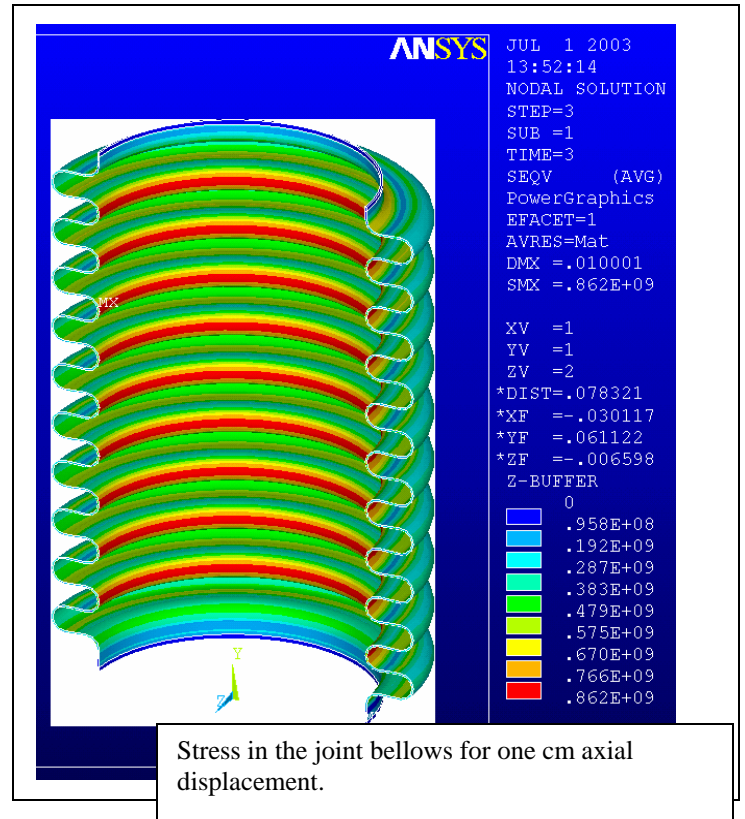
1. Clean for cryogenic service, plastic bag, test certification required.
2. All dimensions are in inches.



Repaired Bellows?

Joint and LN2/He Bellows.

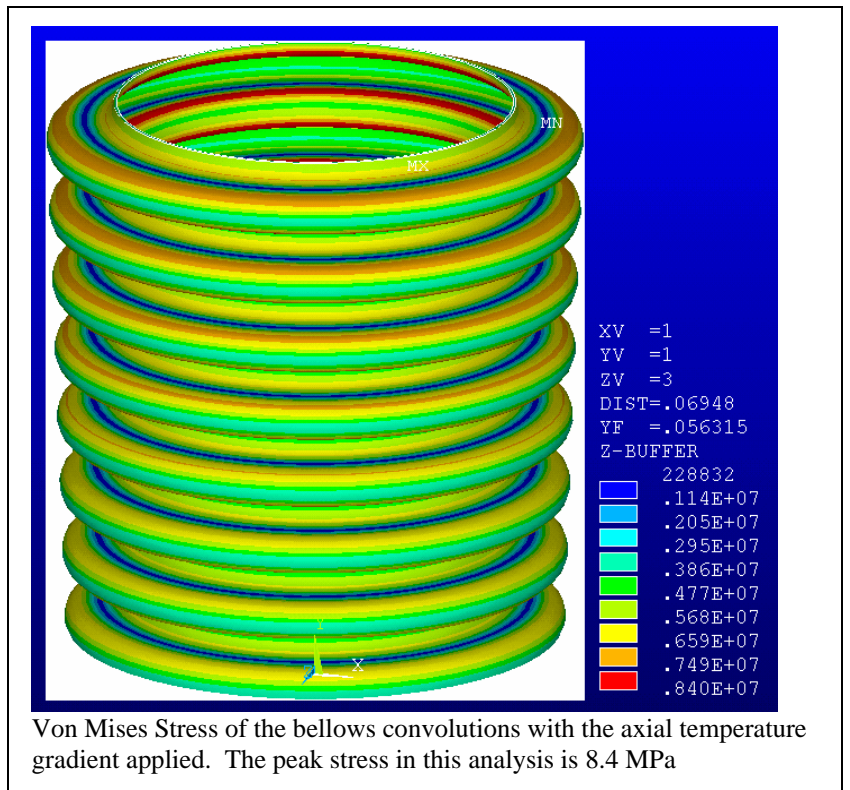
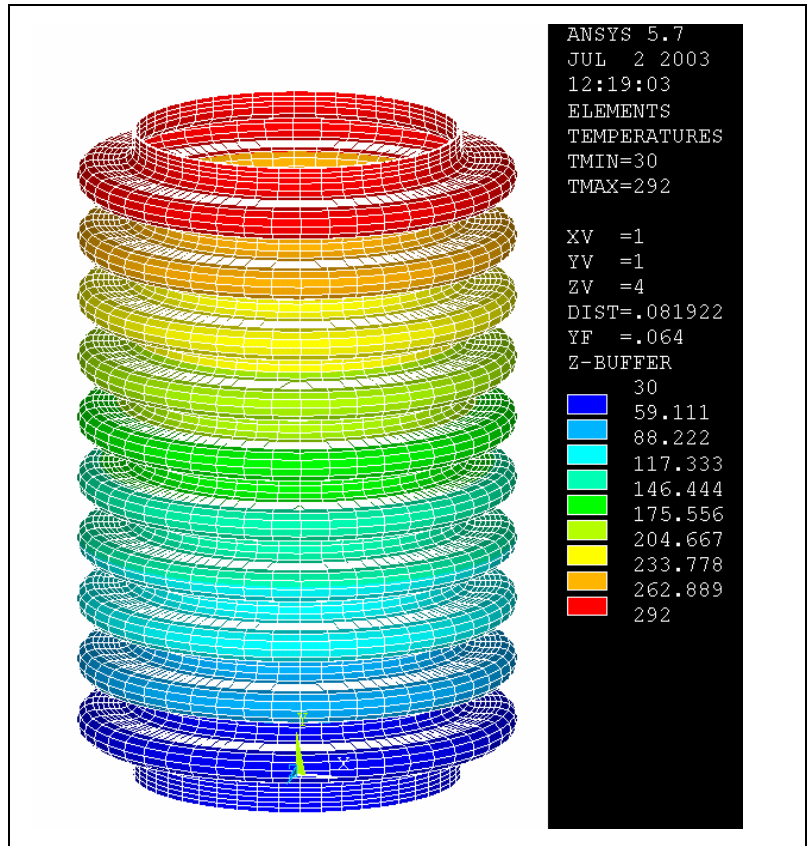
These bellows must accept the axial differential thermal motions between the 30 K cryostat and the magnet. Nominally this is the difference between cryostat, and the coil at 100K. Both Copper and Stainless Steel have coefficients near 10×10^{-6} , and the magnet/cryostat assembly is about 1 m. long. The intention is to have the plenum inlet side of the magnet and plenum to be held to the same displacement via spacer blocks, and the joint end of the magnet is spaced off the cover with a Belleville spring stack. The differential displacements thus appear in the joint bellows. Nominal differential motion is $1 \text{m} * 10 \times 10^{-6} * 70 \text{K} = .0007 \text{m}$ or 0.7mm. A worst case differential axial motion might occur if the cryostat was at 30 K and the magnet accidentally was heated beyond it's normal high temperature, to RT. The system interlocks should shut down the magnet current well before this. This would yield a differential axial motion of 2.62 mm. There is a radial differential motion. The outer joint at a radius of .35m would see a little over 1/3 the axial motion or about 1mm. Early in the design a large axial compression was needed to expose joint connections but the gland nut- penetration design does not require this.



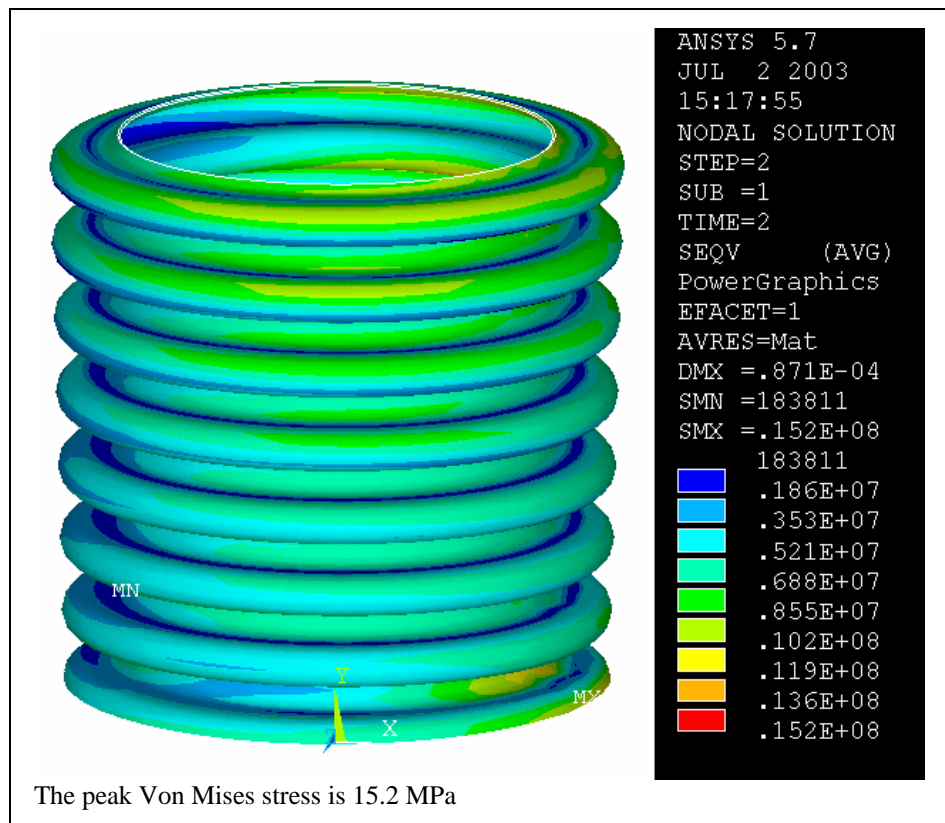
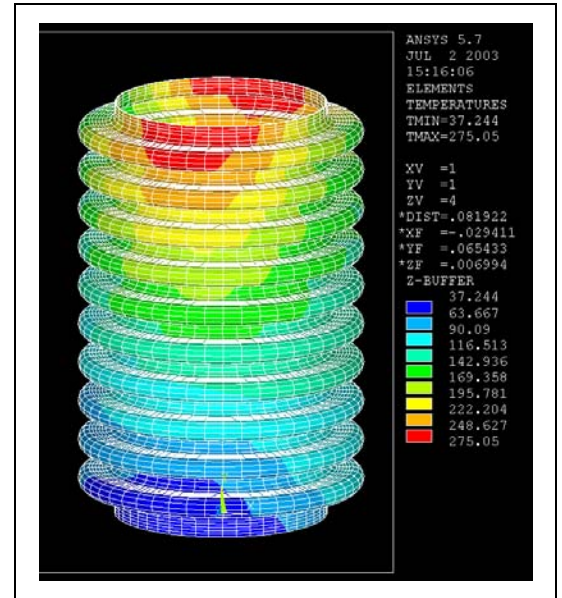
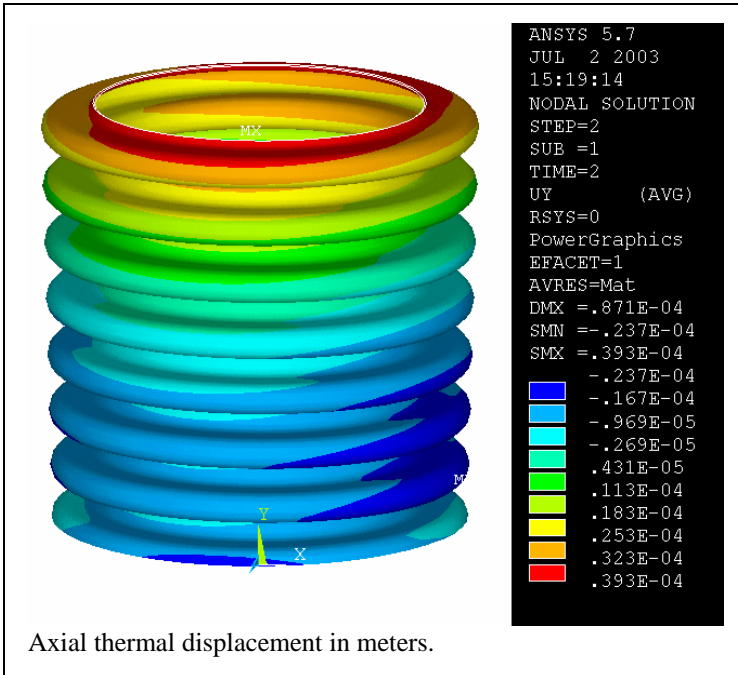
Bellows used for the Joint penetration. The overall length of this model is .127m or 5 inches. The convolution pitch is 12mm with 5mm flats. The inner diameter is 6.25 cm or about 2.5 inches. The stock thickness is .016"

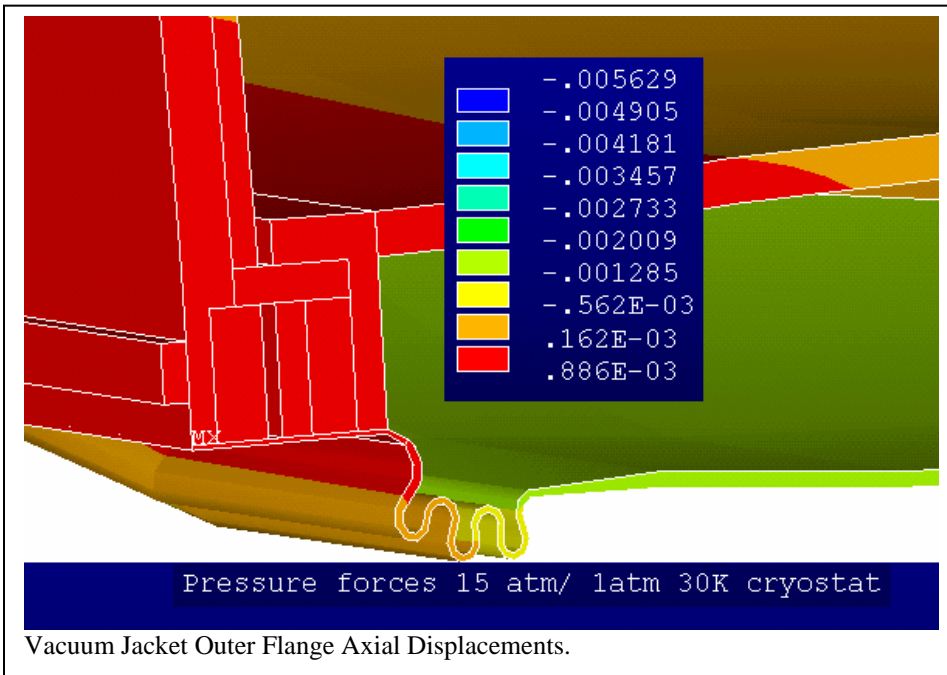
Joint Bellows Thermal Stresses

The bellows are used to absorb differential motions, and also to provide long thermal paths to minimize heat leaks – See section 14.0. The joint bellows model was analyzed with a 30K to RT (292K) axial gradient. With this linear axial thermal gradient, appropriate for conduction from the 30K cryostat cover to the LN2 or RT end of the joint break-out, the stresses are minimal.



An attempt was made to “invent” a temperature distribution that wasn’t quite so uniform. This is shown at right. The axial (vertical in these plots) displacement is not uniform. . The peak stress for this non-ideal temperature gradient is 15.2 MPa, up from the previous 8.4 MPa, but it is clear that thermal stresses are not critical for the temperature gradients we expect at the joint penetrations.







For Reference Only

American BOA, Inc.
Expansion Joint Division

Quotation#: 1-44304
Customer: CVIP
Date: 8/20/2004
Prepared By: PV
Exp. Jt. Item #: *dwg # 23 0003*

BELLOWS DESIGN CALCULATION

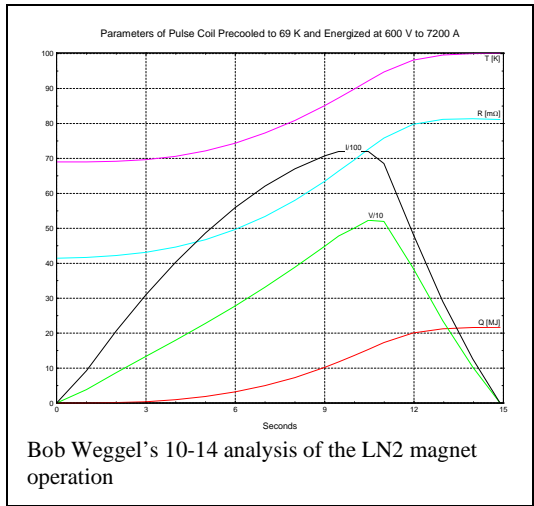
Author: M. Lanz

Revision 12, 7/22/04

Design Basis:	The Expansion Joint Manufacturer's Association Standard, 8th Edition, 2003	
Allowable Stress Basis:	ASME Sect. II, Division D, 2001 Ed., 2003 Add.	
Bellows Element Geometry		
Bellows Material:	SA240-T316	
Collar Material:	N/A	
Bellows Inside Diameter (in.)		5.00
Bellows Outside Diameter (in.)	H= 0.600	6.22
Number of Convolutions		4
Individual Ply thickness (in.)		0.012
Number of Piles		1
Bellows Element Length (in.)	2r= 0.250	1.88
Bellows Effective Area (in. ²)		24.7
Design Information		
Condition Type (i.e design, upset)		Design
Design Pressure (Psig)		-15
Design Temperature (deg. F)		60
Axial Compression (in.)		0.080
Axial Extension (in.)		0.080
Lateral (+) (in.)		0.100
Lateral (-) (in.)		0.000
Angular (+) (deg.)		0.00
Angular (-) (deg.)		0.00
Bellows Stress Analysis		
	Actual	Allow.
	Stress	Stress
S1 Tangent Circumferential Membrane Stress Due to Pressure, psi	3,133	20,000
S'1 Collar Circumferential Membrane Stress Due to Pressure, psi	N/A	N/A*
S2 Circumferential Membrane Stress Due to Pressure, psi	1,585	20,000
S'2 Reinforcing Ring Membrane Stress Due to Pressure, psi **	N/A	N/A**
S3 Meridional Membrane Stress Due to Pressure, psi	397	N/A
S4 Meridional Bending Stress Due to Pressure, psi	11,975	N/A
S3+S4 Meridional Mem. + Bending Stress Due to Pressure, psi	12,372	60,000
S5 Meridional Membrane Stress Due to Deflection, psi	1,629	N/A
S6 Meridional Bending Stress Due to Deflection, psi	207,353	N/A
Maximum Design Pressure Based Upon Squirr, psig	82	
Fatigue Characteristics		
Total Stress Range for All Movements (St), psi		217,643
Fatigue Life (cycles to failure)	EJMA	3,882
Expansion Joint Spring Rates		
Axial Spring Rate (lbs./in.)		251
Lateral Spring Rate (lbs./in.)		2,623
Angular Spring Rate (in.-lbs./deg.)		17
Torsional Spring Rate (in.-lbs./deg.)		37,716
	(Maximum Allowed Torsion = 0.062 deg.)	
Pressure Thrust at Design Pressure (lbs.)		-371
* Collar not utilized		
** Un-reinforced bellows, reinforcing rings not utilized		
	Bellows P/N	0

13.0 Cryostat Eddy Current Analysis

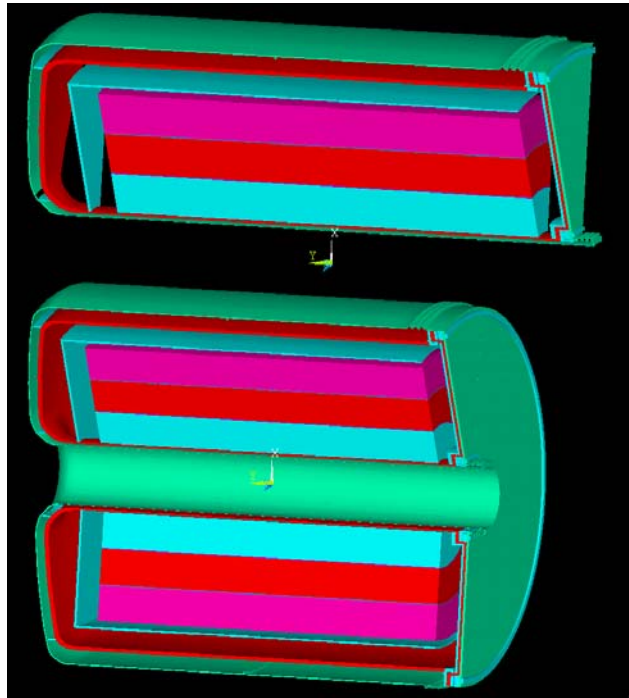
There is a concern that pulsing the magnet can induce eddy currents in the shells sufficient to cause a significant load. The eddy currents coupled with the large bore field, may significantly load the bore vacuum jacket. In addition the flat plenum plate does not have a curvature to stabilize it and would see bending stresses rather than hoop membrane stresses. To address this concern, an electromagnetic transient analysis was performed. The solution is a vector potential solution, That traces the current history at the right. The worst point will be the beginning of the ramp-down. An additional concern is the field loss due to the opposed currents in the bore shells, but this field loss is of the order of a few milli-Tesla. Stress Levels also turned out minor – of order 2 MPa. The ANSYS Input listing is included at the end of this section.



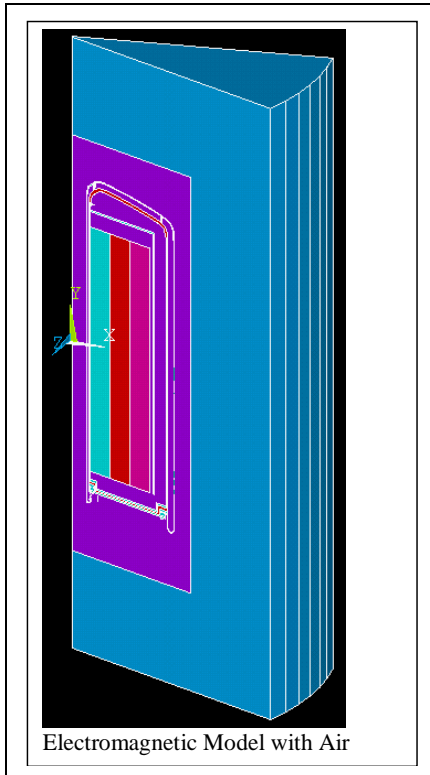
Resistivity Values for Metals

	%IACS	Ohm-m	Reference
OFHC Copper	100	1.7074e-8	
Stainless Steel 316		74e-8	MHASM1
Inconel 625, at T=292 deg. K		103e-8	

MHASM2=ASM Metals Handbook--Volume 2, Tenth Edition

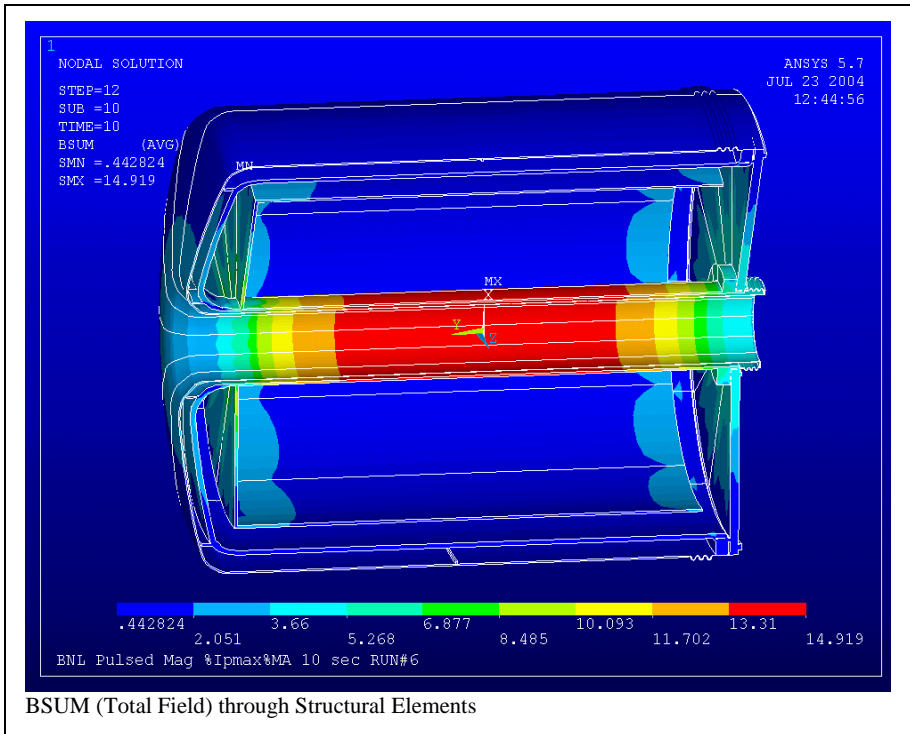


Structural Mode (Sub-Set of E-M Model) Shown with Coils that are removed for structural analysis. The upper plot is the 30 degree cyclic symmetry model, below is a 7 segment symmetry expansion.

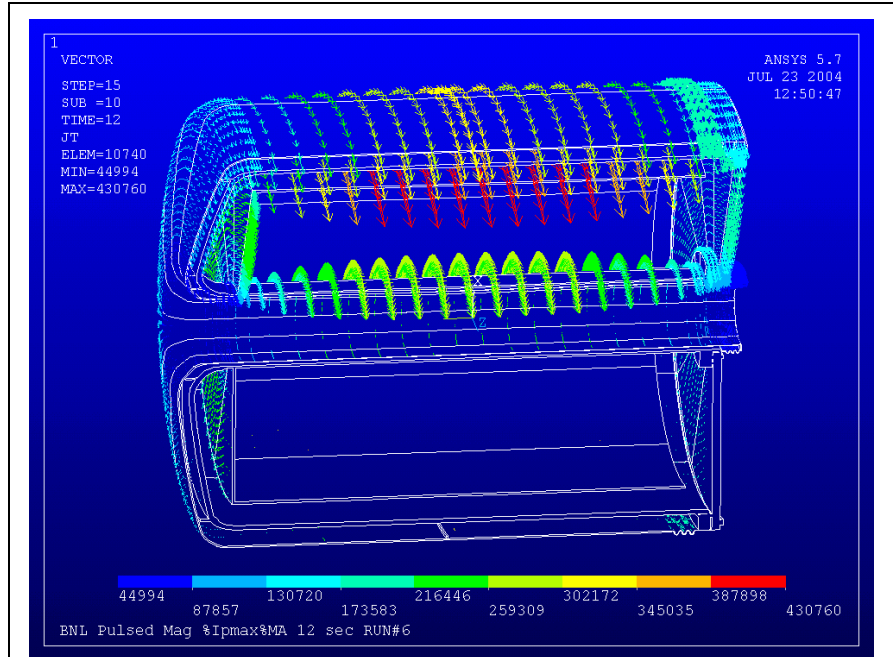


Electromagnetic Model with Air

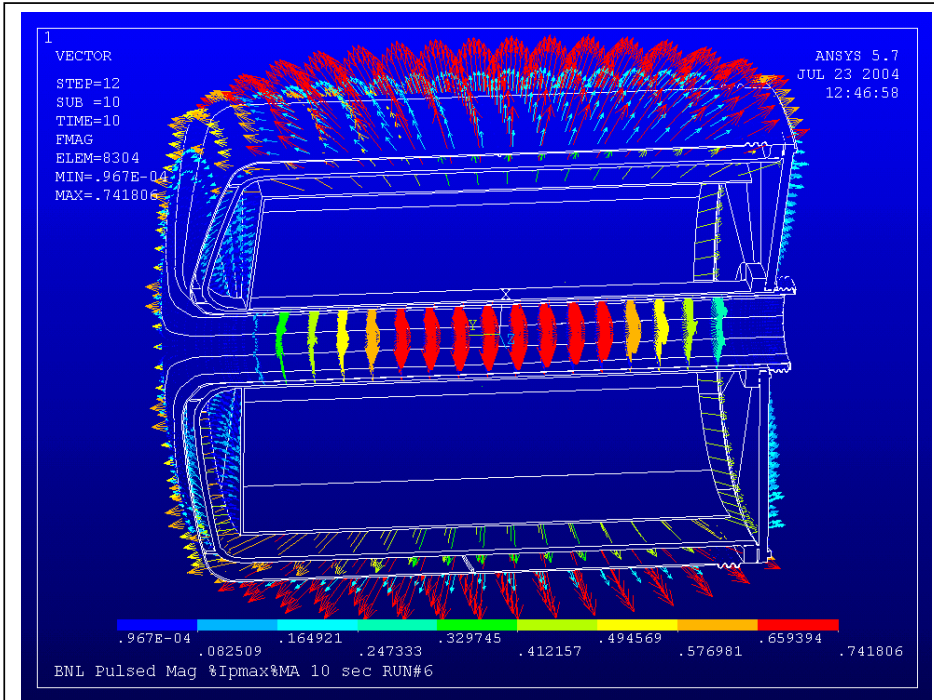
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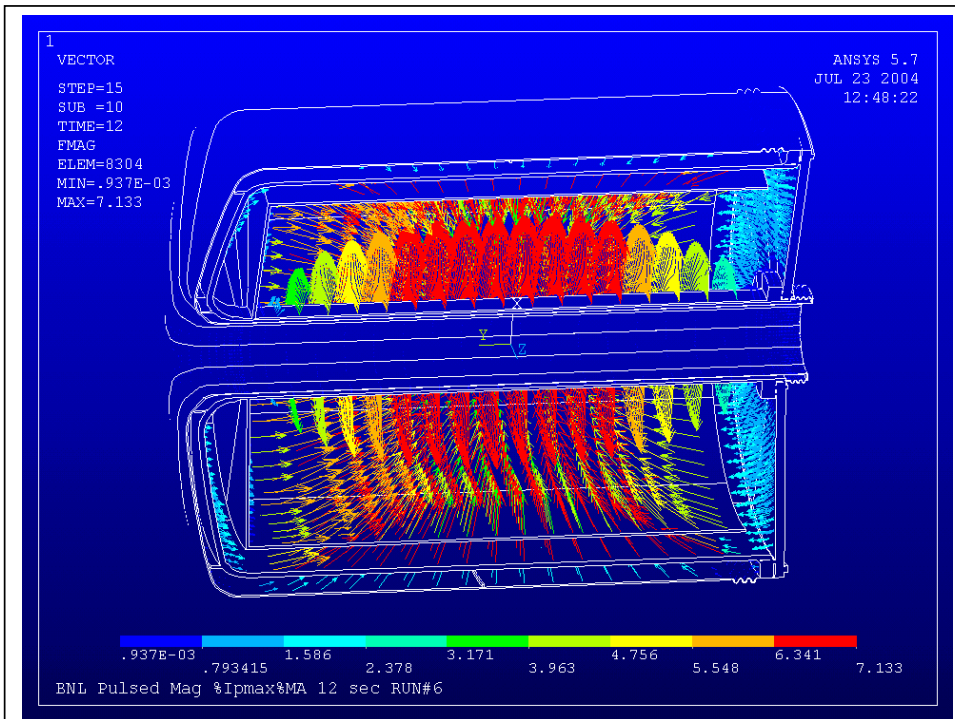
BSUM (Total Field) through Structural Elements



Eddy Currents At Max Stress Point. –Actually During Ramp-Down



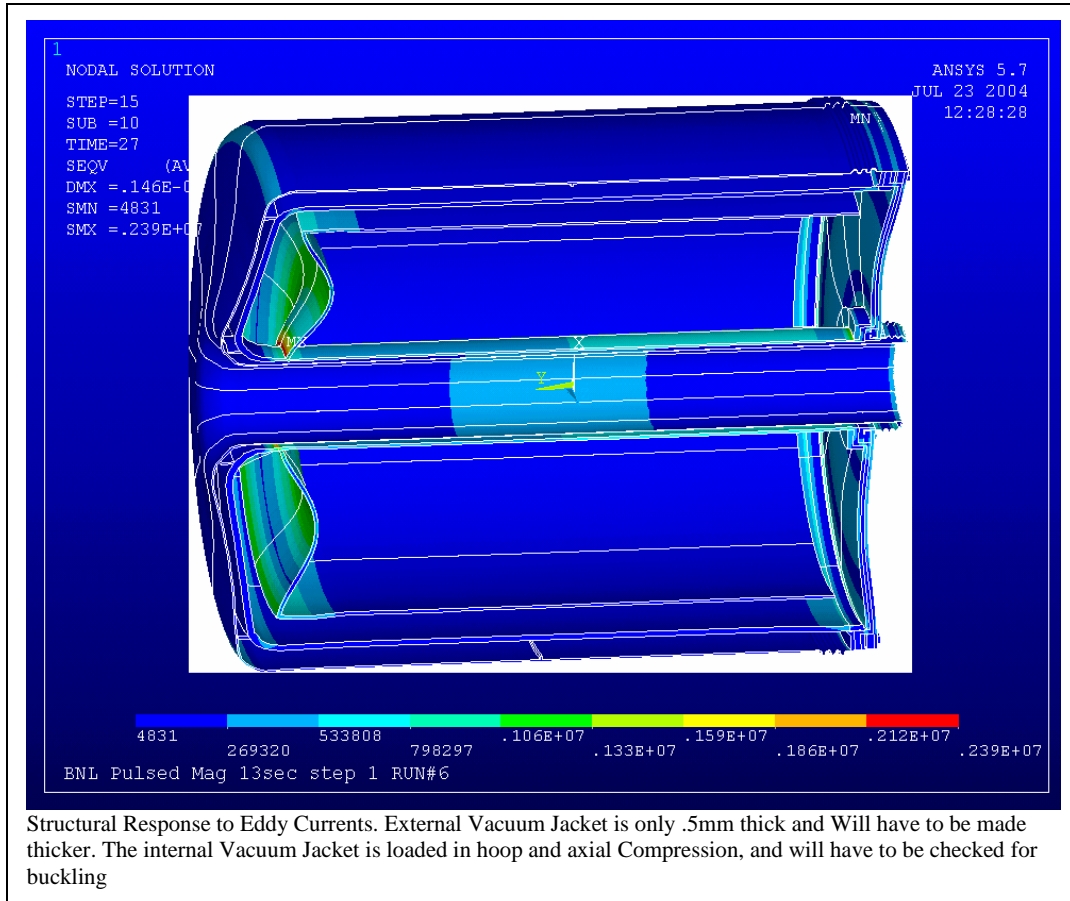
Eddy Current Forces During Ramp-Up



Eddy Current Forces During Ramp-Down

Cryostat Eddy Current Analysis – Stress Pass

The nodal Lorentz forces are produced in the electromagnetic run and can be read into the structural model with an LDREAD ANSYS command. This was done. $j*B*r$ should give an estimate of the hoop stress in any of the cryostat shells. With current densities of $.2MA/m^2$ and fields of 10 Tesla, and radii of $.1$ to $.5$ meters, the hoop stress should be only $.2MPa$. The peak stress is a bending stress and occurs in the plenum plate.



Structural Response to Eddy Currents. External Vacuum Jacket is only $.5mm$ thick and Will have to be made thicker. The internal Vacuum Jacket is loaded in hoop and axial Compression, and will have to be checked for buckling

```

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/com
/com

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/PREP7
antype,trans
/COM 11-25-98 Electrical/Thermal
analysis
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current regions
ET,2,97,0 ! (Ax, Ay, Az) EM for air
ET,3,97,0 ! (Ax, Ay, Az) EM for air
TREF,292.0
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/com MAT,10
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*end
*use,matt
/com
/input,vma6,mod
/com
nall
eall
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CSYS,12 $NROTAT,all
esys,12
/com Coupled Flux Condition

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d,all,ax,0.0 ! Added 2-2-98
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nset,z,1.75,2
nset,z,-2.0,-1.75
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d,all,ay,0.0
d,all,az,0.0 !added 2-2-98
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nall
eall

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nelem
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Wsort,y

```

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nelem
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eall
nall
save
fini

/solu
      Transient Current Parameters
t1= 0.100000 $i1= 10.0 $p1= .01e6
t2= 1.000000 $i2= 800.0 $p2= 1.0e6
t3= 2.000000 $i3= 2000.0 $p3= 1.0e6
t4= 3.0000 $i4=3100.0 $p4= 1.5e6
t5= 4.000000 $i5=4000.0 $p5= 1.0e6
t6= 5.0000 $i6= 4900.0 $p6= .5e6
t7= 6.0000 $i7= 5650.0 $p7= .001e6
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t15=13.000000 $i15= 3000.0 $p1= 0.0e6
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time,.01
/com Electromagnetic Solution
estep=estep+1
solve
save

! SETUP TRANSIENT MACRO
*create,load
/com solve electromagnetic problem
time,timpt
csys,0
curd= i%ls%*6375
esel,mat, 1
easel,mat, 3
easel,mat, 5
bfe,all,js,1,0,0,curd
eall
nall
solve
save
*end

! START TRANSIENT

*do,1s,1,15,1
timpt=t%ls%
curamp=i%ls%
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*use,load
*enddo
fini

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/prep7
antype,static

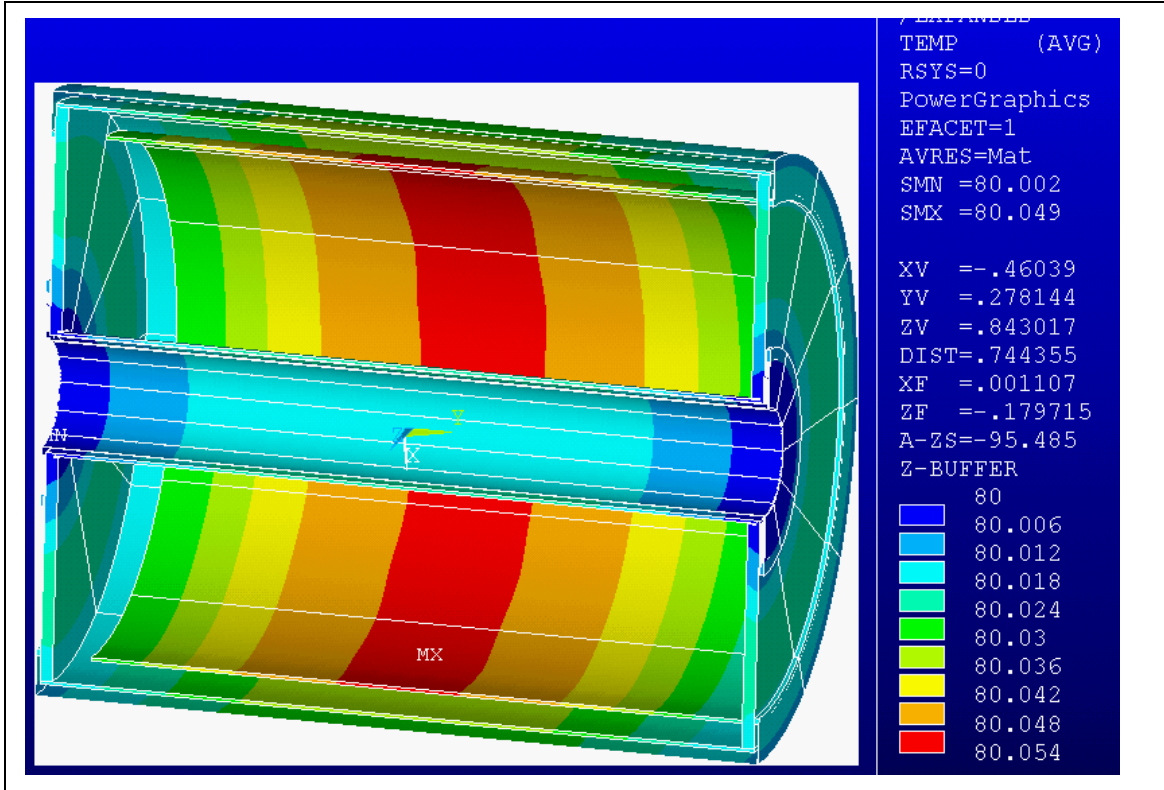
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et,2,45
ex,1,70e9
ex,2,70e9
ex,3,70e9
ex,4,70e9
ex,5,70e9
ex,6,70e9
ex,7,70e9
ex,8,70e9
ex,9,70e9
ex,10,200e9
ex,11,200e9
ex,12,200e9
ex,13,200e9
ex,14,200e9
LOCAL,12,1,0,0,0,-90.0,0.0
CSYS,12 $NROTAT,all
nset,y,-16,-14.99
naset,y,14.99,16
d,all,uy,0.0
nset,z,-.006,.006
d,all,uz,0.0
nall
eall
save
fini
/solu
*do,ldnum,1,15
ldread,forc,ldnum,,,elect,rst
eusel,mat,90
eusel,mat,91
eusel,mat,92
eusel,mat,1
eusel,mat,2
eusel,mat,3
eusel,mat,4
eusel,mat,5
eusel,mat,7
nelem
solve
*enddo
save
fini

/exit

```

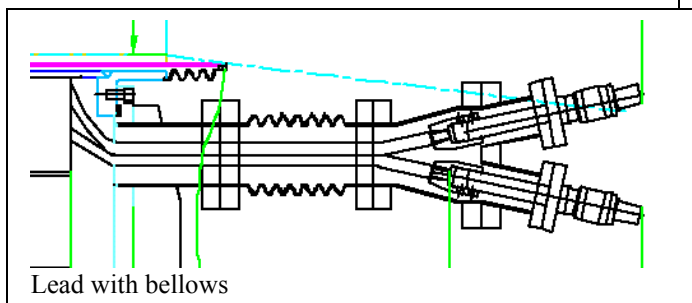
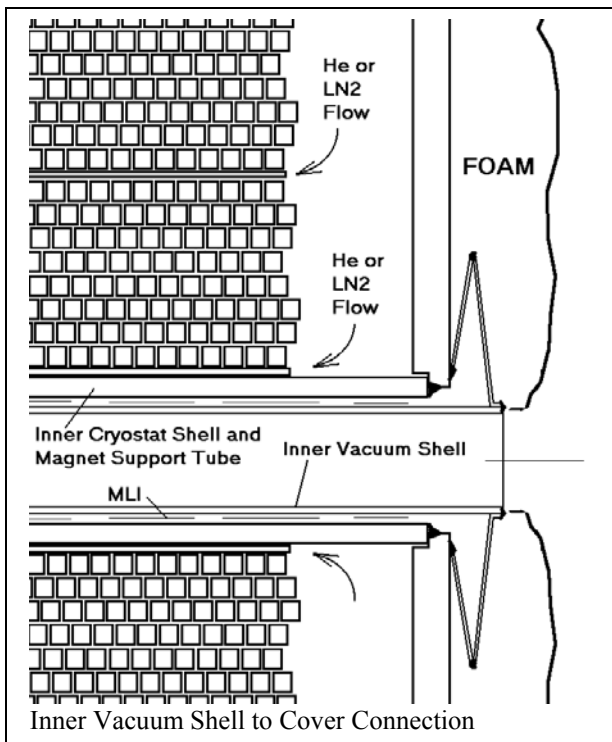
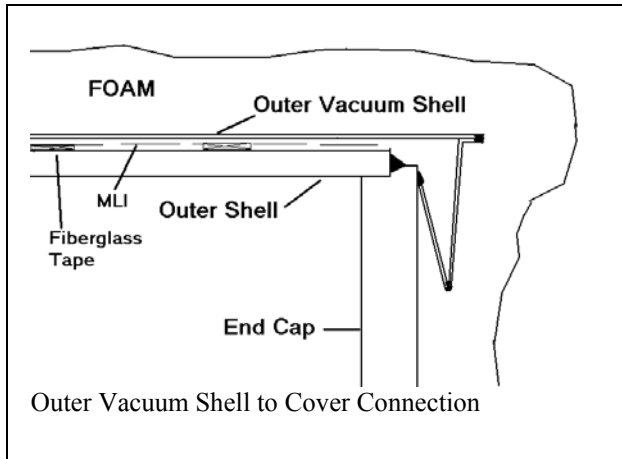
Cryostat Eddy Current Analysis – Temperature Pass

Heat-up due to the eddy current loading on the cryostat produces less than 1 degree K



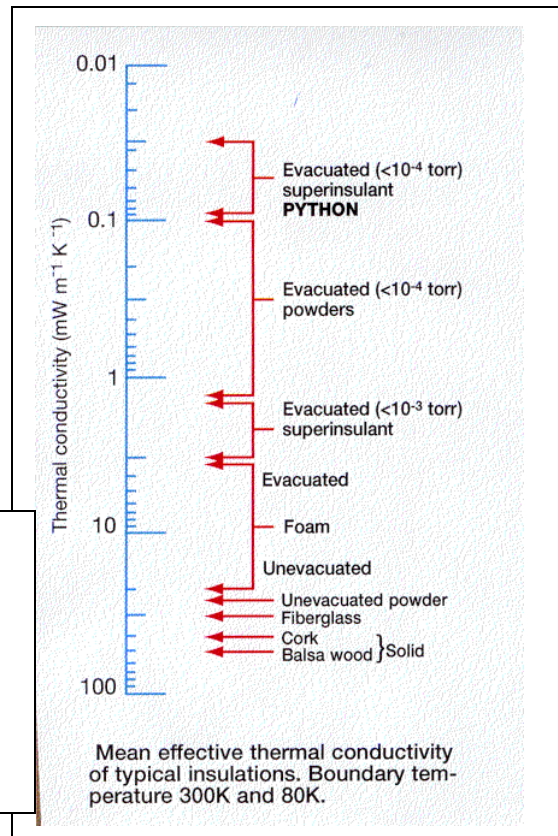
14.0 Steady State Heat Gain.

The specification requires that the cryostat heat gain should be <200 W at 22 K; i.e., it should boil off no more than ~ 500 liters/day of liquid hydrogen. The specification also states that liquid nitrogen is available for cooling the leads to the magnet. –This implies that the 200 watt limit for the cryostat excludes the leads.



A concept which has a 234 watt heat gain has been developed that employs vacuum at the outer and inner shells and foam at the ends around fluid and electrical penetrations. This can be reduced to the required heat leak by adding vacuum jackets to the end flanges and/or increasing the foam thickness beyond 10 cm.

Because of the space limitations at the bore, only a vacuum insulation system has been considered here. The possibility of using foam insulation has been investigated on the remaining surfaces of the cryostat. Initial estimates of the heat gain on the outer shell of the cryostat were excessive if only foam were used, so a vacuum shell is proposed on the OD as well. If piping penetrations are moved to the end plates, it is relatively easy to add vacuum shells on the ID and OD. The connection between the room temperature vacuum shell and the 22 to 30K end caps also conducted too much heat, so the foam has been extended to the outside shell. To clear the experimental bore volume foam can't be used at the ID so here the connection must bridge between room temperature and the cold



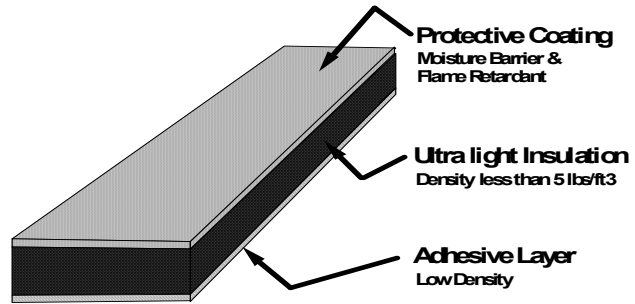
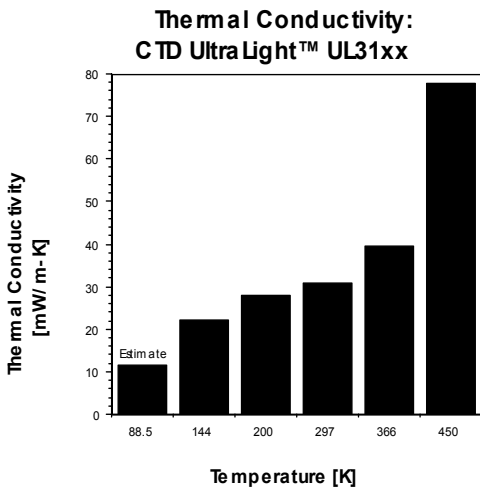
Helium/LN2 can. One virtue of this arrangement is that the magnet can be supported off the inner cryostat shell, and the system gravity supports can reach through the foam or vacuum boundary. In the heat gain calculations the gravity support is simply modeled as four G-10 columns or straps. With the magnet supported off the inner tube, the annular gap used for coolant flow will be readily maintained. If the magnet was supported independent of the inner cryostat tube, the alignment of the magnet to cryostat would be critical to maintain the 2mm annular space. For the heat transfer calculations, the cryostat is assumed to be a cylindrical geometry about 1.2m long and a meter in diameter.

Heat Gain Summary

Component	Material	Thermal conductivity W/m/degK	Area m ²	Length m	delta T	Heat rate watts
Inner shell vacuum with mli	Vacuum/MLI	*	.75398224	*	292-22	<20
Inner shell vacuum extensions	.0005m thick sst	16.27	6.283e-4	.2	292-22	13.8
Outer shell (foam option)	CTD Cryo foam insulation	.03	3.77	.1	292-22	303
Outer shell foam in series with vacuum+mli	Cryo foam insulation	.03		.1	292-220	49**
Outer shell Vacuum Extension	sst	16.27	3.14159e-3	.2	292-220	18.4
End Cover foam (1 ends)	CTD Cryo foam insulation	.03	1.508	.1	292-22	62.85
Dished head End Cover	Vacuum/mli					
Leads	Copper (22 to 80K)	396.5	8.64e-4	.4	22-80	49.6 (3 pairs)
Leads	Copper (80 to 292K)	396.5	5.4569e-4	.4	80-292	114.7 (2 pairs)
Lead bellows	sst	16.27	4.7124e-4	.4	292-22	5.33
Coil Support pads	g-10	.15	.0016	.05	292-22	1.296
Total bold red						220

* Radiation heat gain at bore= 37.281177 watts (no MLI) Stefan Boltzman Constant = 5.668e-8 watts/m²/degK⁴ $q_{rad} = area * emis * stefboltz * (trt^4 - tcold^4)$, emis=.12 polished sst From ref [8]: page 152. the heat flux should be divided by the number of MLI layers, conservatively it was divided by 2 – many more layers are practical in this space. – See the mli specification at the end of this section.

** Radiation and Foam conduction in series. The intermediate temperature (128.5K) of the vacuum shell was found by trial and error assuming a temperature and matching the heat flux for radiation and conduction.



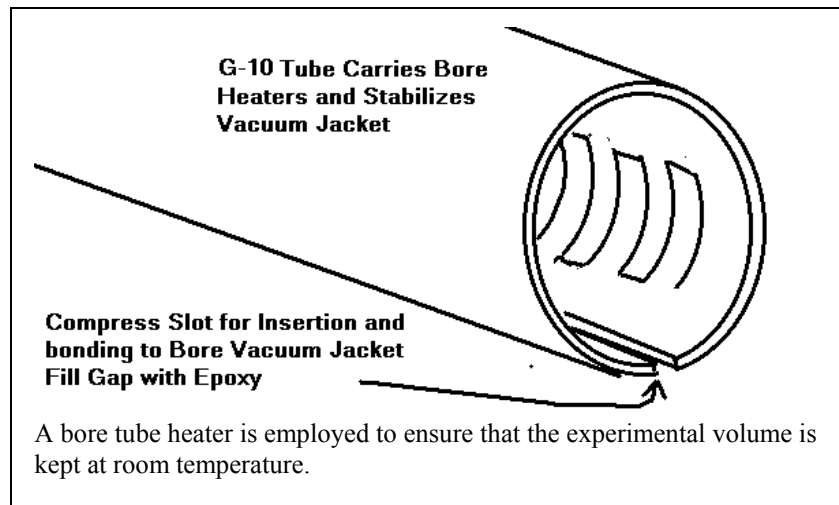
Cryostat Bore Tube Geometry

From the specification: “The Cryostat bore, including heater wire or tubing to keep it warm, should be at least 15 cm. If the bore tube is so long that its upstream end is >60 cm from magnet midplane, flare it conically at half angle of 125 mr.”.

Building from the Magnet ID and working towards the centerline:

Component	thickness	radius
The ID of the magnet winding		.15-.98/2= .101
Coolant Channel	.002	.099
Cold Cryostat Shell	.004762(3/16in.)	.094237
Vacuum Space	.008	.086237
Vacuum shell	.0005	.085737
Strip heater	.001	.084737

This leaves a clear bore diameter of .16947m, well above the .15 m required. These dimensions may be optimistic. More thickness may be required for the MLI or for the cryostat tube that supports the magnet weight.



MULTI-LAYER INSULATION (MLI) Specification from: “LHC Interaction Region Quadrupole Cryostat”, Thomas H. Nicol, Fermi National Accelerator Laboratory December 3, 1998, P.O. Box 500, Batavia, IL 60510, ref [14]:

“The stack height of each 32-layer blanket is 8.86 mm, with a mean layer density of 3.61 layers per mm. The blanket design incorporates 32 reflective layers of double-aluminized polyethylene terephthalate (PET) film. The reflective layers consist of flat polyester film aluminized on both sides to a nominal thickness not less than 350 angstroms. The spacer layers consist of randomly oriented spunbonded polyester fiber mats. The mean apparent thermal conductivity of an MLI blanket comprised of these materials has been measured to be 0.52×10^{-6} W/cm-K.”

MLI insulation should be applied to the outside of the BNL Helium cryostat. 32 layers is overkill. A few (5 to 10) is sufficient. A stainless steel screen ring

15.0 Cooldown

First, Some physical properties and characteristics of the magnet:

Working Fluid Properties

Fluid	Density	Sensible Heat, Liquid Phase	Sensible heat Gaseous Phase	Heat of Vaporization
Units		Joule per Kg degK	Joule per Kg degK	Joule per Kg at 77Kat1atm
Helium		4545 at boiling pt.	5190	20280
LN2	0.8043@78K Kg/l		2042	199100 (77K, 1atm)
Gaseous N2 @ 80K	4.44kg/m ³			
Gaseous N2 @RT	1.2506Kg/m ² @stp		1040 RT?	

Magnet Quantities

	Segment 1	Segment 2	Segment 3	Total
Volume (m ³)	9.2362824e-2	.15393804	.21551326	.4618
numturn	624	624	624	
weight (kg)	748.04651	1246.7442	1745.4419	3740.232

Magnet Flow Area Properties:

The coolant channel flow area is $(.1+.2+.3+.4)*2*\pi*.002 = .0126\text{m}^2$ or about 39 square inches
 The wetted perimeter of the channels is $(.1+.2+.3+.4)*2*2*\pi = 12.6\text{ m}$ (ignoring the 2mm ribs)
 The channel surface area is 12.6 multiplied by the one meter length of the coil or 12.6m^2
 The magnet surface area is $(.1+.2*2+.3*2+.4)*2*\pi = 9.42\text{m}^2$
 The Hydraulic diameter of the channels is $4*.0126/12.6 = .004$

15.1 Initial Cooldown using LN2

The specific heat of copper at 100K is 2183978 Joule/m³.
 The density of copper is 8930 kg/m³
 The initial cooldown to 77K requires:

$(292-77)*.4618*2183978/(199000+2042*(180\text{K}-77)) = 530\text{ Kg}$ of LN2 assuming an exhaust gas temperature of 180K

15.2 Cooldown Between Shots

Phased implementation of cooling systems is expected for the project. Within the coil, two cooldown methods are being investigated: an option using liquid nitrogen is discussed in section 15.3 and an option which uses helium gas is discussed in section 15.2. It is expected that design of the magnet and cooling channels will allow either working fluid. Use of liquid nitrogen as the only coolant is contemplated for the initial operation of the magnet. It is possible to use the inner vessel around the magnet as a vacuum vessel,

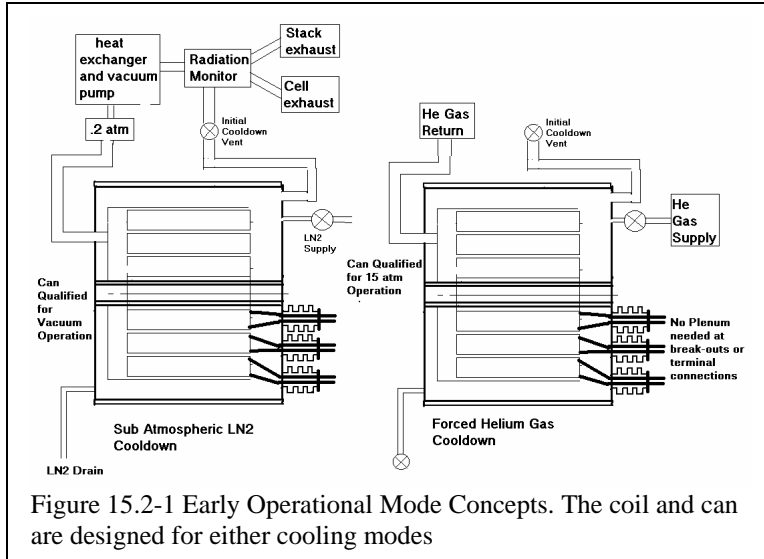


Figure 15.2-1 Early Operational Mode Concepts. The coil and can are designed for either cooling modes

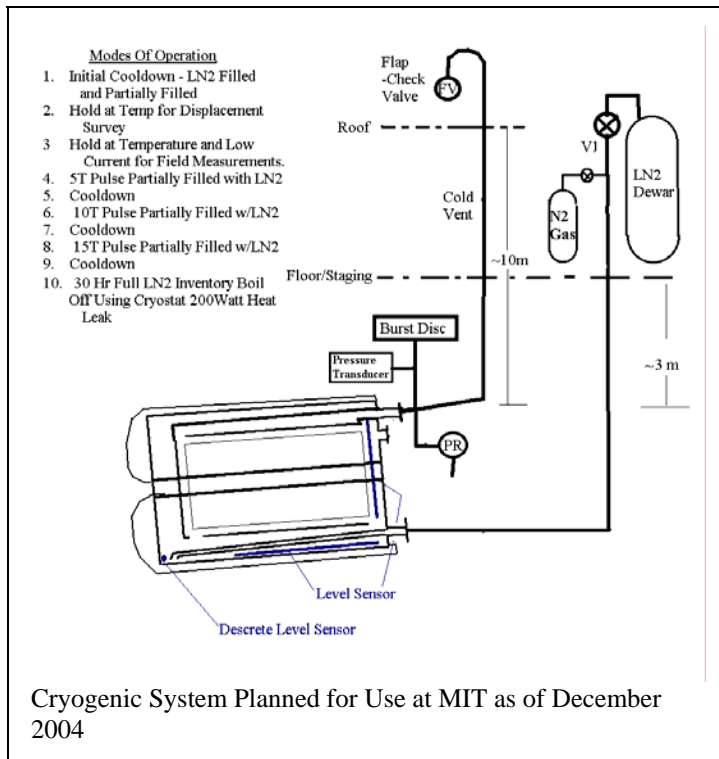
and sub-cool liquid Nitrogen to 66K. Helium gas operation uses liquid nitrogen to cool the helium gas in a heat exchanger, and later allows use of liquid hydrogen to cool the helium and obtain improved fields and/or pulse lengths. The heat transfer characteristics of liquid nitrogen operation have only been conceptualized. Heat transfer calculations using gaseous helium will be presented here.

For Liquid Nitrogen cooldown, between shots, the temperature range is about 110K to 80K for the full power pulse. The average temperature is 95 degrees. The specific heat

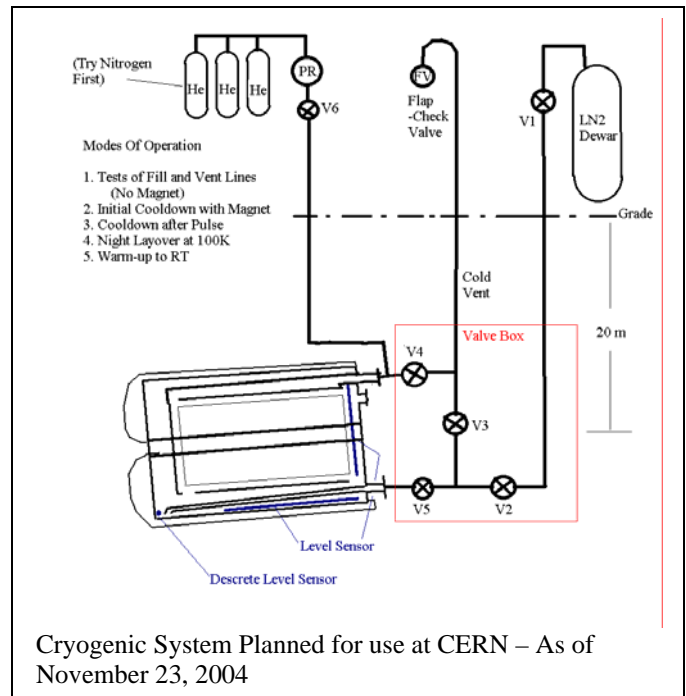
of copper at this temperature is 2090800. The heat that must be removed is:

$$(110-80) \cdot .4618 \cdot 2090800 (199000) = 145 \text{ Kg of LN2 based only on vaporization of LN2}$$

$$(110-80) \cdot .4618 \cdot 2090800 / (199000 + 2042 \cdot (95\text{K} - 80)) = 126 \text{ Kg of LN2 assuming an average exit exhaust gas temperature of 95K.}$$



Cryogenic System Planned for Use at MIT as of December 2004



Cryogenic System Planned for use at CERN – As of November 23, 2004

15.3 Finite Difference Transient Conduction Model Used in Cooldown Calculations

Initially the magnet was to be cooled via axial flow of cold Helium gas. At the end of the design process this was changed to LN2 pool cooling. The simulations available, however are for axial flow of coolant, and have been adjusted to address axial flow of LN2 and then circumferential flow of LN2 as in a pool boiling configuration.

The solenoid has three groups of 8 layers of 1/2 inch square conductors separated by set of annular cooling channels. The model employed, could model any linear stack of .5 inch square conductors cooled from the ends of the stack whether layer wound - then there would be a layer of channels every eighth layer of conductor, - or pancake wound, where there would be radial channels every sixth pancake. The solution is a simple finite difference transient analysis. Cracked conductor/Kapton tape interfaces have not been modeled. There is some effort to "encourage" cooldown crack formation that is parallel to the heat flux and delaminated insulation layers should not degrade radial conduction. This is discussed in the thermal stress section 6.0 of this report. The analysis starts with a specified mass flow which is apportioned to the 4 coolant channels based on the flow area of each channel. Coolant flow characteristics are modeled, including velocity Reynolds number, and surface heat transfer characteristics.

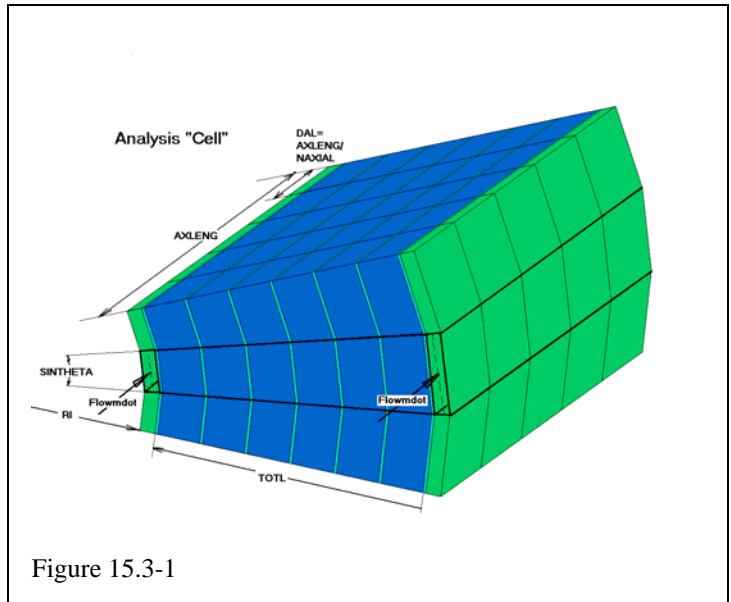
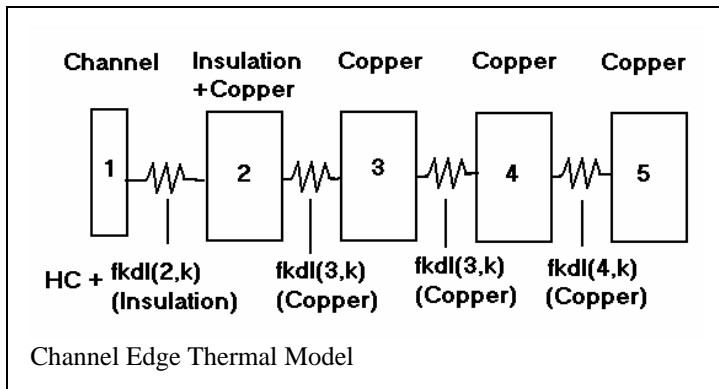
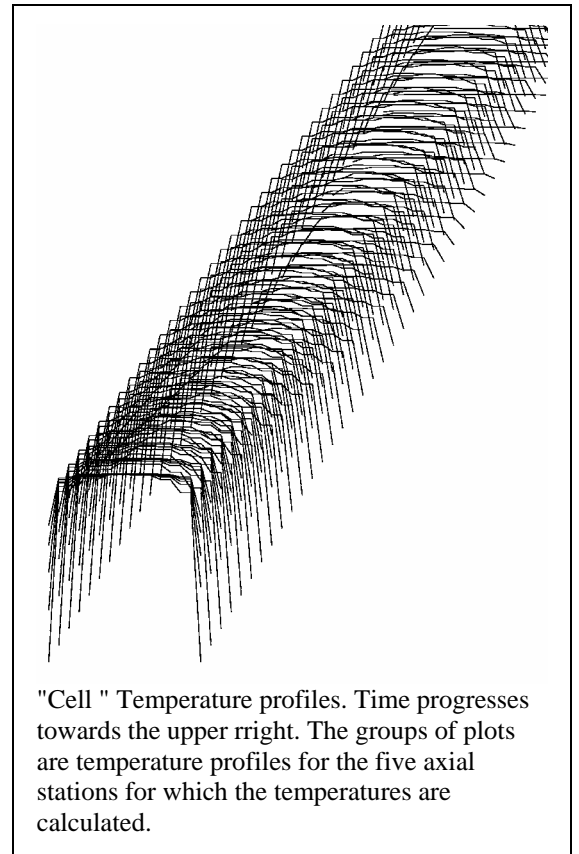


Figure 15.3-1



The insulation layer is modeled three, .001" thicknesses of Kapton tape. The thermal conductivity of the tape is about .14 W/(m-K) at 100 K and was taken from "Thermal Conductivity of Polyimide Film between 4.2 and 300K With and Without Alumina Particles as Filler" Rule, Smith, and Sparks, NISTIR #3948, August 1990.

The surface heat transfer coefficient at the channels was initially taken as 170 W/m² for nitrogen gas at 100K flowing at 40 m/s in a channel with a 2cm hydraulic diameter. This comes from an Oak Ridge CIT report # ORNL/FEDC-85-10 Dist Category UC20 c,d October 1986. The Helium gas coefficient has been



calculated from the Nusselt, Reynolds, and Prandtl Numbers using the relation quoted in the Oak Ridge document, which is a more generic heat transfer coefficient correlation, and is good for ideal gasses.

2.1.3 Convective Heat Transfer

It is important to estimate how much heat the superheated nitrogen gas ($T > 77$ K) could absorb before exiting the cooling channel. The convective heat transfer coefficient, h , could be obtained from⁹

$$h = \frac{K\text{Nu}}{D_e} = \frac{0.023\text{Re}^{0.8}\text{Pr}^{0.4}K}{D_e} \quad (14)$$

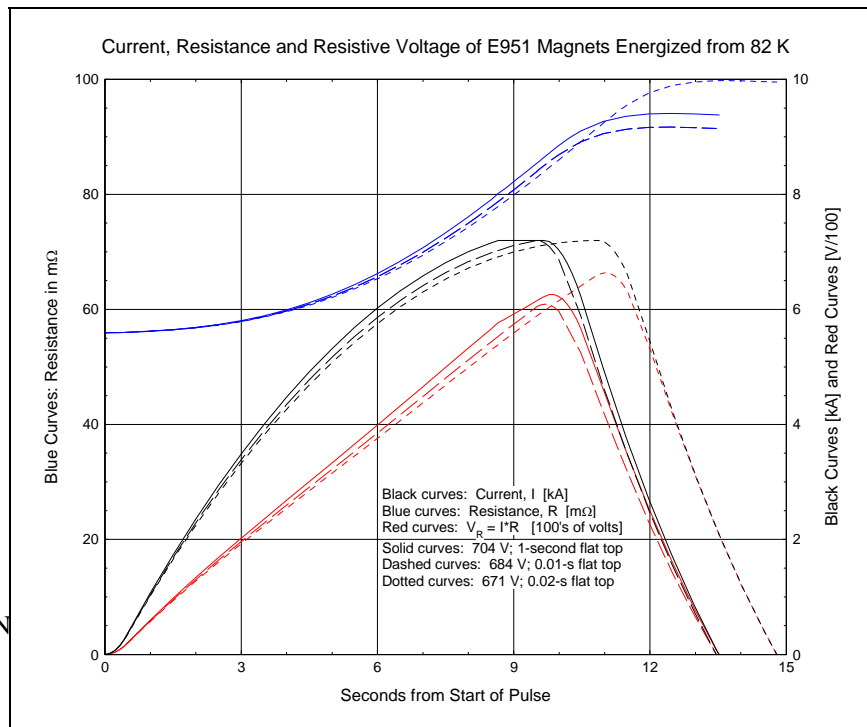
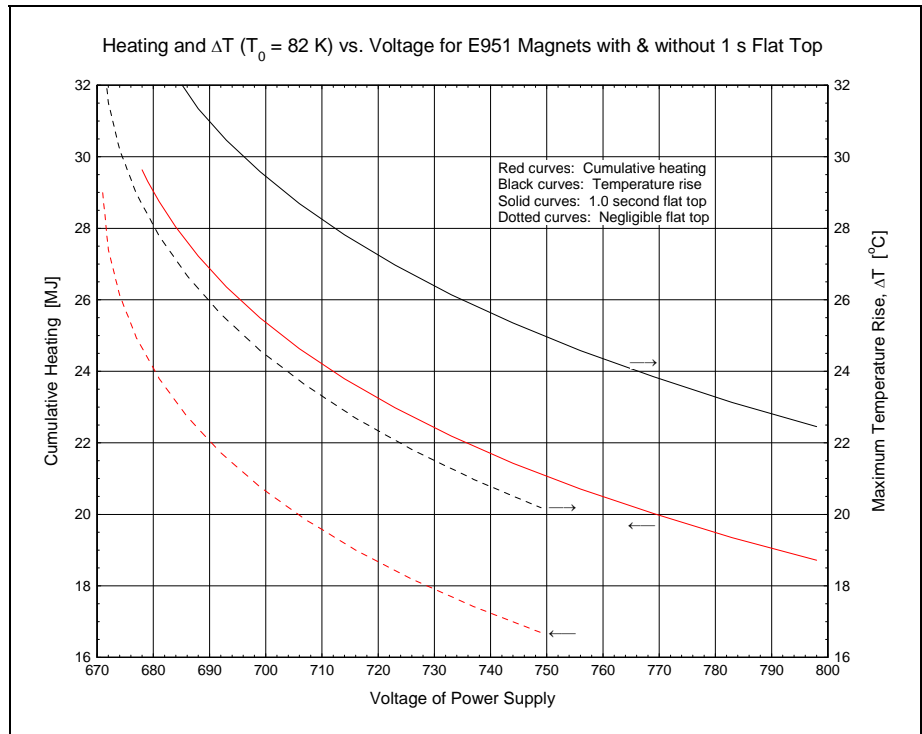
This coefficient is about 21×10^{-3} W/cm² K at a vapor temperature of 200 K, vapor velocity of 40 m/s, and hydraulic diameter of 2 cm. It drops to 17×10^{-3} W/cm² K at a vapor temperature of 100 K, keeping the mass flow rate constant. It is interesting to note that the heat transfer coefficient for film boiling at 200 K from Fig. 4 is about 12×10^{-3} W/cm² K, which partially justifies the third assumption in Sect. 2.1.

Excerpt from: ORNL/FEDC-85-10 Dist Category UC20 c,d October 1986

15.4 LN2 Two Phase and Metered Flow Options.

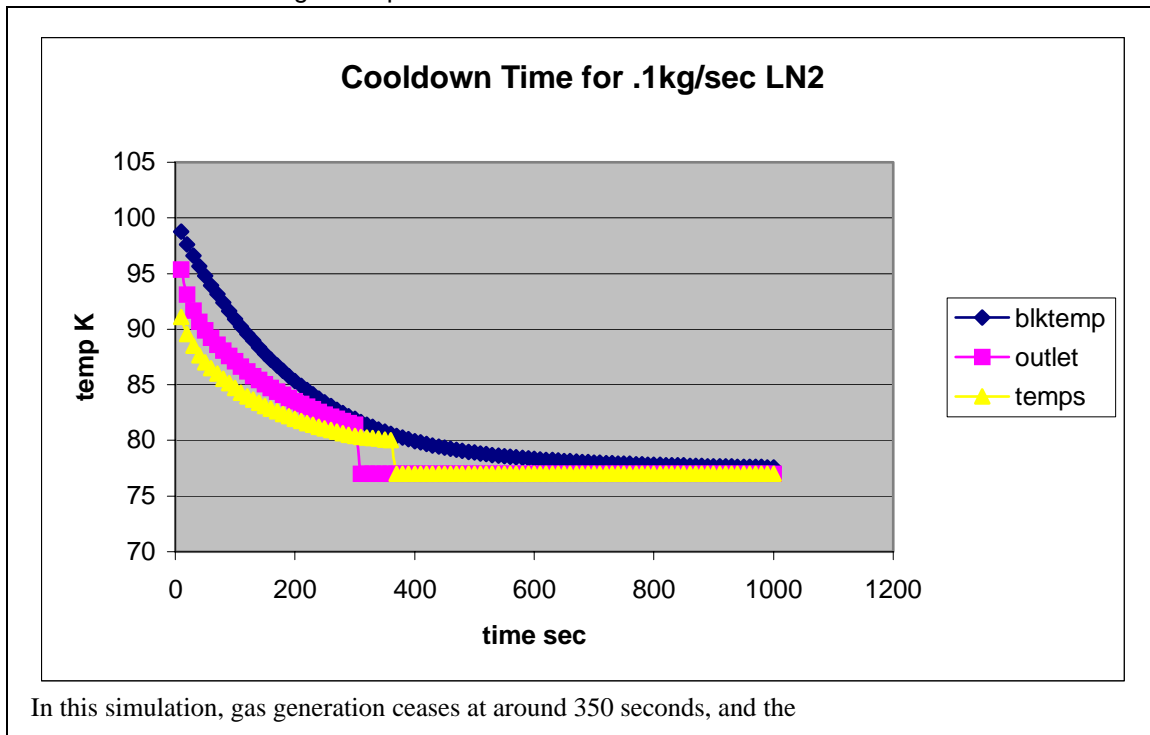
Bob did several calculations for us. His results show that it is better to run with as high a voltage as possible. For your cooldown calculations I recommend using as a baseline: 82deg K, 704 V, 1.0 sec flat top, 25 MJ energy deposition, del T 29 deg K.

A simplified LN2 cooling system is being considered for the pre-operational tests, and This may be an operational mode as well. The system is much simplified over the Helium gas system. The heat exchanger is not required, and LN2 is cheaper and easier to handle.



BN

npro	Project number	1.000
dtim	Intergration time step	0.001
tend	End time	1000.000
tinc	Print-Out Time increment	10.000
wflu	Coolant	3.000
comd	coolant mdot	0.100
nlen	Number of Nodes in conduction length	0.000
np11	Number of Nodes in a layer	10.000
naxi	Number of Nodes axial or channel direction	10.000
sint	Sin theta in Radians -Model slice angle	0.175
chan	channel height (halfheight in meters)	0.001
rein	reinforcement thickness (in meters)	0.000
numk	number of .001 in. Kapton wraps per layer	3.000
tini	inner channel start temp	100.000
tino	outer channel start temp	100.000
tcoo	Coolant inlet temperature	77.000
riii	inner channel radius	0.100
totl	conduction length between channels	0.098
axle	channel/coil axial length	1.000
nmat	Coil Material	1.000
numa	Operating Pressure (atmos)	1.000
curo	Current Distribution computation option	1.000
cooo	Cooling side option	2.000



Pressure Drop Through the Magnet

The maximum flow is .1 kg/sec.

The density of LN2 at 73.13K is $.8265 \text{ g/cm}^3 = 826.5 \text{ kg/m}^3$ ref [16] page 283 table 9.13

The density of LN2 at 74K is $1/(1.214e-3 \text{ m}^3/\text{kg}) = 823.7\text{kg/m}^3$ ref [17]

The Liquid Volume flow into the magnet is $(.1 \text{ kg/sec}) / (825 \text{ kg/m}^3) = 1.212 \times 10^{-4} \text{ m}^3/\text{sec}$.

The density of Nitrogen gas at 74K is $1 / (317.8 \times 10^{-3} \text{ m}^3/\text{kg}) = 3.1466 \text{ kg/m}^3$ (This is sub atmospheric, about .6 atmospheres)

The density of Nitrogen gas at 77K is $1 / (225 \times 10^{-3} \text{ m}^3/\text{kg}) = 4.44 \text{ kg/m}^3$ (This is one atmosphere)[17],table 3

The density of Nitrogen gas at RT is 1.2506 kg/m^3 (This is at one atmosphere)

If fully vaporized the gas flow at the exit of the magnet is $.1 \text{ kg/sec} * .3178 \text{ m}^3/\text{kg} = .03178 \text{ m}^3/\text{sec}$ for the sub cooled condition, and $.0225 \text{ m}^3/\text{sec}$ at atmospheric, and 77K

The coolant channel flow area is $(.1 + .2 + .3 + .4) * 2 * \pi * .002 = .0126 \text{ m}^2$ or about 39 square inches

The flow velocity of 74K gas at the magnet exit is $.03178 / .0126 = 2.52 \text{ m}^3/\text{sec}$.

The flow velocity of 77K gas at the magnet exit is $.0225 / .0126 = 1.7857 \text{ m}^3/\text{sec}$.

The velocity head is $3.1466 \text{ kg/m}^3 * (2.52 \text{ m/sec})^2 / 2 = 7.97 \text{ N/m}^2$ at the subcooled temperature of 74 K ,sub atmospheric

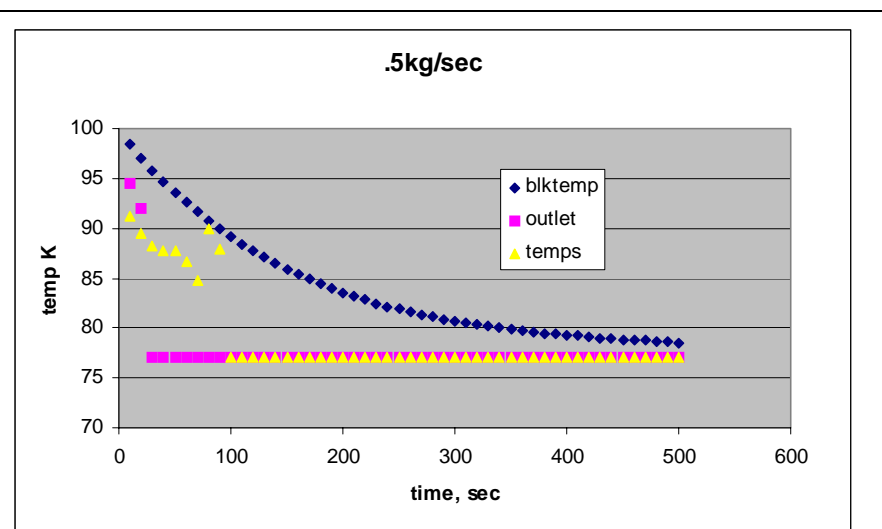
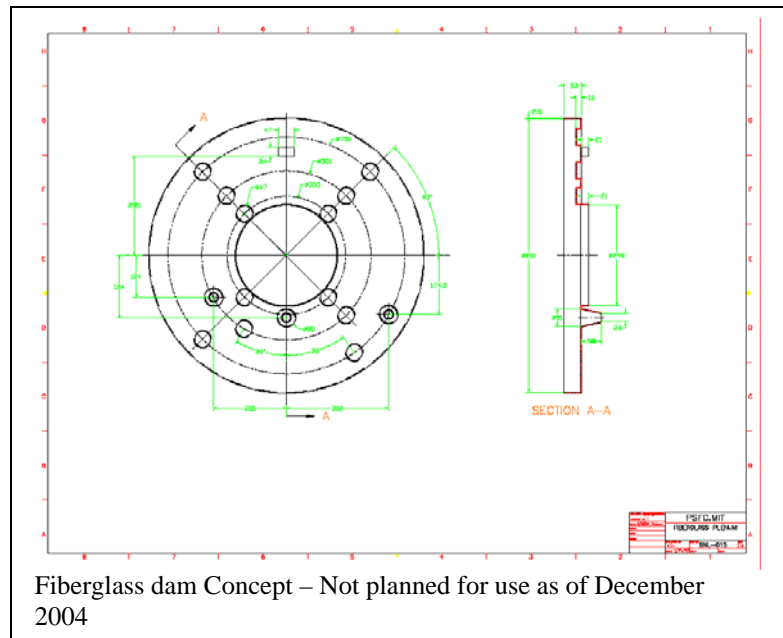
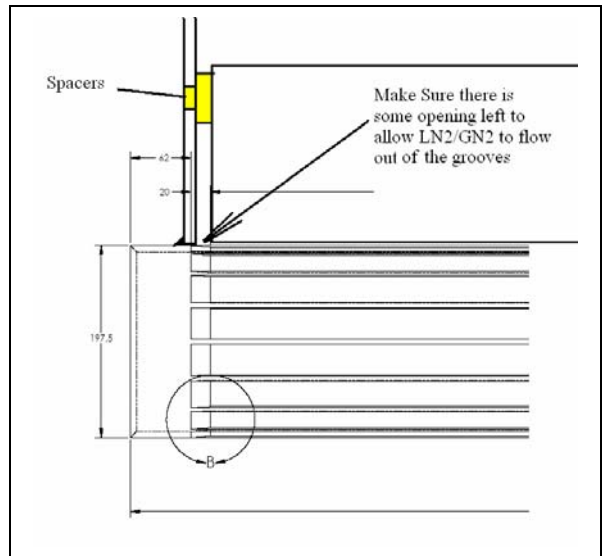
The wetted perimeter of the channels is $(.1 + .2 + .3 + .4) * 2 * \pi = 12.6 \text{ m}$ (ignoring the 2mm ribs)

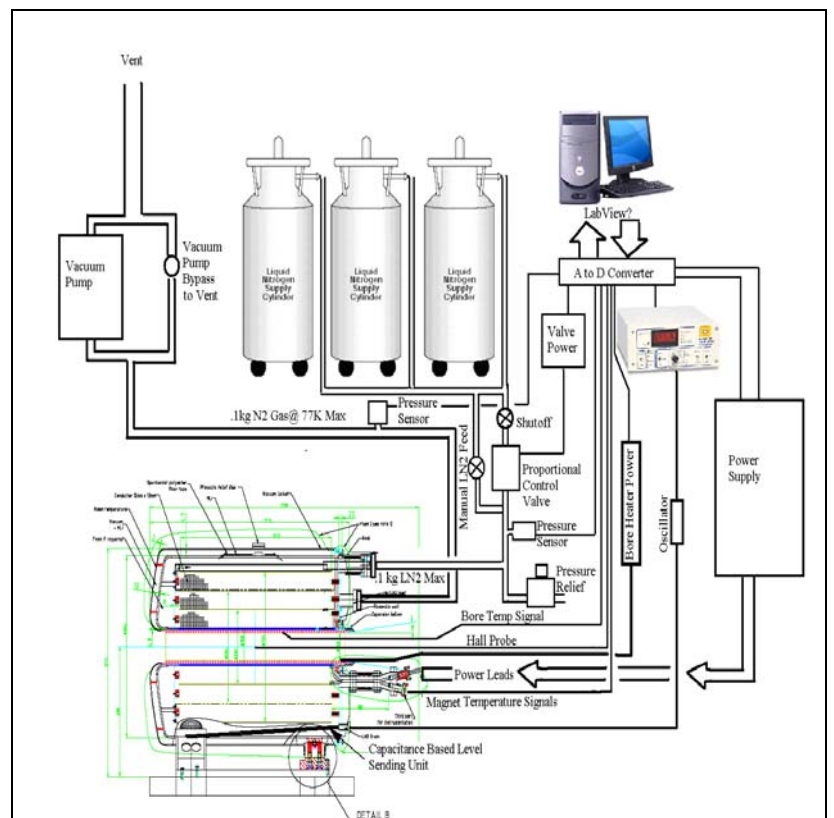
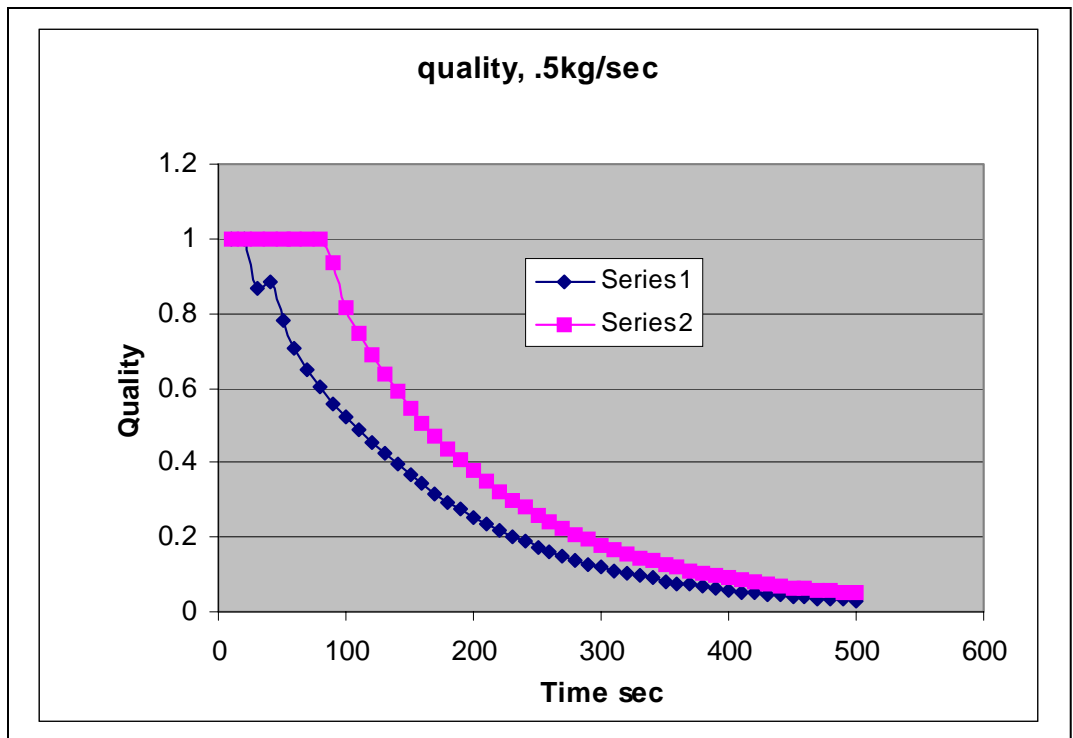
The Hydraulic diameter of the channels is $4 * .0126 / 12.6 = .004 \text{ m}$

The length of the magnet is 1m

The pressure drop in the channel $= 7.97 * 1 / .004 * .02 = 39.95 \text{ Pascal}$

This is too low to expect uniform flow from top to bottom of the magnet. If LN2 cooling is to be employed, and it is not desired to fill the full volume of the cryostat, then a way to reduce the volume of the trapped LN2. A dam at the joint/lead end is proposed to accomplish this.





Proposal for “complex”, drainless cryogenic system for pre-operational Tests at MIT. Tests will employ only LN2 cooling. Gravity fed flow is being considered, with a proportional control valve to meter the LN2 into the magnet at a flow which as nearly matches the amount of LN2 vaporized as possible. The cooling surface heat flux from the two phase flow simulation sets the flow needed to completely vaporize the LN2. It is assumed that quality calculated in the simulation can be used to compute the required flow.

The basic principle of operation with regard to handling excess LN2 is to use the magnet to vaporize the LN2 so that no external vaporizing is needed. The intended operation is to run through the metered cooldown with an effort to trail-off flow towards the end of cooldown, to limit the accumulation within the dammed volume and sump volume. The heaters are used to vaporize the LN2 that remains.

The dammed volume is

The coolant channel flow area is $.098 * (.1+.2+.3+.4) + 2 * \pi * .002 = .0123 \text{m}^3$

The left hand plenum volume is $(.4^2 - .1^2) * \pi * .025 = .01178 \text{m}^3$

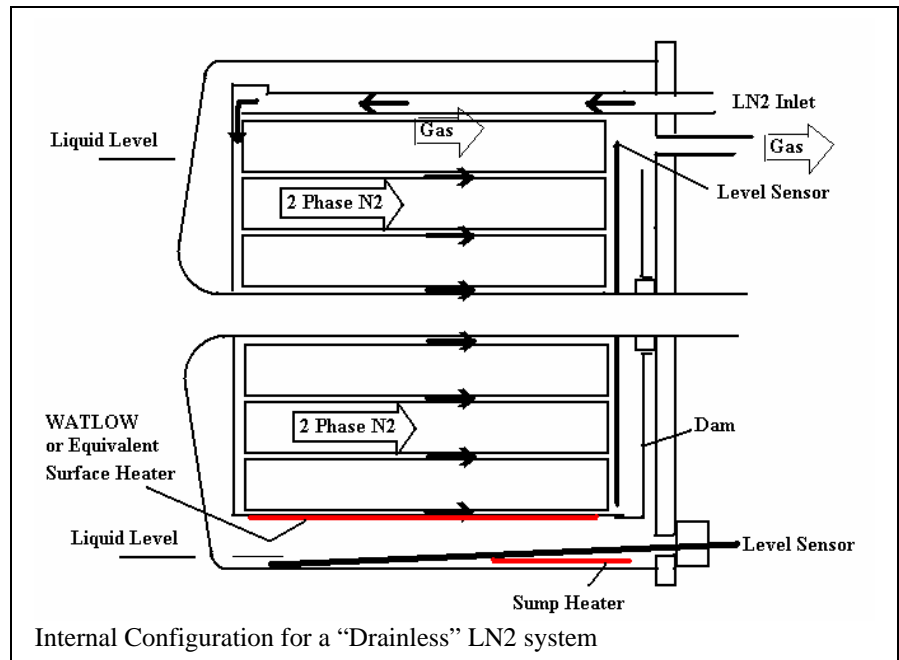
Assuming the plenum at the dam is comparable to the plenum at left, then the total “full” volume is $.0123 * 2 + .01178 = .03647 \text{m}^3$. If the liquid height is set at 80% volume then the dammed volume is $.03 \text{m}^3$. or about 30 liters. A typical Cooldown takes only 110 –120 kg LN2

One solution is to shut-off flow at 70 kg and rely on residual energy in the coil to remove the last 20 to 30 Kg of LN2, but this would take a great deal of time to conduct all of the heat to the bottom of the plenum. A better solution is to add a heater

Flow Scenario Using Heaters:

The target time-temp profile is one in which the bulk of the twenty-sum MJ is extracted in the first 300 seconds. This leaves 4 or 5 degrees K in the magnet to be removed or 3.6MJ – and this amounts to just about the inventory of LN2 in the dammed volume (about 23 liters now with the reduction in plenum gaps).

At this point the heater can be turned on. This gives you 900 seconds to remove the trapped LN2, and the heat in the magnet. If you are only worried about removing the LN2, the size of the heater would be 4kw. – But you also have the remaining heat in the magnet, but once the dam volume is full, and you have shut off the LN2 supply, the heat transfer on the upper part of the magnet is poor. However the boiling action of the LN2 being heated by the heater can be used to drive mixed flow and cold gas to the top. To allow this, I would propose that we machine circumferential grooves in the G-10 stand-off ribs. Everson will have to machine the outer surface of these ribs anyway to fit the ID of the next segment. To uniformly heat the underside of the magnet I propose to mount the heater at the bottom of the metal shroud surface in the gap between the shroud and inner vessel wall. We use thin film heaters on C-Mod in LN2/gaseous N2 environment to heat the vessel shell to room temp. These might be a better form than the calrod type heater. Two loops on the outside of the shroud would allow the 4 m of calrod type heater if needed. The heater will produce 2 phase boiling conditions in the trapped volume of LN2. This will reduce the effective trapped volume of LN2. – At the 300 sec time point we will back off on LN2 flow, start cycling the heater, watching the level indicator (mounted in the face of the dam, and away from 2 phase conditions that might alter it’s reading). We may find that turning the heater on at the beginning of the cooldown will help the



flow throughout the entire process. We will monitor the magnet temperatures. Currently, the Cernox temperature sensors are placed in the top of the magnet. I would propose shifting at least 2 to the bottom to monitor the temperature near the heater to ensure that the heater is not over-heating the shroud. Towards the tail end of the cooling cycle, while we are trying to extract the last couple of degrees from the magnet, there would be a nearly empty dammed volume. The flow in the inner segments is going to be complex, and may not be as good as for the outer segment where LN2 flow is driven by the two phase flow. We might end up with less than ideal temperatures, by about 2 or three degrees. This might be 3 or 4 degrees at the end of a pulse. this is OK from a stress standpoint. Using sub-cooling, and higher initial flows (shortening the 300 seconds) can help.

This still requires some testing to find the best timing of flows and heating.

Flow Equalization

The simulations assume a constant pressure manifold feeding the four annular channels. Flow resistance of each channel should be comparable to produce a uniform cooldown.

Specification Content:

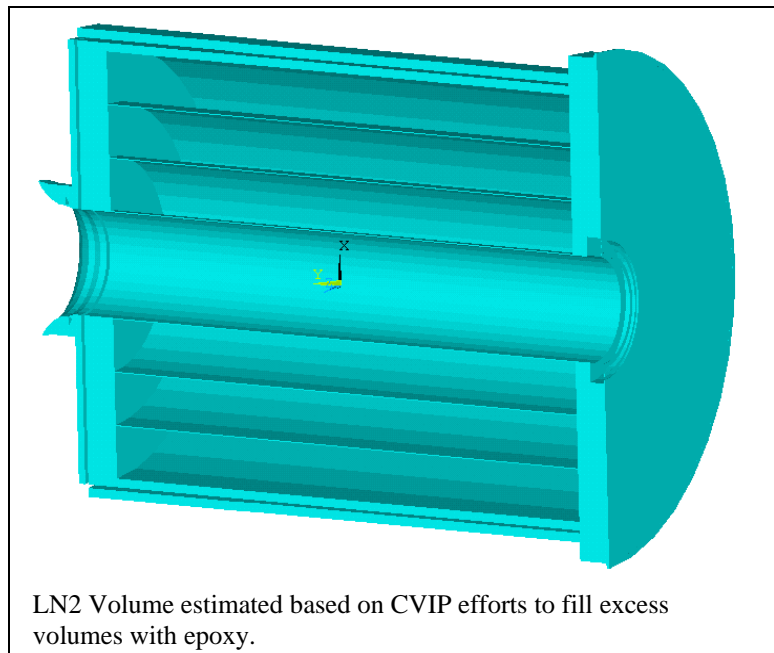
“The Seller shall perform a flow test of the assembled magnet by blowing air in the Helium outlet connection, and with the flat cover of the cryostat removed, flow velocity of each channel shall be measured with a Pitot tube. Restrictions on the channels will be applied until uniform flow is achieved. Restrictions can be in the form of g-10 strips bonded into portions of the channel opening. “

Options Using a Drain, LN2 Dump Flow and Pressure Drop Calculations

There is a sheet metal plenum that surrounds the magnet for the 1, 2 or 3 segment configurations - but only on the outer segment. The down stream plenum side communicates with the vacuum pump/exhaust. After LN2 cooldown, the exhaust should be shut to minimize flow into the plenum. After the LN2 is dumped, the valve to the shuttle tank should be closed, and the exhaust/vacuum line re-opened to purge the exhaust plenum. This can be timed to provide a last bit of cooling before the next shot.

If the coils are purchased with three segments installed (assuming we can afford the three)

the plenum will be installed on the outside of segment three. This means that during early operation with only two segments operating, there will be coolant flow through the third segment even though it remains cold -unless it is shipped with the outer channel plugged. In which case, the cover could be removed and the plugs removed prior to full performance operation. I think wasting one channel worth of cooling is preferable over opening the cryostat, but that can be Brookhaven's choice.



The coolant channel flow area is $(.1+.2+.3+.4)*2*\pi*.002 = .0126\text{m}^2$ or about 39 square inches
 Assuming a 1.5 inch line to the shuttle tank, the flow area is: $\sim .00115\text{m}^2$ or 1.77 sq. in.
 The wetted perimeter of the channels is $(.1+.2+.3+.4)*2*2*\pi = 12.6\text{ m}$ (ignoring the 2mm ribs)
 The Hydraulic diameter of the channels is $4*.0126/12.6 = .004$
 the assumed pipe wetted perimeter is .119m, and the pipe hydraulic diameter is .0381m

The length of the magnet is 1m . assume the line to the shuttle tank is 2 m

The ratio of the pressure drop of the pipe to the channels is:

$$(.0126/.0015)^2*(.004/.0381)*(2/1) = 25, \text{ assuming similar friction factors.}$$

So the pressure across the plenum would be much less than the pressure differential between cryostat and shuttle tank if LN2 was allowed to flow freely out the exhaust. If the exhaust line valve is closed prior to the dump to the shuttle tank, there will be almost no flow through the channels, and no pressure drop.

Estimating the pressure needed to vent $.3\text{m}^3$ in one minute, that is $.005\text{ m}^3/\text{sec}$. The flow velocity in the 1.5 in pipe is $.005/.00115 = 4.35\text{ m/sec}$. with a conservative friction factor of .1, the pressure drop is $857\text{kg/m}^3 * 4.35^2/2 * .1 * 2/.0381 = 42563\text{ Pa} = 6.2\text{ psi}$.

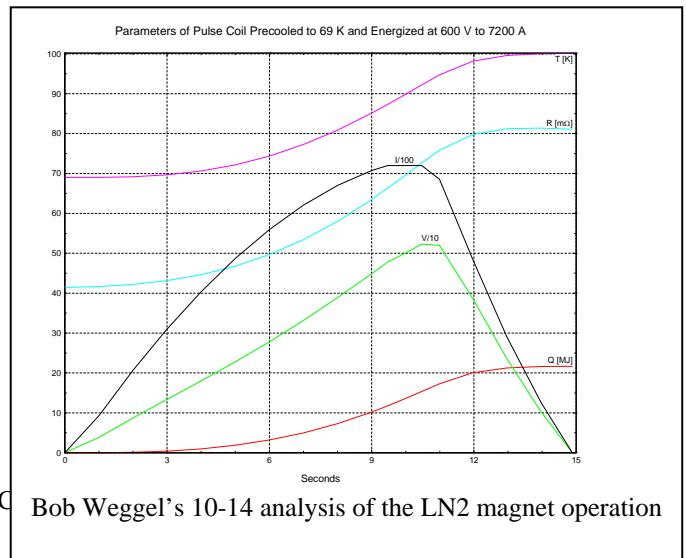
The shroud will rest on stand-offs the same as for the coil segments. I don't think we can maintain concentricity without the stand-offs. These are the strips bonded on the outside of the wound coil and machined to allow a slip fit of the shroud. We have 16 strips at the ID of segment 1, 32 between segments 1 and 2, 64 strips between segments 2 and 3 and a comparable (yet undetermined) number on the outside of segment 3. With 64 strips, the channel opening would be about 4cm wide. Assuming the shroud sheetmetal is a beam supported by the strips, and it actually sees the 2.0 atmospheres, the bending stress is $w*I^2/12/(t^2/6) = 14.7*2*1.5^2/12/(.0625^2/6) = 8467\text{ psi}$. At LN2 temperature the yield of the 316 will be over 100 ksi. This is assuming 1/16 inch sheet stock for the shroud. I think Val specifies thicker material. The stress is lower if the shroud supports the pressure in hoop compression.

The external pressure on the annular flat plate that forms the plenum on the downstream side of the magnet is thick, and is supported by blocks against the coil. If the downstream plenum volume becomes pressurized with respect to the cryostat pressure, this flat plate might bend, and the magnet might slide on the cryostat bore tube, but I can't think of a circumstance where this pressure could be higher than the cryostat pressure.

The present configuration is structurally adequate to take the likely pressures that might build up during the dump to the shuttle tank. I think we will need to add some stiffeners to the plenum annular plate because it needs to support the coil load during vertical assembly. Can you update me on your estimate of the pressure to achieve the 300 l/m?

Estimate of cooldown time for LN2 Cooling Mode

For the 100K to 69K cooling in the LN2 mode of operation, a 17 minute cooldown time between shots is possible with some forced circulation through the channels. Without circulation, the cooldown time could be longer than 40 minutes. It all depends on the behavior of the stagnant fluid/gas in the channels. Without clearing the bubbles with some forced circulation, I would envision a "blurp" mode in which a small amount of LN2 enters the channels, flashes to gas and is expelled at the ends forcing fluid out as well. The channel geometry could be altered, but it will be difficult to obtain something like a pool boiling mode inside the channels which would require vertical free surfaces. An

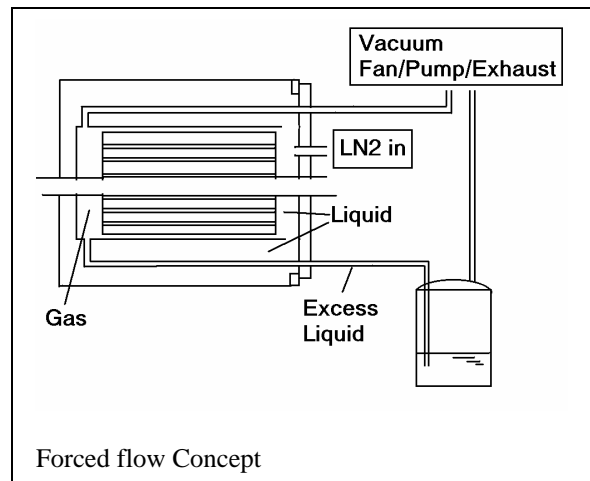


analysis of the coil cooldown with only conduction through the build of the coil yielded a 40 minute cooldown time, but this assumed conduction across the interior channels. If the interior channels act as thermal resistance because they are gas bound, the cooldown could be longer. Bubble clearing is an important issue in superconducting magnets. In MECO we have many flow channels cut into the mandrels in the production solenoid. These are vertically oriented. Channels in the BNL magnet are presently horizontal. In LN2 cooled Tokamaks, cooling is gravity fed with vertical channels or they have forced flow.

One option to get some flow is to pump gas on the exhaust side and cool with a mode where LN2 at the lower part of the magnet boils, and the upper part of the magnet is cooled by gas flow. An analysis of the Nitrogen gas forced flow yielded very long cooldown times. The analysis was similar to the helium cooling analysis with .1kg/sec coolant flow, but the specific heat of N2 gas is a fifth that of Helium gas. This approach doesn't look viable. The possible "fixes" are:

- Spiral channel ribs might offer some (difficult to quantify) bubble clearing behavior. This produces a bit of a complication in maintaining coil torsional registration – but may be the simplest approach. With this approach the plenum and shroud details could be removed.
- Forced flow with a circulator. The existing Helium pump could be used for this? This would potentially yield lower cooldown times than the 17 min quoted above, because true liquid forced flow would be substituted for natural convection.
- Forced flow via pressure differentials is also a possibility. A forced flow concept is presented here. The existing drain line would be connected to the downstream side of the plenum to be used to drain LN2 that hasn't been vaporized in the magnet.

Liquid fills the inlet plenum and passes through the channels forced by a pressure differential between the inlet and outlet sides of the magnet. The pressure differential is maintained by exhaust fans or vacuum pumps. The flow through the channels is needed to clear bubbles. Some LN2 is not converted to gas, and collects on the downstream side of the magnet. This is drained to the dump tank which is used as a phase separator. Ultimately a .2atm pressure is maintained by the vacuum pumps to reach the subcooled temperature of the LN2. Channel flow exiting to the outlet plenum enters as two phase flow. If the dump tank gets filled, the cooling process would be stopped, and the fluid in the dump tank would be transferred to the LN2 supply tank by closing valves and pressurizing the dump tank. This process would not carry much more time penalty than the "flush" of Nitrogen that is currently planned. The inventory of LN2 below the drain in the cryostat could be minimized with fillers. A simulation has been done with the transient conduction code previously used for the Helium gas cooling mode. The code was modified to include forced N2 gas cooling, and pool boiling heat transfer coefficients.



Magnet Flow Area Characteristics:

The coolant channel flow area is $(.1+.2+.3+.4) \times 2 \times \pi \times .002 = .0126 \text{m}^2$ or about 39 square inches

The wetted perimeter of the channels is $(.1+.2+.3+.4) \times 2 \times 2 \times \pi = 12.6 \text{ m}$ (ignoring the 2mm ribs)

The channel surface area is 12.6 multiplied by the one meter length of the coil or 12.6m^2

The magnet surface area is $(.1+.2 \times 2+.3 \times 2+.4) \times 2 \times \pi = 9.42 \text{m}^2$

The Hydraulic diameter of the channels is $4 \times .0126 / 12.6 = .004$

To cool the magnet down from 100 to 69 K, 22 MJ is required. This will be done using 66K subcooled LN2. Approximately, this will vaporize $22e6/199000 = 110$ kg of liquid. At 66K, and 1 atm, the specific volume of the gas is $.937 \text{ M}^3/\text{kg}$. 103.6 m^3 of 66K gas would be produced. If we want to cool down in roughly 20 min, then the flow velocity at the exit of the magnet would be: $103/.0126/(20*60) = 6.85 \text{ m/sec}$ with a mass flow of $110/(20*60) = .0916 \text{ kg/sec}$. The volume exiting the exhaust is $103/(20*60) = .086 \text{ m}^3/\text{sec}$ at 66K, and $.086*292/66 = .38 \text{ m}^3/\text{sec}$ after heating to room temperature.

The magnet surface area, exclusive of the ends, is 9.42 m^2 . The ends are excluded because they will be thermally insulated with the transition filler pieces.

For a 20 minute cooldown, the heat flux will have to be:
 $22e6/9.42 \text{ m}^2/1200 = .194 \text{ W/cm}^2$

Using the pool boiling correlations for Nitrogen (the same as G. Mulholland.) As George points out, The delta Temperature needed to obtain the required heat flux is quite small. Thermal conduction through the winding and insulation needs to be simulated. With LN2 wetting the surface of the channel, the channel wall may be at the LN2 temperature, making even the small deltaT needed for heat removal unachievable. The conduction/gas flow simulation was modified to incorporate the pool boiling correlations. This is only an approximation inside the channels. Some form of fluid motion will be required to match the convective heat transfer behavior.

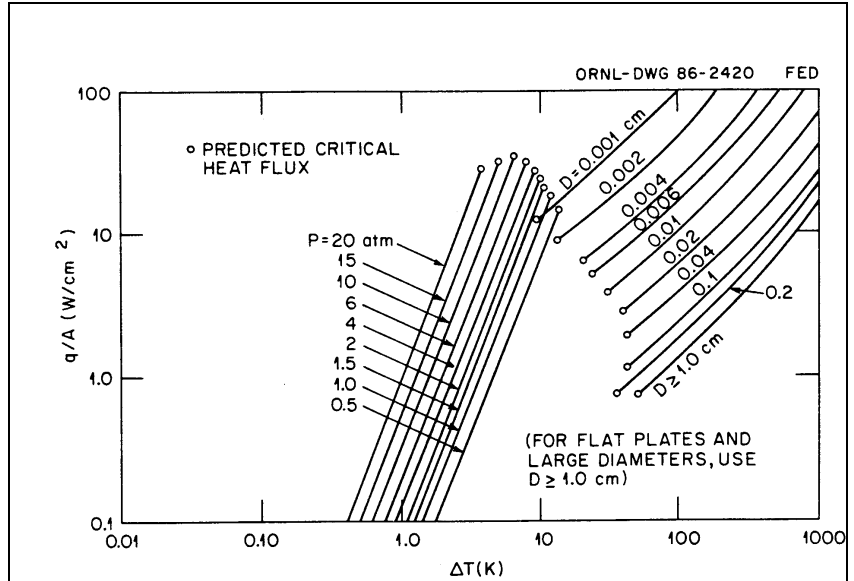


Fig. 4. Nucleate and film pool boiling correlations for nitrogen.
 from: ORNL/FEDC-85-10 Dist Category UC20 c,d October 1986

2.1.1 Pool Boiling Correlations

The following equations were obtained as a best fit for the correlation shown in Fig. 4, for the conditions of interest (coolant pressure = 1 atm, tube diameter > 0.2 cm). For the nucleate boiling regime,

$$\frac{q}{A} = 0.034 \Delta T^{2.58} \text{ (W/cm}^2\text{)} \quad 1 < \Delta T < 13 \text{ K} \quad (1)$$

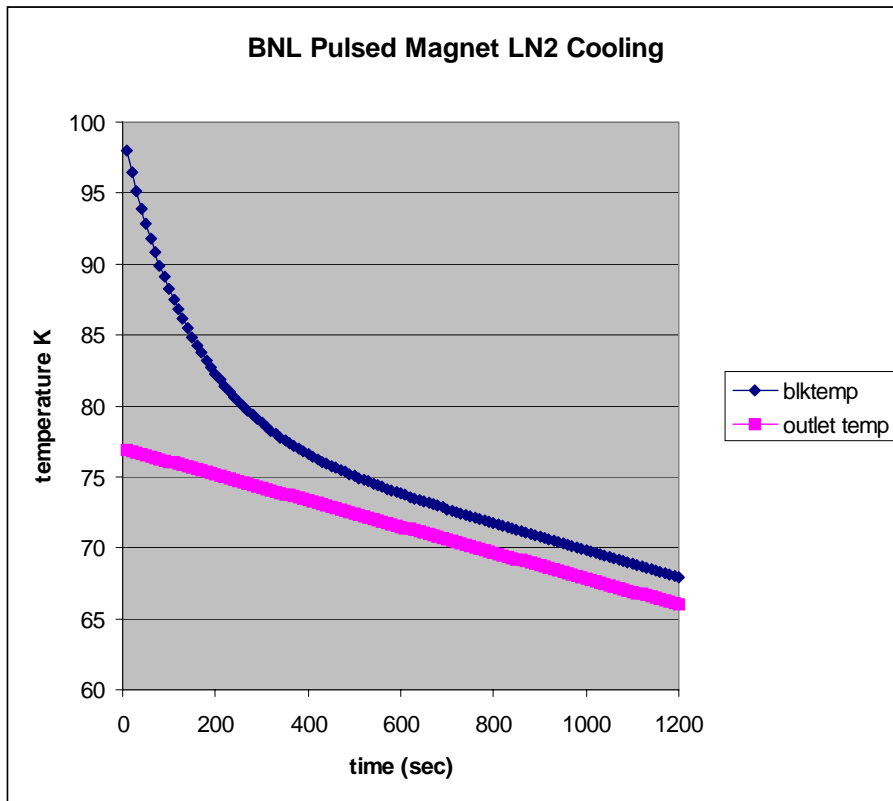
and for the film boiling regime,

$$\frac{q}{A} = 0.0143 \Delta T \text{ (W/cm}^2\text{)} \quad \Delta T > 53 \text{ K} \quad (2)$$

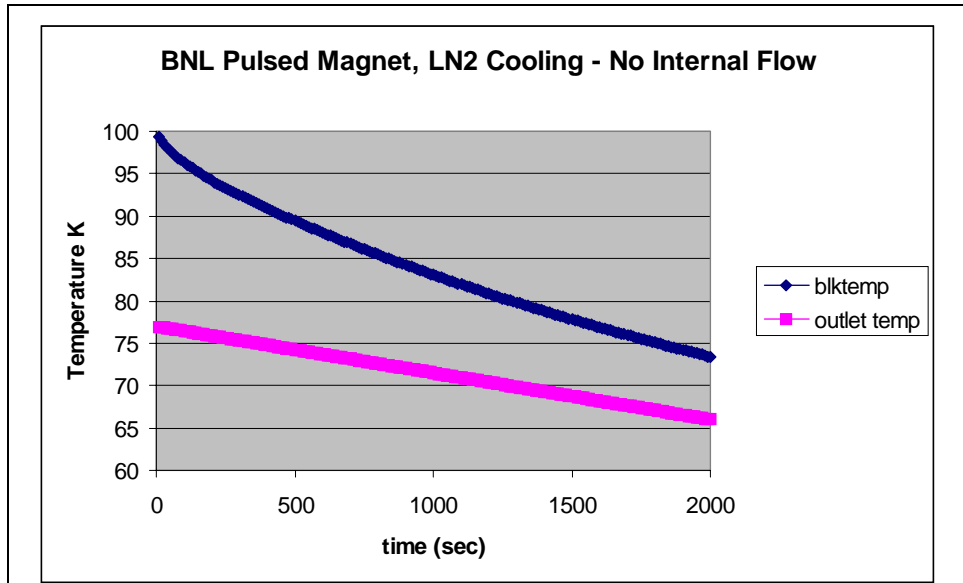
from: ORNL/FEDC-85-10 Dist Category UC20 c,d October 1986. Note that the correlations are not a function of the diameter in the nucleate boiling regime. Equation (1) above is used in the computer simulation. Channel hydraulic diameter was used to estimate the heattransfer coefficient in the film boiling regime.

Code Modification for Coolant Temperature Gradual Reduction:

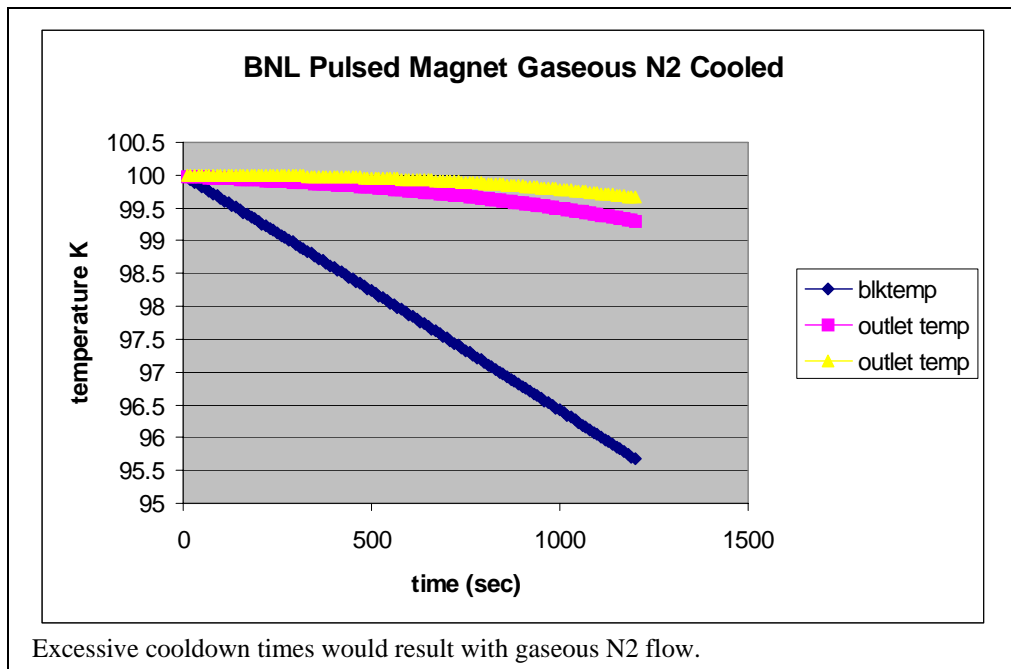
```
if (iwflu.eq.3) then  
t(1,k)=77-11.0*etime/tend  
t(nleng+2,k)=77-11.0*etime/tend
```



Cool down behavior, Fully immersed, but circulating LN2. LN2 pressure gradually reduced from 1atm to .2 atm. Time to 69 degree K bulk temp is about 1000 sec.



LN2 Pool Boiling Cooldown behavior Conduction through the full coil build – Interior channels are ignored. LN2 pressure gradually is reduced from 1atm to .2 atm. If after 2000 sec the LN2 temperature can be maintained at 66K a guestimate for the final coil cooldown to 69K would be about 2500 sec or 42 minutes. – But this assumes conduction across the interior channels



Excessive cooldown times would result with gaseous N2 flow.

15.5 Cooling Time with Helium gas as a Working Fluid

Estimate of Cooldown time vs. deltaT for Helium Cooling Mode

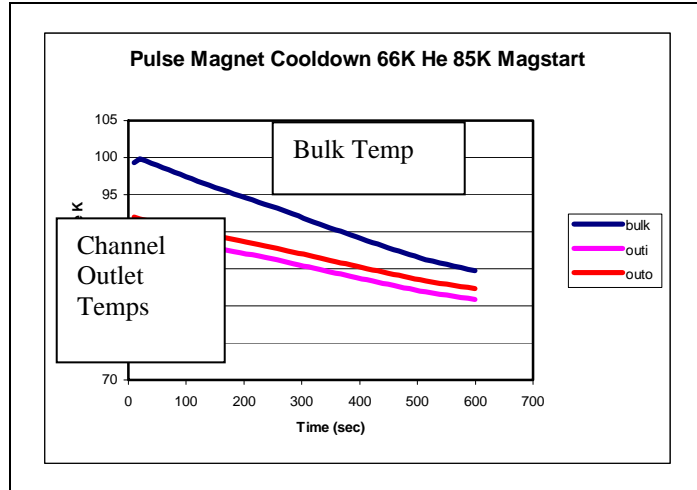
A simple check of the behavior of the magnet is available from the fluid mass flow. The power removed in a flow of fluid is $\dot{M} \cdot \text{specific heat} \cdot \Delta T$. \dot{M} is .1 kg per second, the specific heat of Helium gas is 5190 Joules per kg. The magnet energy change from a post pulse temperature of 78.3 to 30K is 15.2 MJ. The estimate of the energy for the 100K example is $15.2 \cdot 70 / 48.3$ or 22 MJ. The time to remove the heat from the magnet is:

$$\text{Cooldown time} = 22e6 / (.1 \cdot 5190 \cdot \Delta T)$$

Or, the temperature change from inlet to outlet is:

$$\Delta T = 22.05e6 / (.1 \cdot 5190 \cdot \text{cooldown time})$$

The ΔT required for a 20 minute, 1200 sec cooldown time is 35 degrees. This is the difference between the inlet and outlet temperatures, so a 24K inlet yields a 59 degree outlet. This is for a steady state flow. To meet the 1200 seconds, much of the time the ΔT must be higher than that.



Operational Scenarios:

Case #	Peak Field	T after pulse	T coolant	Start Bulk Temp for Next Shot	Guestimated Time
I	5T	90K	66K	84K	~200 sec
II	10T	96K	66K	74K	~800 sec
III	15T	78K	22K	30K	~1500 sec

Requested Simulations, Start Temperature, 100K, 100 g/sec He Gas flow rate at 20 atm.

Case #	Peak Field	T after pulse	T coolant	Start Bulk Temp for Next Shot	Cooldown Time (sec)	Initial delta T through the Magnet	Pressure drop
I	5T	90K	79K	84K			
II	10T	96K	68K	74K			
III	15T	78K	24K	30K			

Typical results for 66K He cooling, .1kg/sec, 100 K end of pulse temp. 85K target Magnet start temp. The cooldown time is 600 sec. to reach 85K bulk temp, but not thermal equilibrium.

Number of Atmospheres Operating Pressure ;10

Enter Channel Height in mm ;2

Rinner, radial build 0.1000000 7.6200157E-02

inner coil start temp 100.0000

outer coil start temp 100.0000
 inner coil radius 0.1000000
 model cell energy 1644.685J (100 to 85K bulk)
 model cell volume 5.5099601E-05
 volume cpp 1989954.
 nlength, naxial, 120 5
 Mass flow rate= 4.1666666E-05kg/sec
 Volume flow rate= 5.5507730E-06
 flow velocity= 2.120239
 Hydraulic Diameter= 2.8944151E-03m
 Velocity Head= 1.721665 Pascal
 Pressure Drop= 31.46283 Pascal
 Pressure Drop= 3.1041747E-04Atmospheres
 Helium density= 7.506462 kg/m³
 Helium viscosity= 2.6448268E-07
 Prandl #, Reynolds # 4.0756337E-02 174174.1
 Heat transfer coefficient 115984.9

From $\dot{m} \cdot c_p \cdot \Delta T$ for a 20 deg inlet-outlet difference the cooldown time is about 950 sec. The simulation with a finer time step ($\Delta t = .0001$ rather than $.001$) yields a 600 sec cooldown. The inlet outlet ΔT ranges from 26K to 16K. The Energy balance or difference between the conduction heat flux and the channel heat flux. Is good at the finer time step

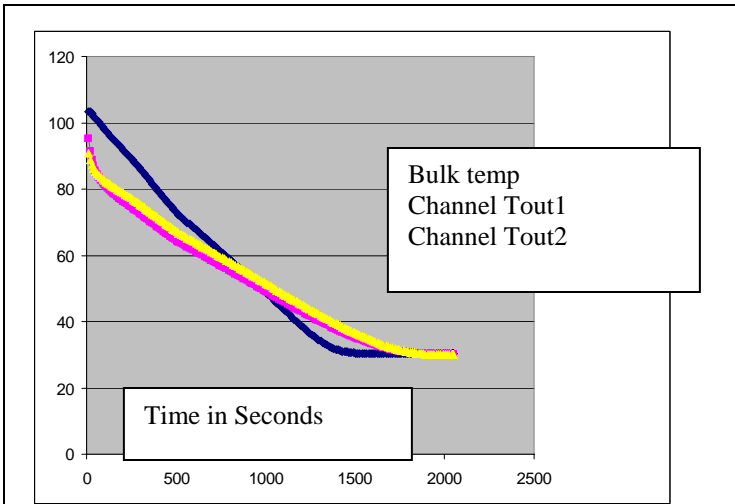
Helium Coolant, Simulation Results

Initial Analyses : Time to target bulk temp from 100K. ½ inch Copper Conductor.

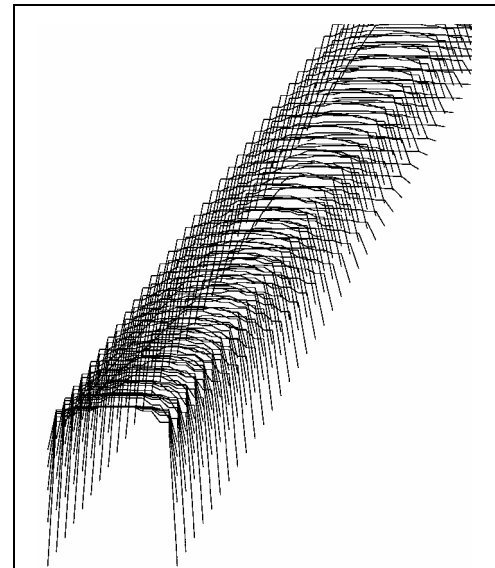
	T after pulse	T coolant	Cond Layers	Time to 85K sec	Time to 30K sec
Equip 5 Kapton .001in wrap	100K	66K	6 layers	600	
Equip 5 Kapton .001in wrap	100K	66K	8 layers	>850	
Equip 3 Kapton .001in wrap	100K	66K	8 layers	450	
Equip 5 kapton .0001in wrap	100K	30K	6 Layers		2000

M dot	Time Step (sec)	Initial Temp	Final Temp	Tcool	Time to cool down (sec.)	Tend-Tstart	Excel File
.1		30	100	30	2000	70	(early#1)
.1	.001	74	100	67	2100	26	hegas1.xls
.1	.0001	53	78	20	570	25	Hegas7.xls
.1	.001	53	78	20	790	25	
.1	.001	84	90	77	1000	6	hegas3.xls
.1	.001	74	100	67	2100	26	hegas1.xls
.1	.0001	74	100	67	2100	26	Hegas8.xls
.1	.0001	80	100	67	1470	20	Hegas8.xls
.1	.0001	30	78	20	1100	48	Hegas7.xls
.1666	.001	84	90	77	650		Hegas9.xls
.1666	.001	74	96	70	1600		Hegas10.xls

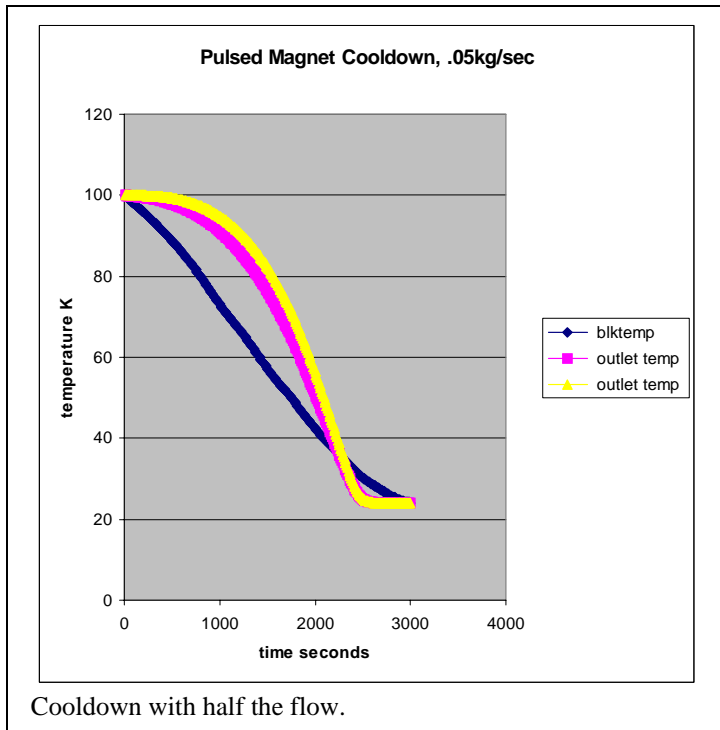
.1666	.001	74	96	70	1670	22	hegas5.xls
.1666	.001	74	100	67	1600	26	hegas2.xls
.1666	.001	84	90	77	660	6	hegas4.xls



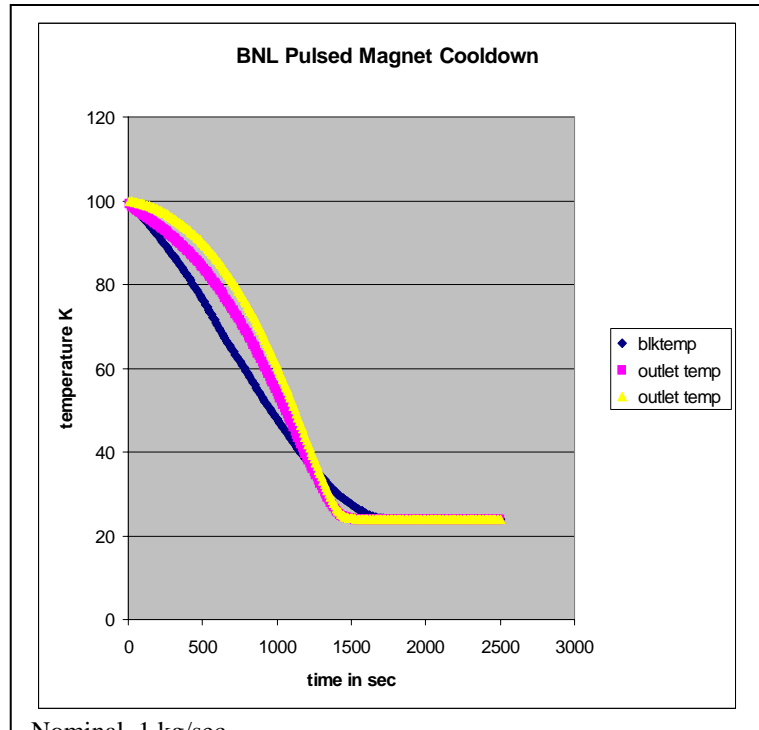
Early run, when bulk temp was approx at a slice mid axial build. In this simulation, Initial outlet temperatures start at only a fraction of a degree below the bulk temp. The magnet temperature at the outlet end is higher – near the channel outlet temps. With the bulk temp estimated mid axial build, the bulk temperature could drop below the channel outlet temp. the channel outlet temperatures are better indicators of when the magnet reaches 30K. tout 1 and tout2 are Outlet Temperatures



"Cell " Temperature profiles. Time progresses towards the upper right. The groups of plots are temperature profiles for the five axial stations for which the temperatures are calculated.



Cooldown with half the flow.



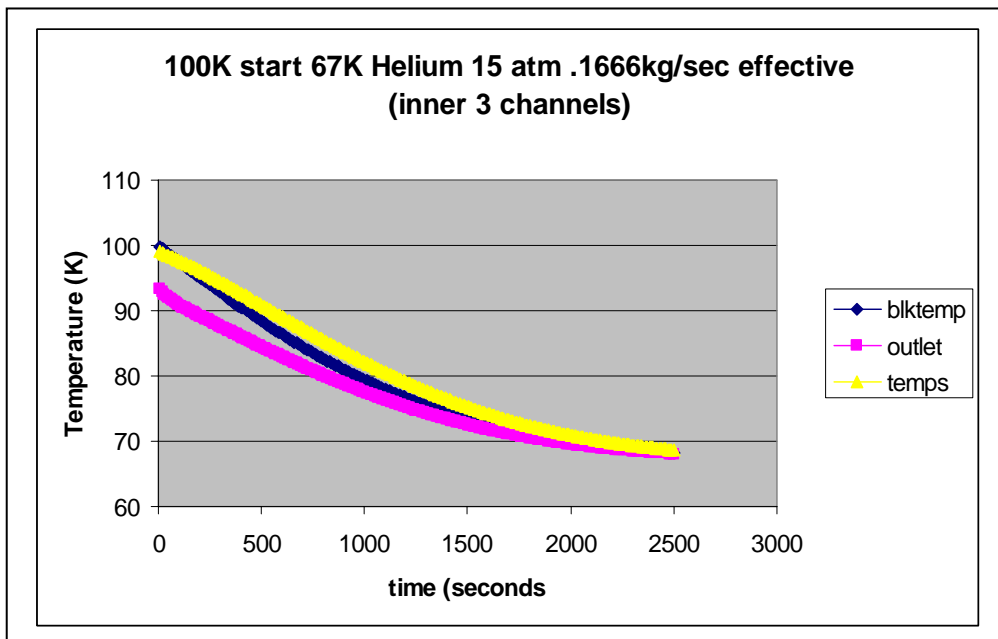
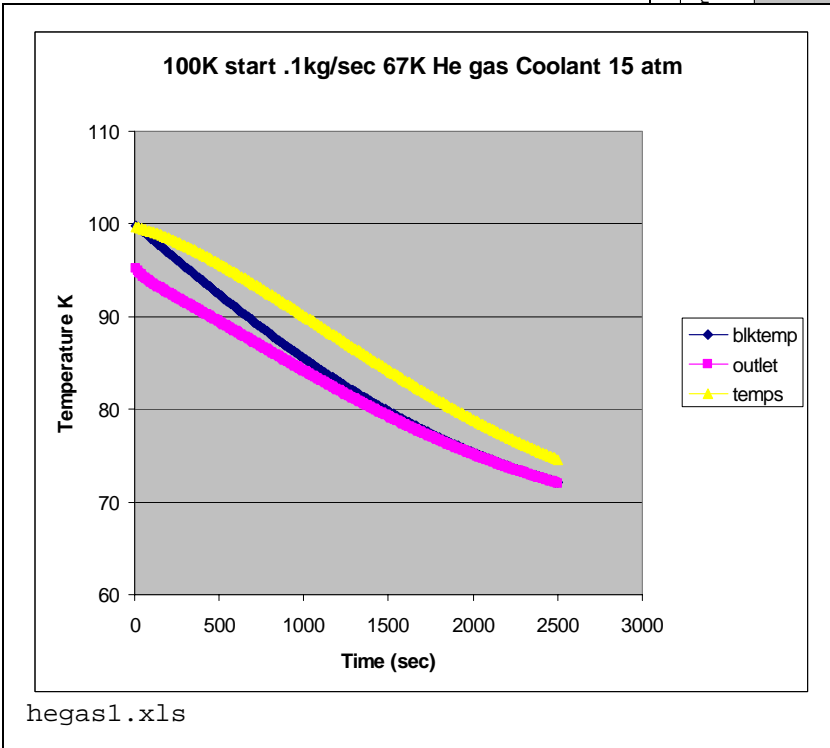
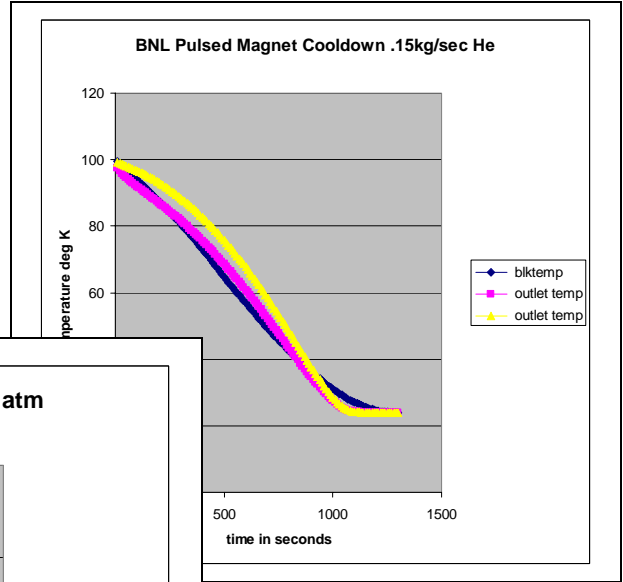
Nominal .1 kg/sec

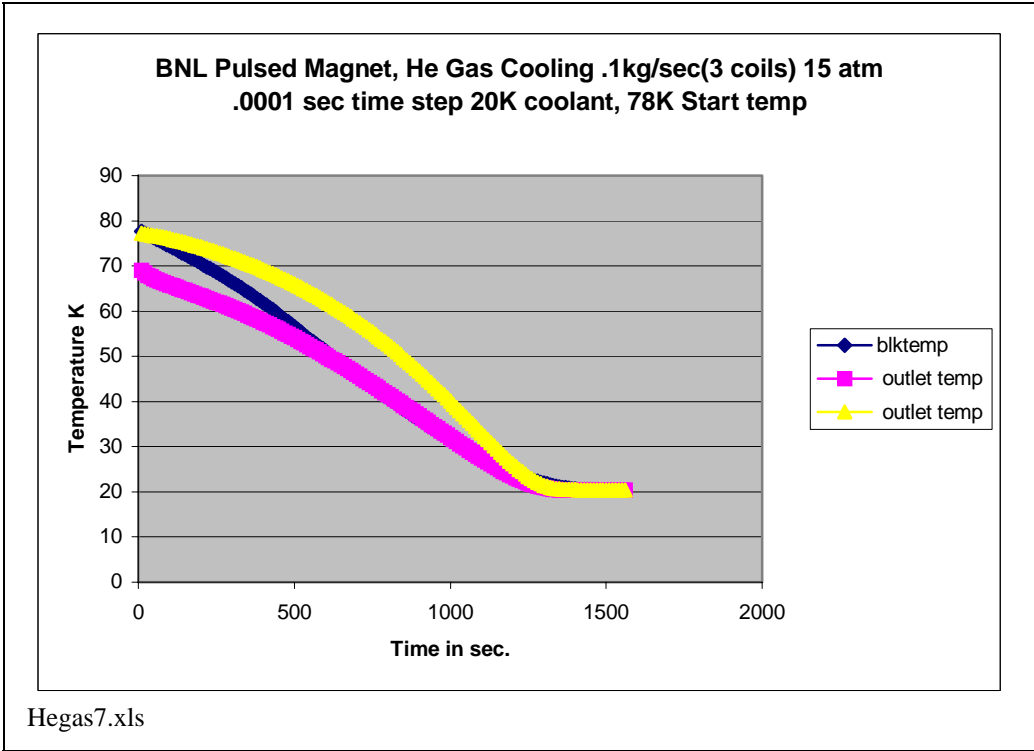
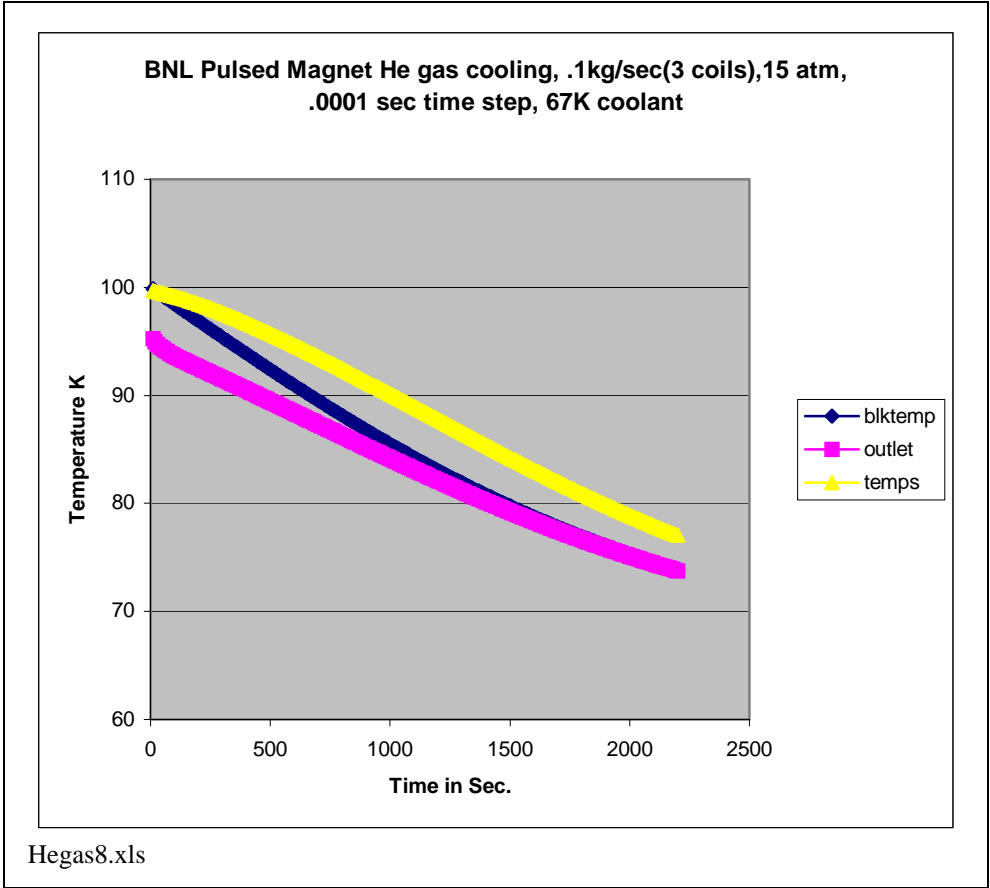
Another Check Calculation (To Be Added Later)

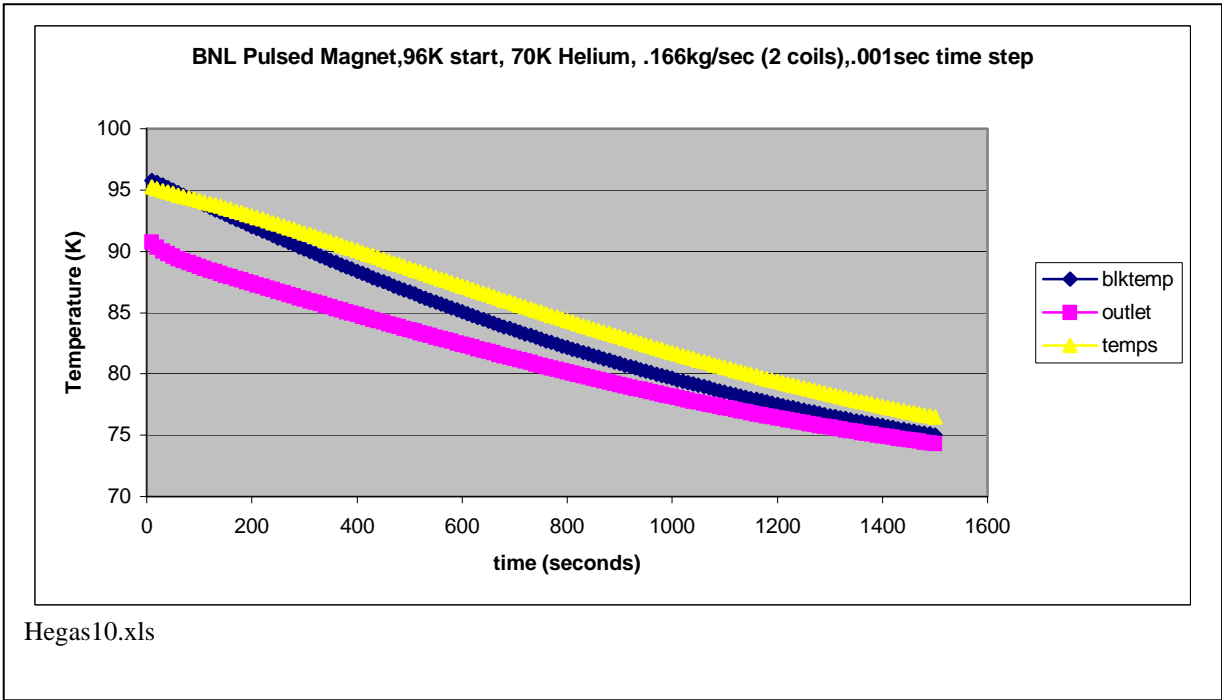
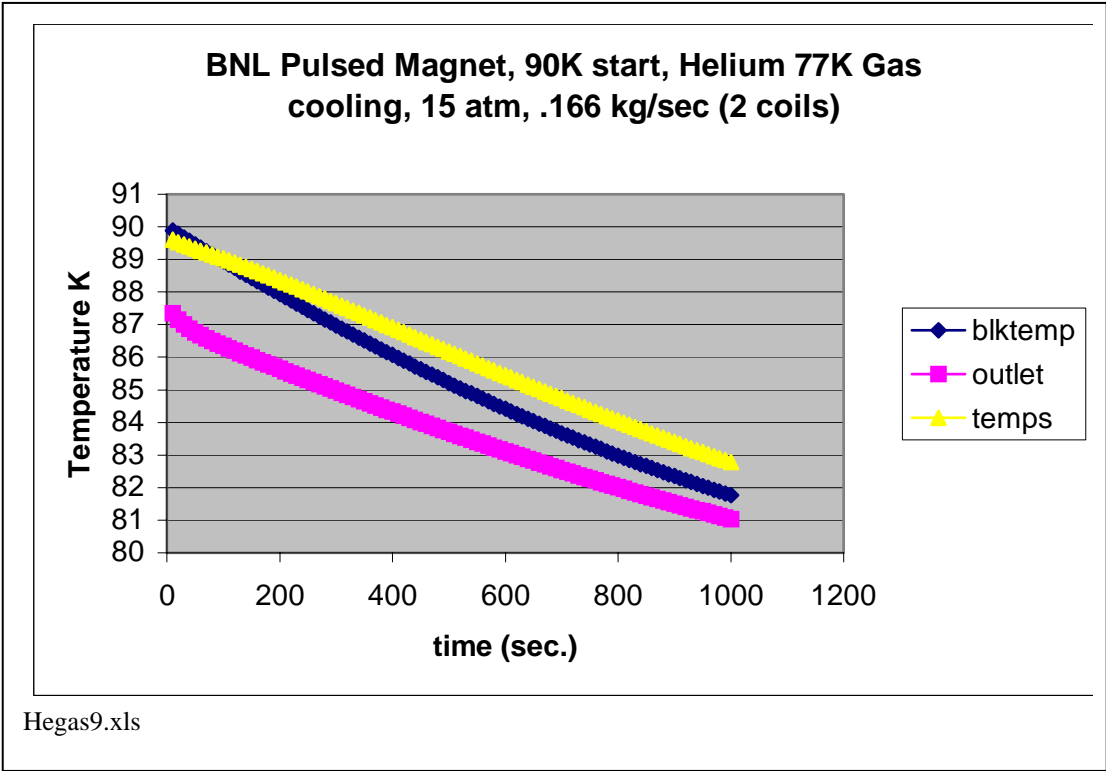
An FTM model with 1 mm elements modeling the kapton was run. 14 W/(m-K)
 The effective thermal conductivity for 1mm modeling three Kapton layers is:

$$.14 * \text{AREA(I)} / (\text{numkapt} * .001 / 39.37) * .001 = 1.83$$

7







15.6 Cryogenic System Peripherals – Vacuum Pump, Heater, Proportional Control Valve

Vacuum Pump and N2 Gas (and Residual LN2) Heater:

Vacuum Pump, and Inlet Heater

```
! ** Calculations ****
clear
let mflow=.05 !kg/sec Vacuum Pump Flow
let N2gasden=1/.7996 !kg/m^3 STP ref air liquide web site
let N2gasspht=1.04 !kJ/kg/degc ref air liquide web site
print "Gaseous Nitrogen Density=" ;N2gasden;"kg/m^3"
print "Gaseous Nitrogen Specific Heat=" ;N2gasspht;"kJ/kg/degC"
let N2gasden=1.25 !kg/m^3 STP ref air liquide web site
let vflow=mflow/N2gasden*60*60 ! cu meter/hr
print "mass flow=" ;mflow;"kg/sec"
print "volume flow=" ;vflow;"cu-m/hr"
let vflow= vflow*(39.37^3/12^3)/60/60 !cu ft/sec
print "volume flow = " ;vflow;"cu ft/sec"
let area6=.5^2*pi/4
let area4=.33^2*pi/4
print "Exhaust Pipe Flow Velocity, 4in
pipe=" ;vflow/area4;"feet/sec"
print "Exhaust Pipe Flow Velocity, 6in
pipe=" ;vflow/area6;"feet/sec"
let heatpower=mflow*N2gasspht*(292-88) !kJ/sec or KW
print"Heater Power=" ;heatpower;"kW"
end
```

.05 kg/sec of LN2 is 144 cu-m/hr gas flow at RT and 1 atm

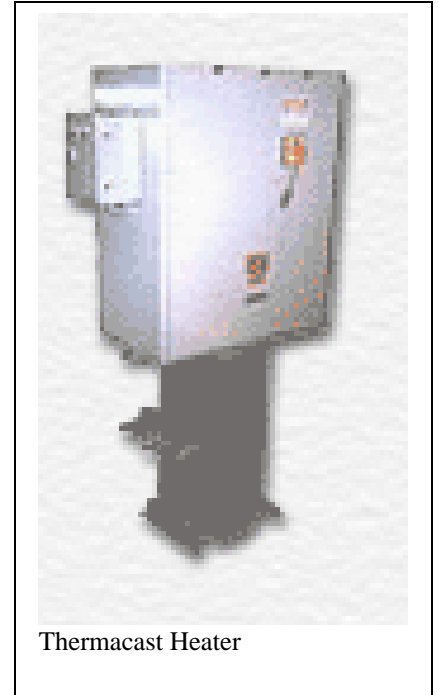
Exhaust Pipe Flow Velocity, 4in pipe= 16.515614 feet/sec
Exhaust Pipe Flow Velocity, 6in pipe= 7.1942017 feet/sec
Heater Power= 10.608 kW

Inlet Heater

For a standard 12KW he provided a budgetary price of \$4600. Because of the low pressure we might need a larger heat transfer area. This was estimated at \$8500. It looks like this would have to be engineered a bit. The unit size is 2' by 1' by 4' tall. So we can get it into our lab. Their web page is at:

<http://www.thermaxinc.com/indirect.htm>

Thermacast™ Electric Vaporizers and Trim Heaters



16.0 Instrumentation

<http://www.lakeshore.com/temp/sen/crtd.html>

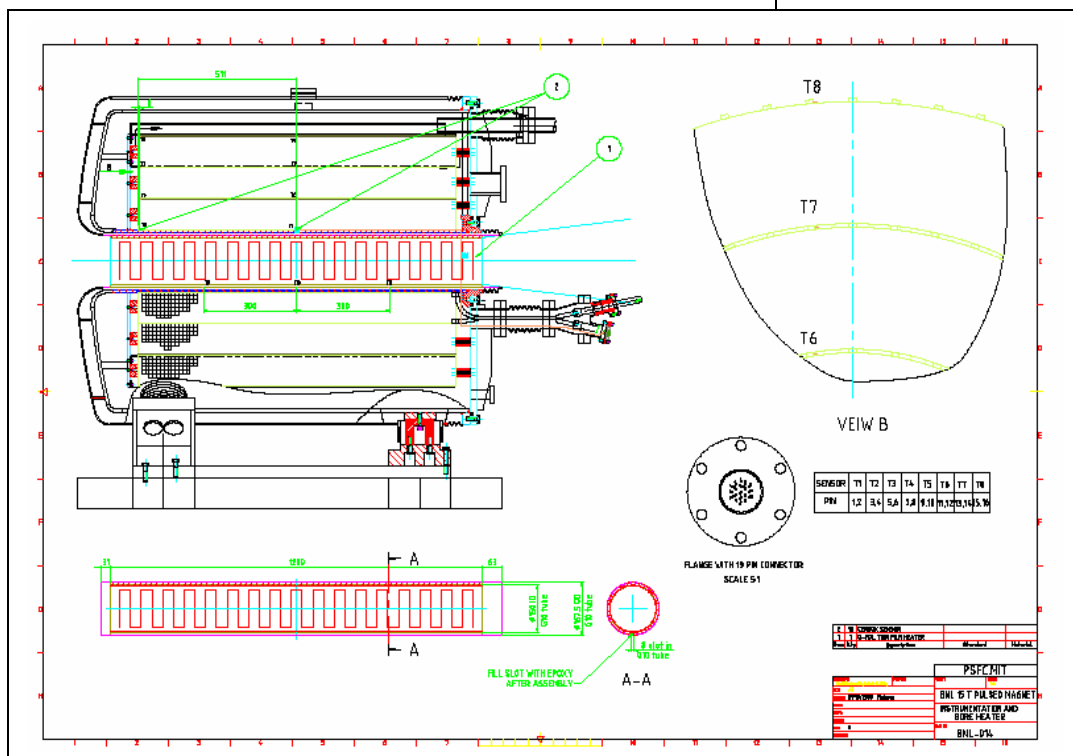
Cernox

thin film resistance temperature sensors offer a negative temperature coefficient, monotonic response over a wide temperature range, low magnetic field induced errors and high resistance to ionizing radiation. Instrumentation to read temperatures from these sensors may be found on the LakeShore site, above.

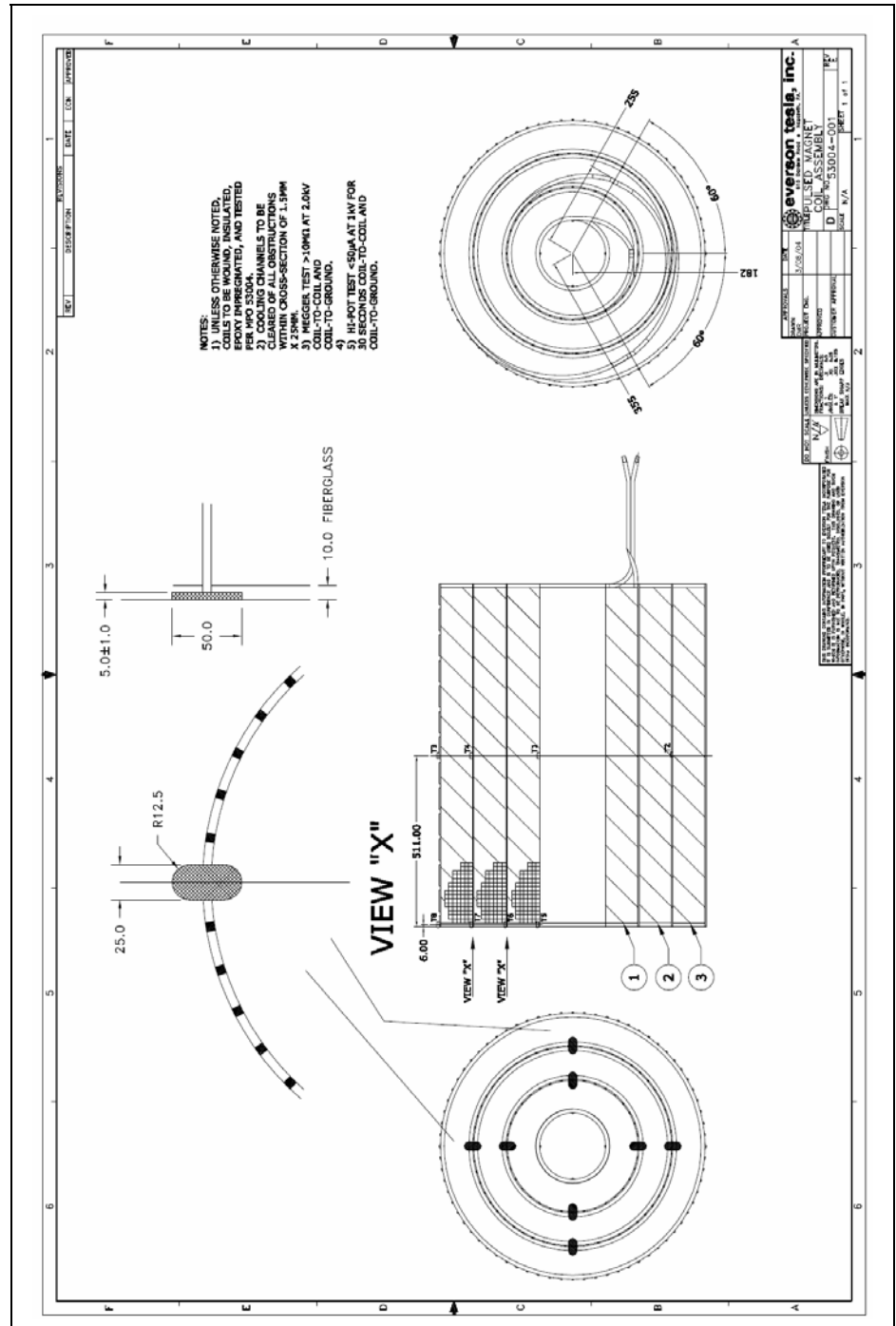
- Low magnetic field-induced errors
- High sensitivity at low temperatures and good sensitivity over a broad range
- Excellent resistance to ionizing radiation
- Fast characteristic thermal response times:
1.5 ms at 4.2 K; 50 ms at 77K (in bare chip form in liquid)
- High Temperature Cernox offers a wide temperature range from 0.3 K to 420 K
- Broad selection of models to meet thermometry needs
- Manufactured by Lake Shore, insuring control over wafer level quality and yield for the future
- Excellent stability
- Variety of packaging options



ISI Insulator Seal

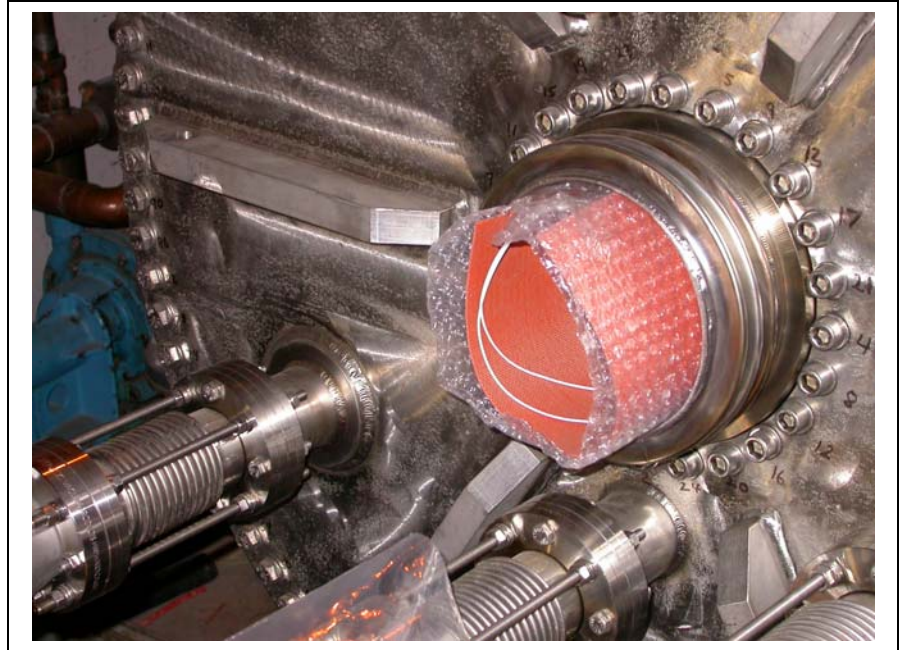


There are 8 Cernox temperature sensors inside the cryostat, requiring 16 pins of a 19 pin penetration. Three are "spares" T9 through T11 instrument the bore liner to allow control of the thin film bore heater. The bore is to be maintained at room temperature. These do not have electrical connections that pass through the pressure boundary. BNL-002 shows two of Item #35, the instrumentation penetration. The second pin/instrumentation flange is a spare, and is intended to accommodate other possible instrumentation BNL may want to add at a later date. The pdf of the instrumentation flange layout in BNL-014, shows 19 labeled pins matching the pin-out specs in the table.



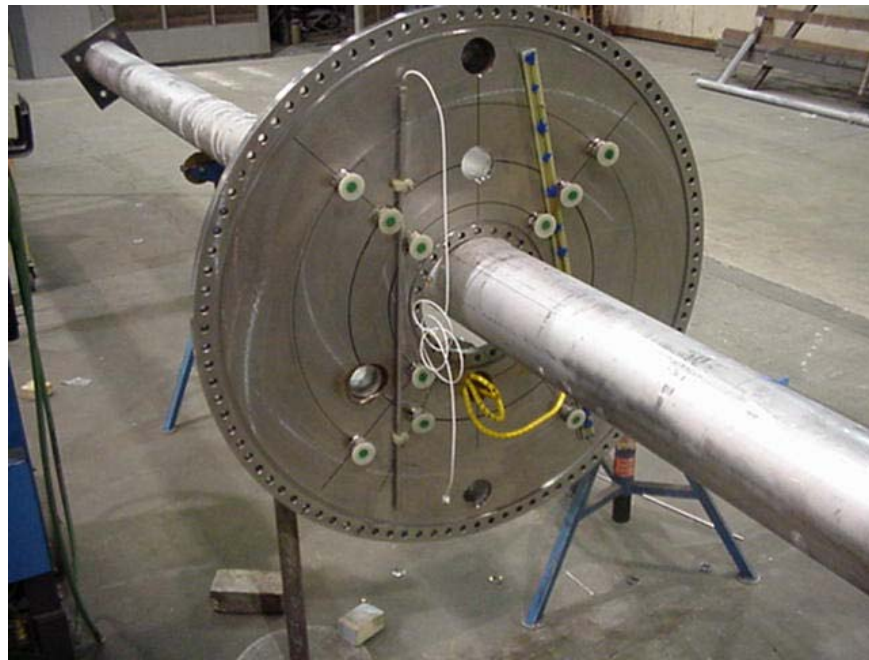
Bore Heater

Bore Heater Silicon Sheet, 300 Watts (Max) 115v. Only installed at the cold end of the bore tube. Manual variable power supply will be used



Jim Zapp
CVIP, Inc.
T (610) 967-1525 x11
F (610) 967-0303
Cell (267) 372-4907
jimz@cvipinc.com

Tue 11/15/2005 11:36 AM



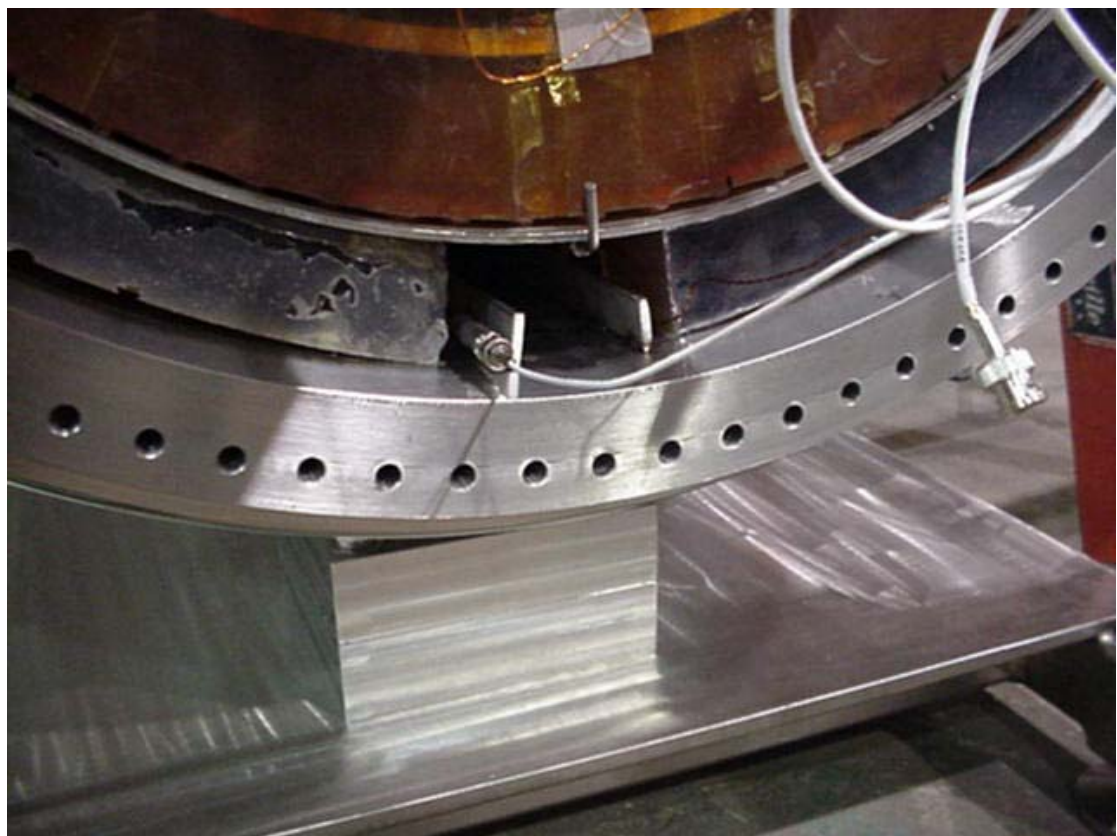
Capacitive level sensor on the left and the discrete level sensor is on the right.



Capacitive level sensor on the left and the discrete level sensor is on the right



Lower capacitive level sensor, before the cover was closed.

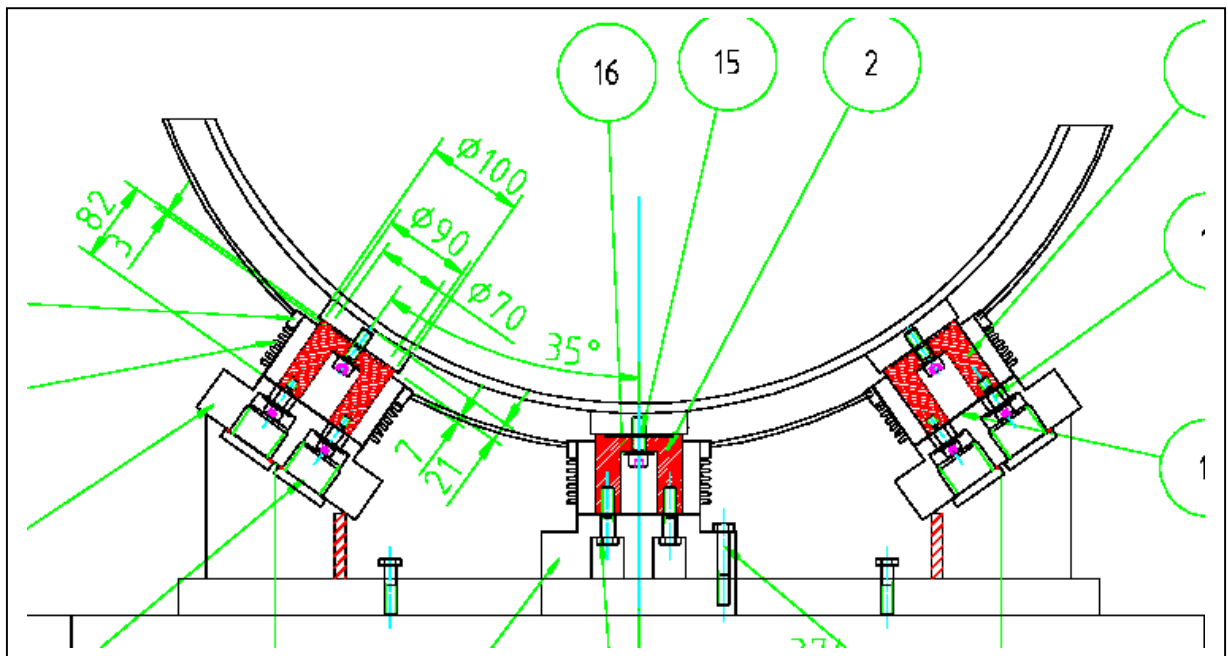
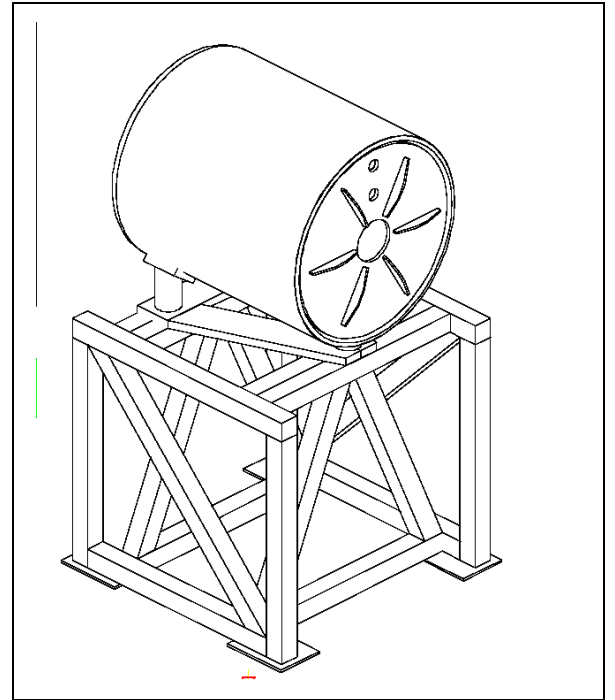


Lower capacitive level sensor, before the cover was closed

17.0 Support Frame and Gravity Supports

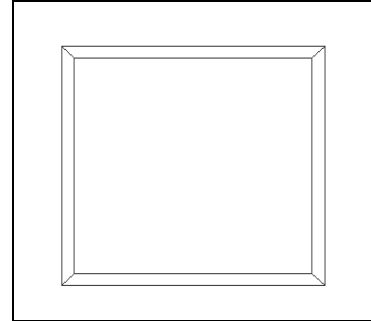
	Volume	Mass
Magnet Segment 1		748.04651
Magnet Segment 2		1246.7442
Magnet Segment 3		1745.4419
He Cryostat	.08727m ³	759kg
Vacuum Jacket	.01748m ³	152kg
Total	.	4651kg
		10253.6 lbs

The frame is made up of 4" square aluminum tubing. 4 verticals of 1.241m length, 2 diagonals of about 1.5m length, 4 longitudinal horizontal members of 1.168m length and 4 triangular legs supporting the center of about 1.0meter length, 6 horizontal transverse members of 1.168m length for about 24m of total length.



Element Group for which Section Properties are to be Calculated

Section Properties for Group Number: 0
AREA= 2.859375 IXX = 6.943665 IYY = 6.943665
MAX DISTANCE TO EXTREME FIBER, CX= 2.000000
CY= 2.000000
RINX= 6.943665 SNX= 3.471833 RAD OF GYR= 1.558328
RINY = 6.943665 SNY= 3.471832 RAD OF GYR= 1.558328
PRODUCT OF INERTIA ABOUT ORIGEN=RIXYT = -1.2011635E-07
PRODUCT OF INERTIA ABOUT NEUTRAL AXIS=IXYN = -1.2011635E-07
ROTATION ANGLE TO PRINCIPAL MOMENTS OF INERTIA = -13.36959



If only the four corner vertical posts supported the weight, the P/A Column Stress in the frame= $10253.6 \text{ lbs}/(4*2.859)=896 \text{ psi}$. Most of the vertical load however, will be carried by the inverted “V”s . The frame is over built, but there may be unidentified side loads, for example attraction to iron nuclear shields, and there the frame can be used to mount fluid and electrical connections.

18.0 Manufacture and Assembly

18.1 Initial Winding Approaches

Winding Procedure Specification Content:

The seller is responsible for selecting the winding process that will best achieve the required coil geometry, insulation configuration, and impregnation quality. A possible winding method is described here. The bidder shall describe his proposed winding procedure, and any change to the purchaser's suggested procedure as a part of bid proposal. The Seller's winding procedure shall be submitted to the purchaser for review and approval. Prior to purchase of the conductor, the Seller shall perform a test bend of a sample length of conductor, over a mandrel or bend fixture which has minimum radius required for winding the coil segment. The conductor sample shall have the same physical properties, - yield strength ultimate and % elongation, as the specified conductor. The test sample of the conductor shall be wrapped with glass tape, and the mandrel surface shall have a Kapton sheet applied. Cuts in the Kapton, or tears in the fiberglass tape shall be reported to Purchaser for resolution.

Options for "Resolution" :

- Relax cold work
- Increase corner radius
- Pre bend over shaping rollers

Assembly and Manufacture

The Coil is layer wound

The Coil is made in three segments. Phased manufacture is allowed

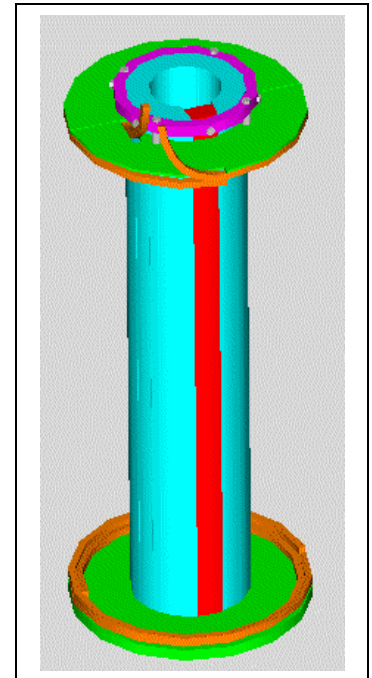
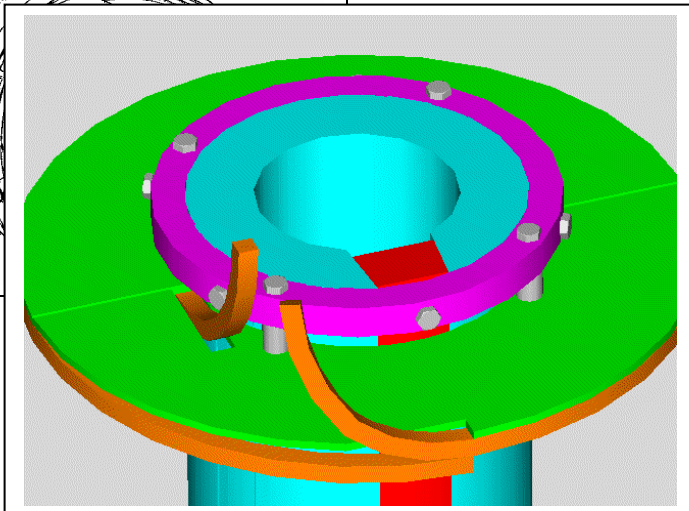
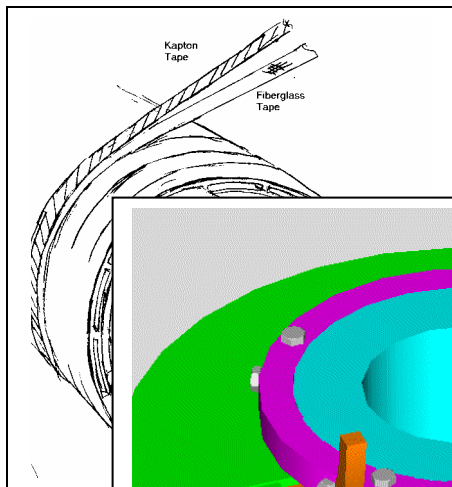
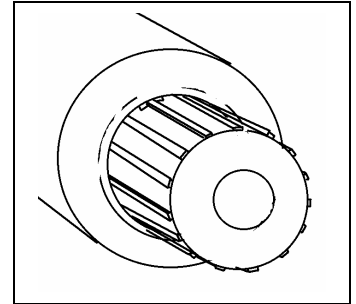
Three separate mandrels are planned.

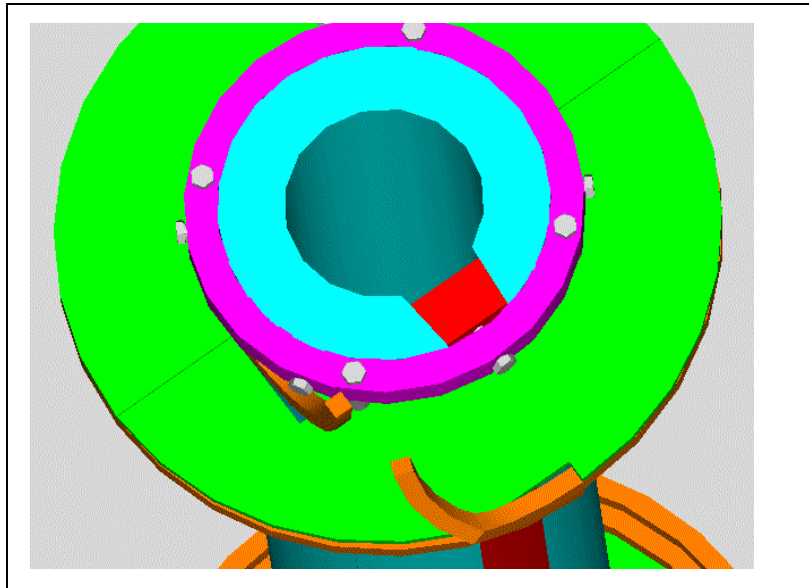
Mandrels maintain a precise bore geometry

Ribs are applied to outer surface of the wound and impregnated coil

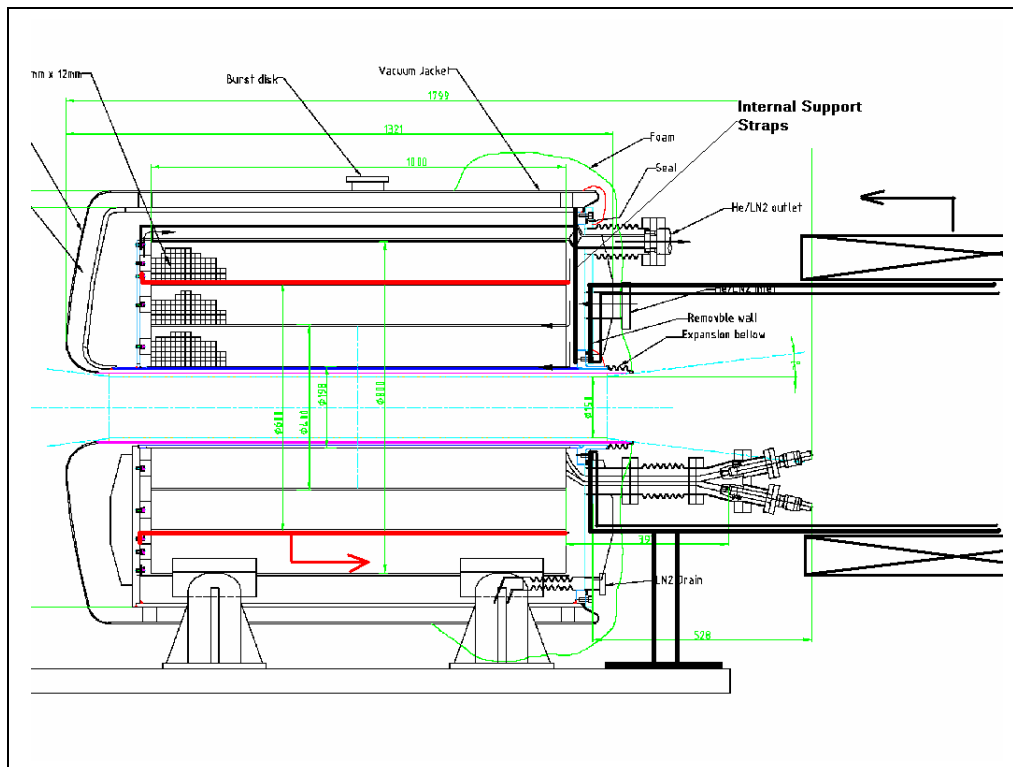
Ribs are machined to match the ID of the next coil segment

Coils are slipped on to one another. – with a temperature difference if needed





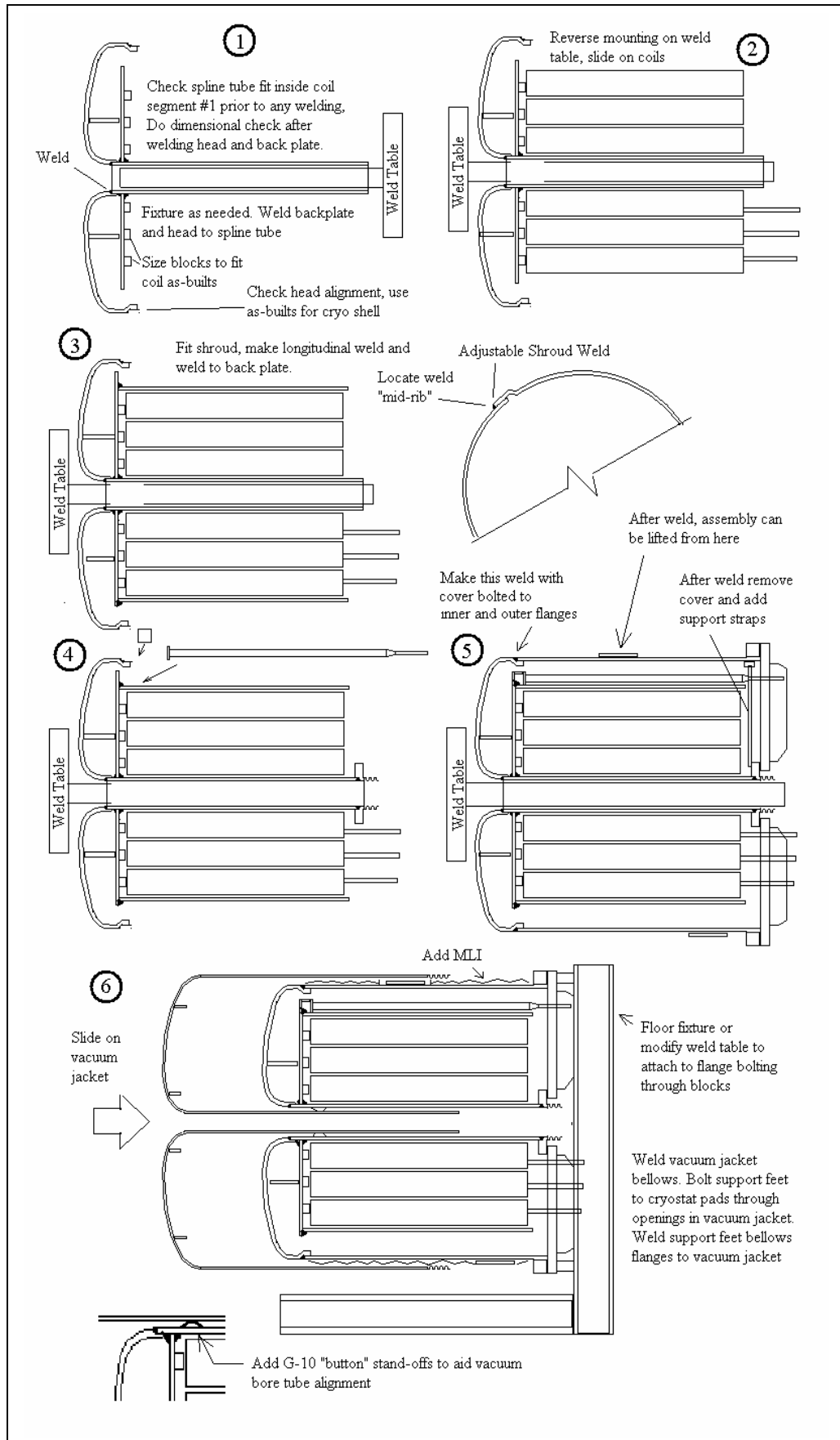
Phased fabrication of the coil segments is the motivation for the mechanical head closure.



The joint flanges are loosened, bellows compressed, and ceramaseal penetration set screws are loosened. The insulating penetration connections are removed. Head lip seal welds (if used) are cut. The closure head is removed. Internal support straps hold the coil weight. The assembly shell is installed, bolted to the inner head bolt circle, Temporary supports/cribbing support the shell and coil weight. The internal support straps are removed. The coolant shroud (red) is removed swapping inner and outer temporary supports. The third coil segment is then slipped into place. The internal support straps are replaced. The assembly shell is removed. The head is replaced, and bolts torqued. The lip seal welds (if used) are made. The ceramaseal connections are re-made.

18.2 Initial Vessel Assembl Approaches

This was an MIT proposal. CVIP used a modified approach.



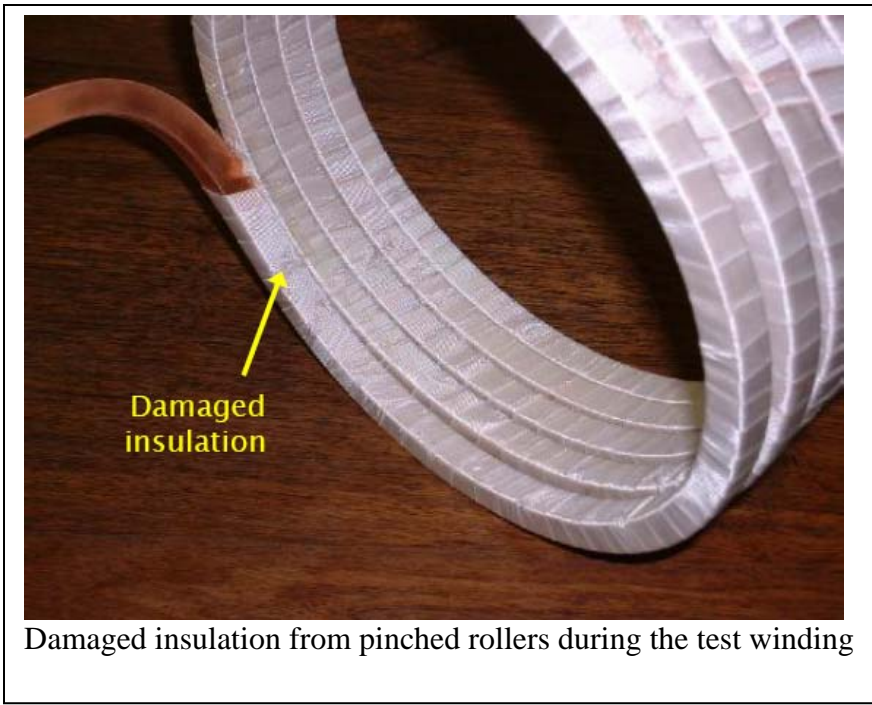
18.3 Magnet Manufacturing Experience

Trial Bending



This was a test winding done by Everson before material purchase, and before the final pricing proposal to CVIP





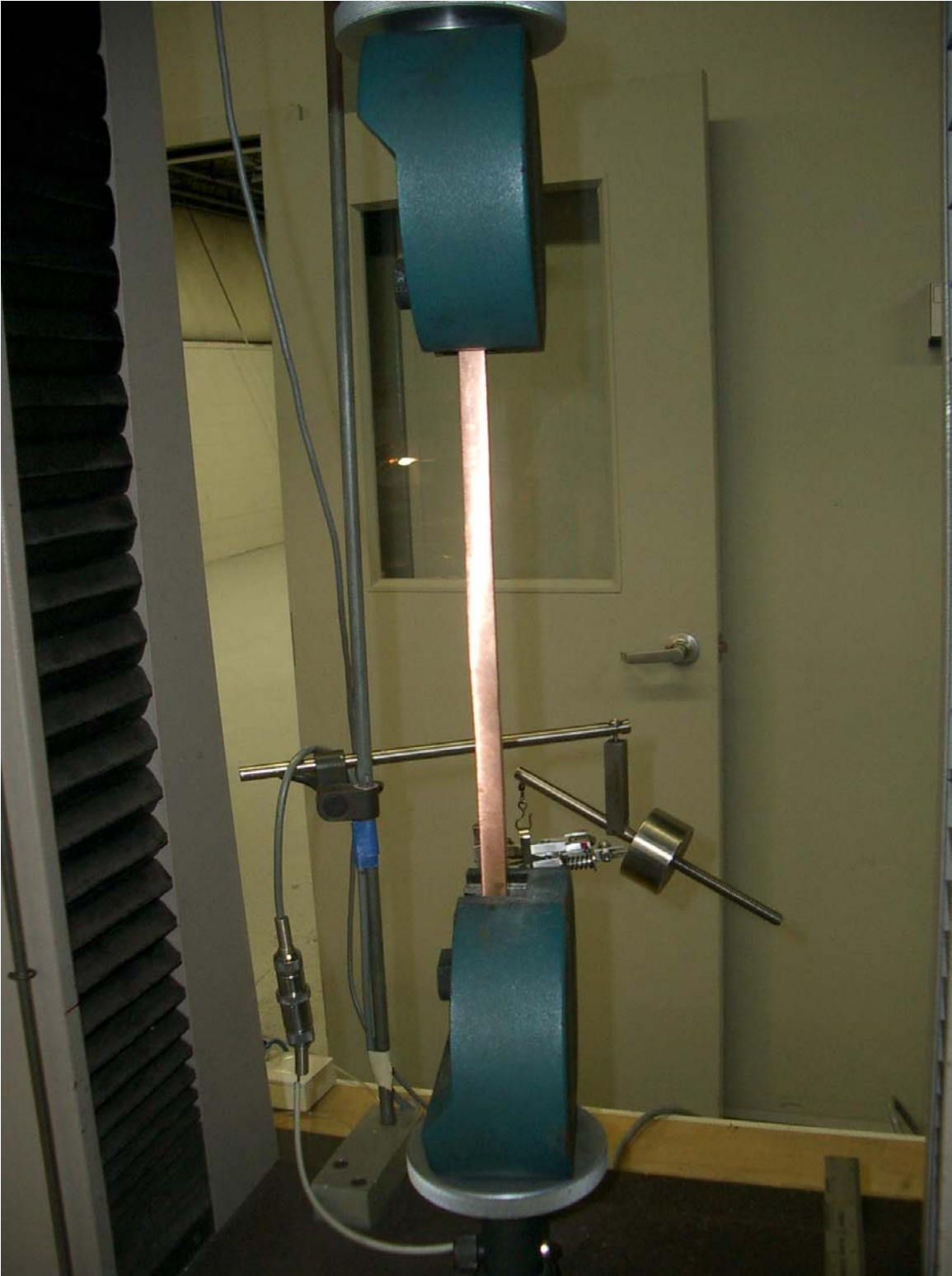
Damaged insulation from pinched rollers during the test winding



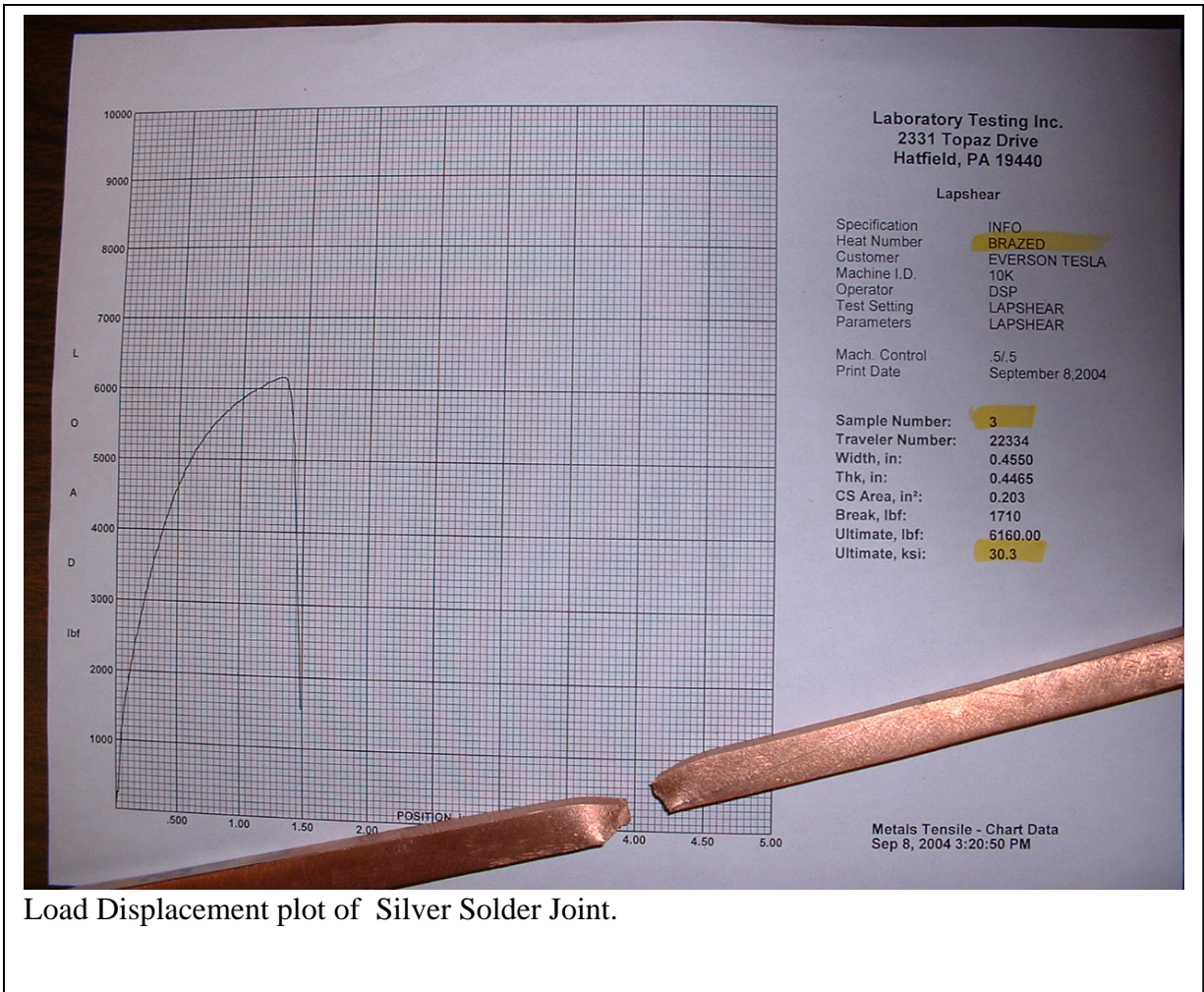
Braze Joint Qualification



BNL Pulsed Magnet Manufacturing Status Sept 8 2004
Faint indication of silver soldered scarf joint

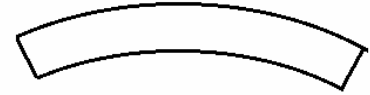
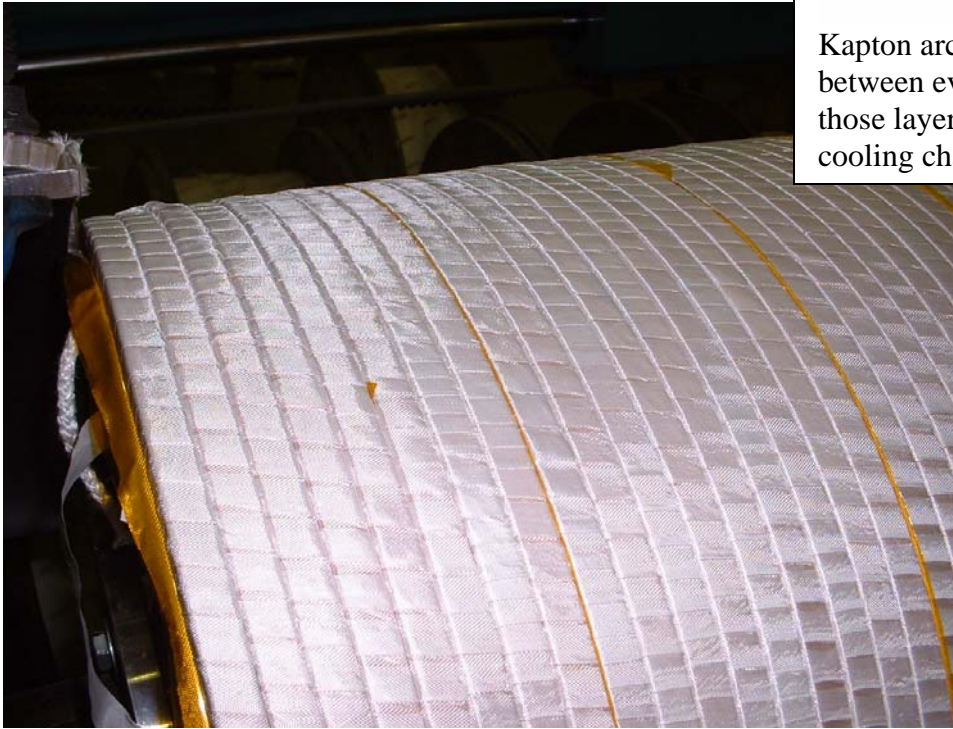


Silver Solder Joint Tensile Test



Load Displacement plot of Silver Solder Joint.

Winding Process

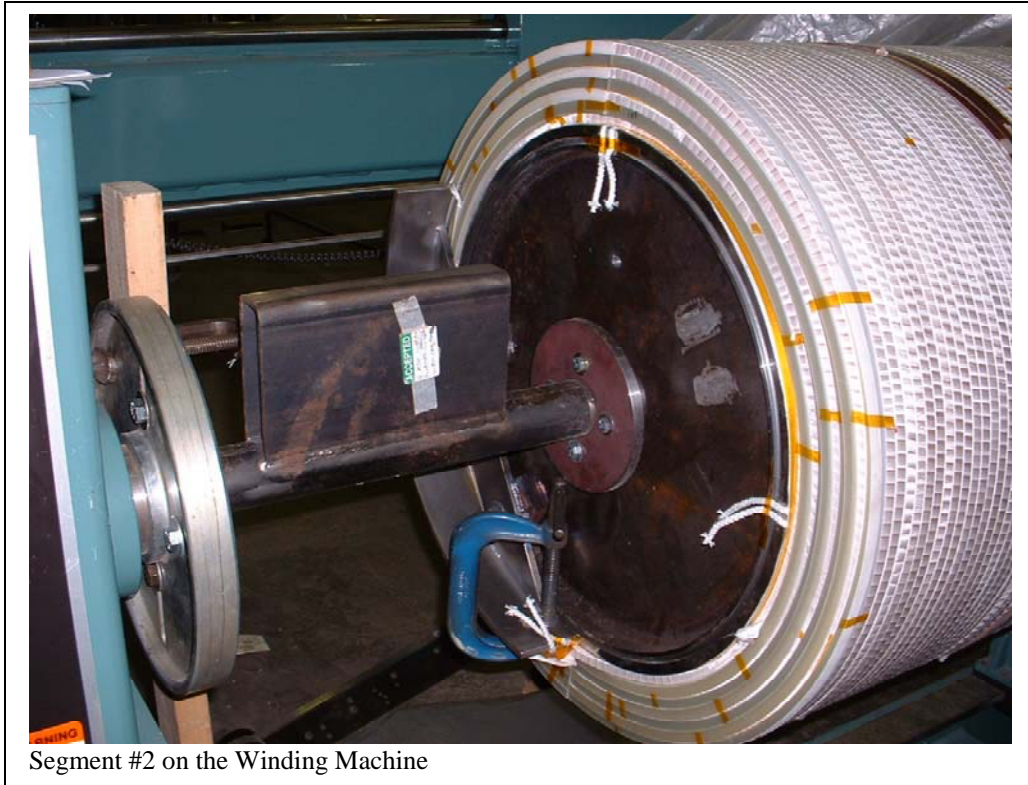


Kapton arc sections inserted between every eighth turn on those layers that face the cooling channels

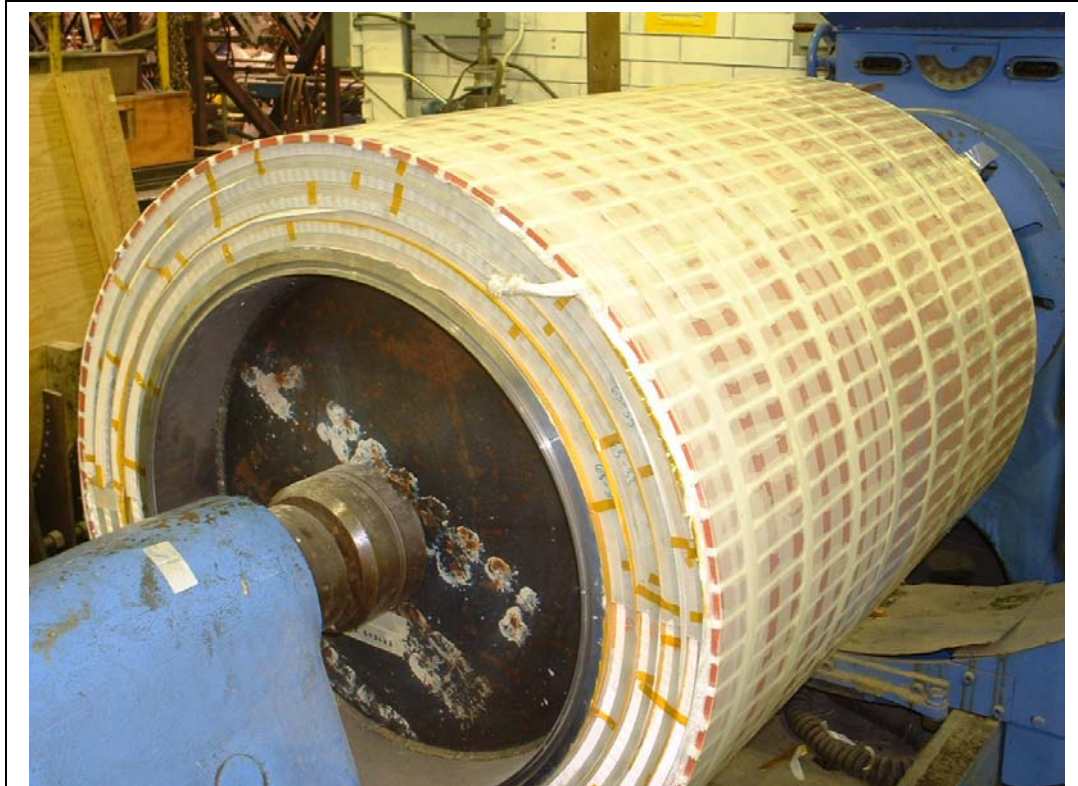
Segment #1 being wound. Photo taken by Dave Rakos at Everson 09-08-04. Kapton layer spaced at every eighth turn relieves axial tension in the layers near the cooling channels. First Layer, Coil Segment#2



Winding Segment #2



Segment #2 on the Winding Machine

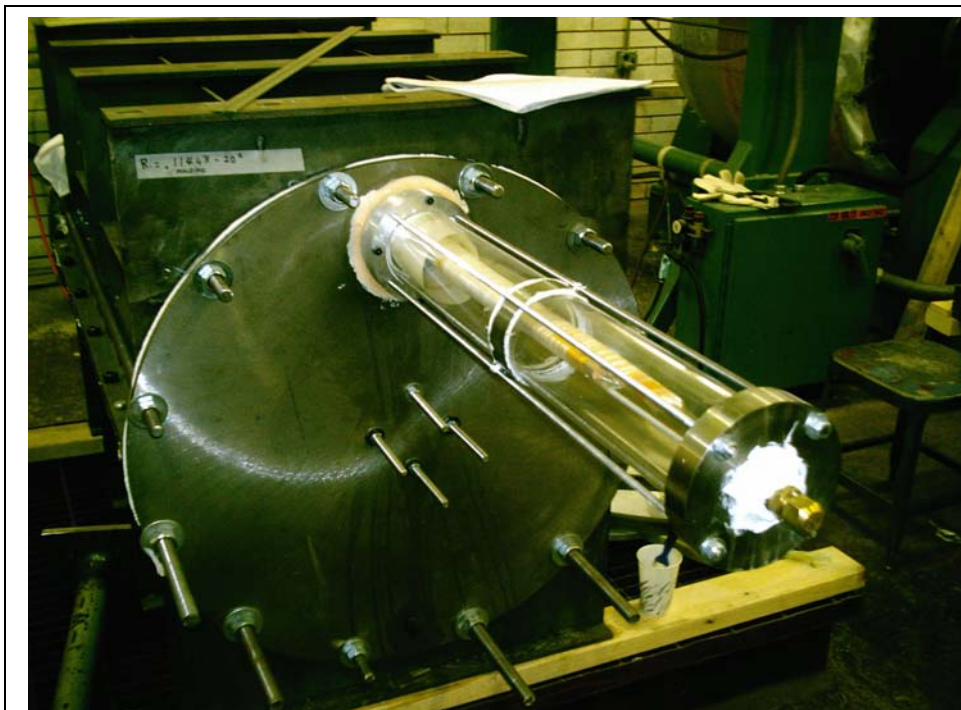


Segment #3 has been wound and the outer “waffle” pattern has been applied. This is scheduled for impregnation Feb 2 to 4



Segment 1 showing formed leads and portions of the mold

Impregnation Process



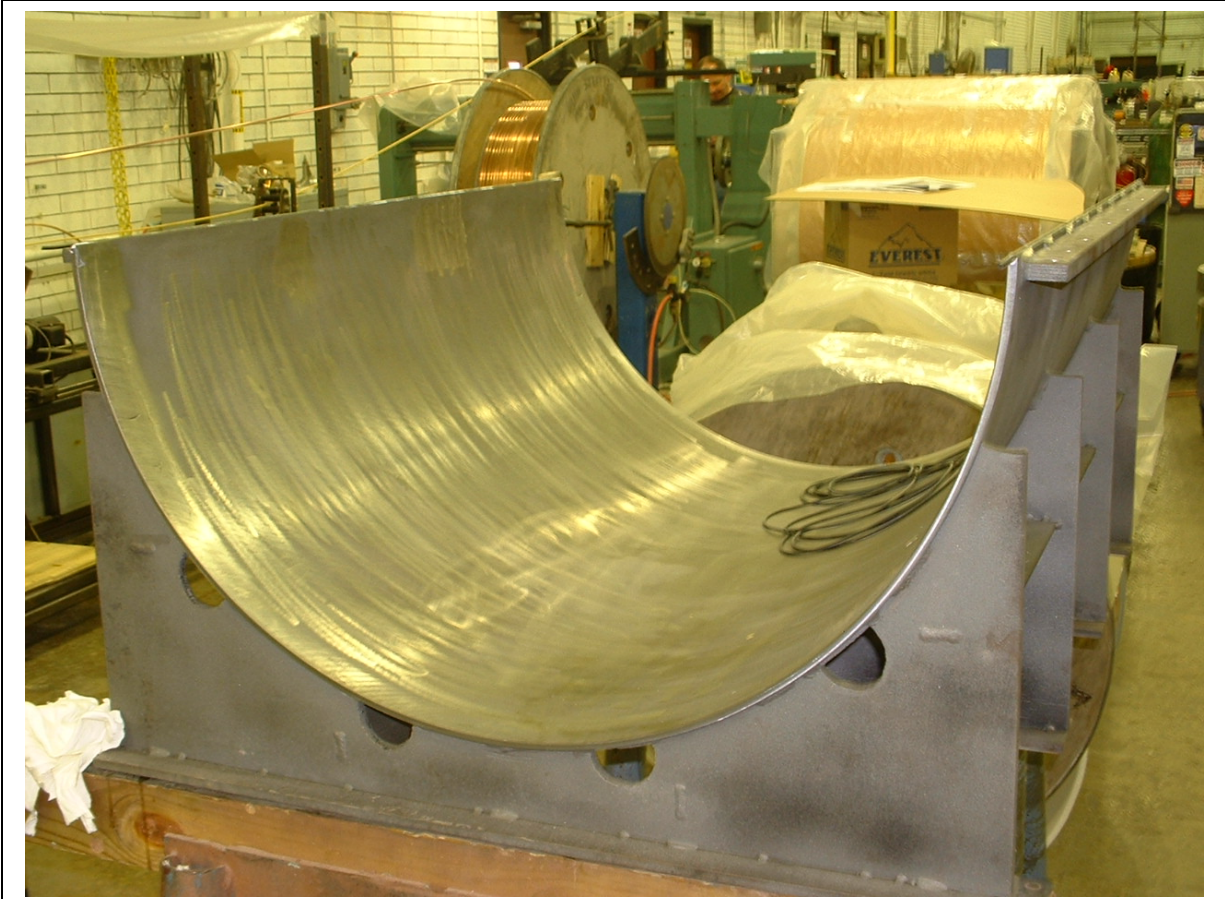
Glass Tube Surrounding the Joint Brea-Out, Coil number one. This is used as a sight glass to watch for bubbles during the vacuum application of the VPI process



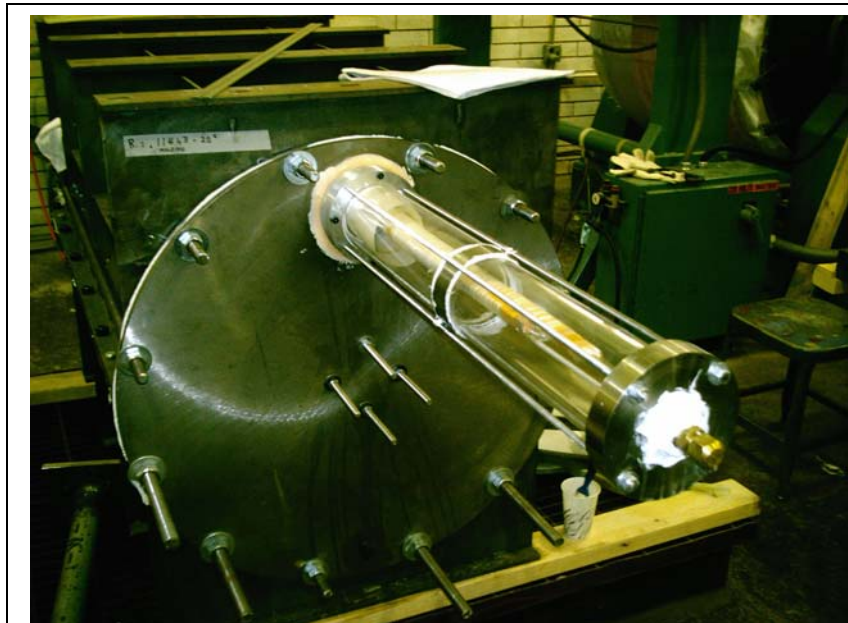
Lenny, an Everson Tech, watching the epoxy fill process
A vacuum is maintained on the mold (inside the autoclave box) with a vacuum pump (not shown) to the right of the autoclave). The epoxy tank is at an atmosphere gage. Ron adjusts epoxy flow manually. After the mold is filled, pressure is cycled to drive off bubbles. The mold is under pressure during the cure cycle to minimize the remaining bubbles.



Epoxy fill site glass. Everson doesn't have a good method to measure the flow going into the coil

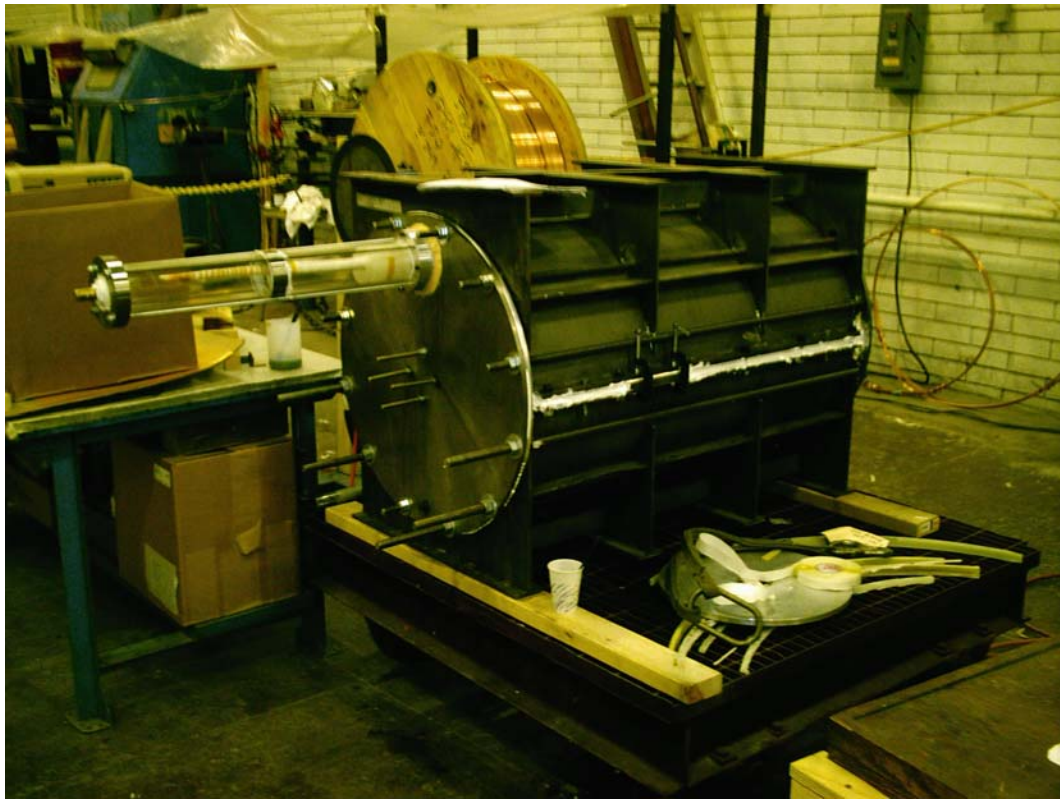


Segment #2 Lower Half of Mold



Segment 1 Mold showing sight glass

There was a leak in the longitudinal seam weld. This required application of RTV caulk and additional time to wait for the cure. This was to be held at 2 atm for 12 hrs prior to impregnation



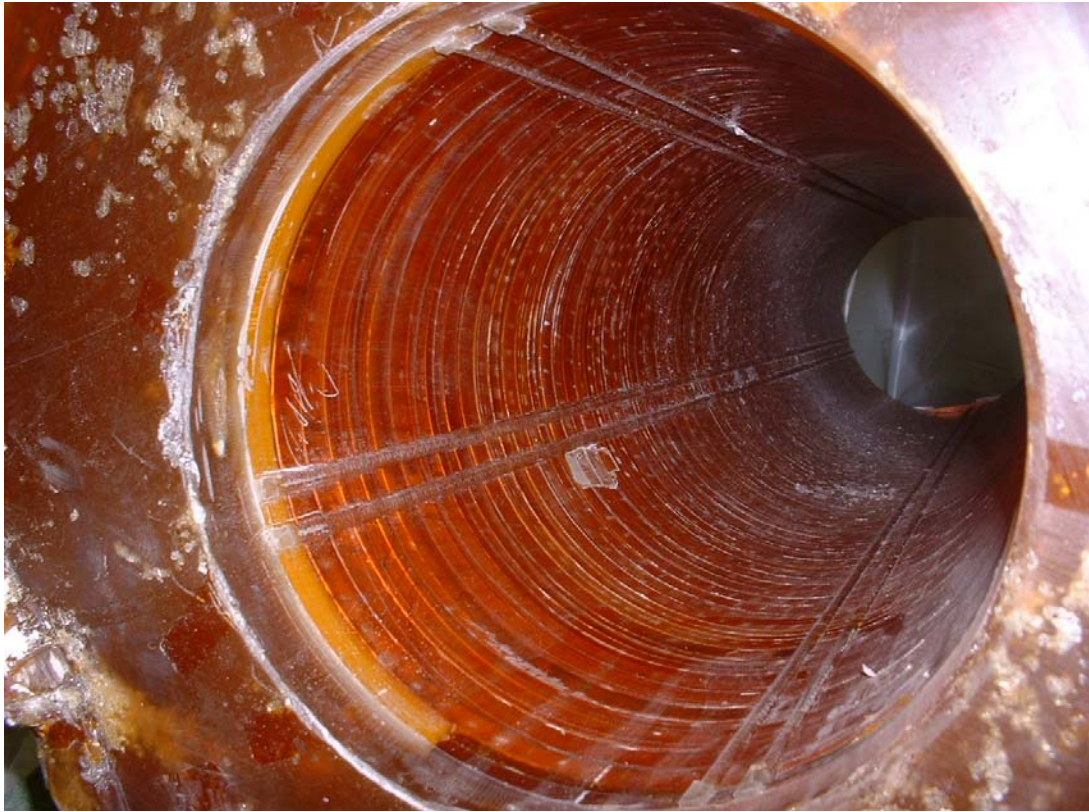
Segment 2 Mold

Impregnated Coils



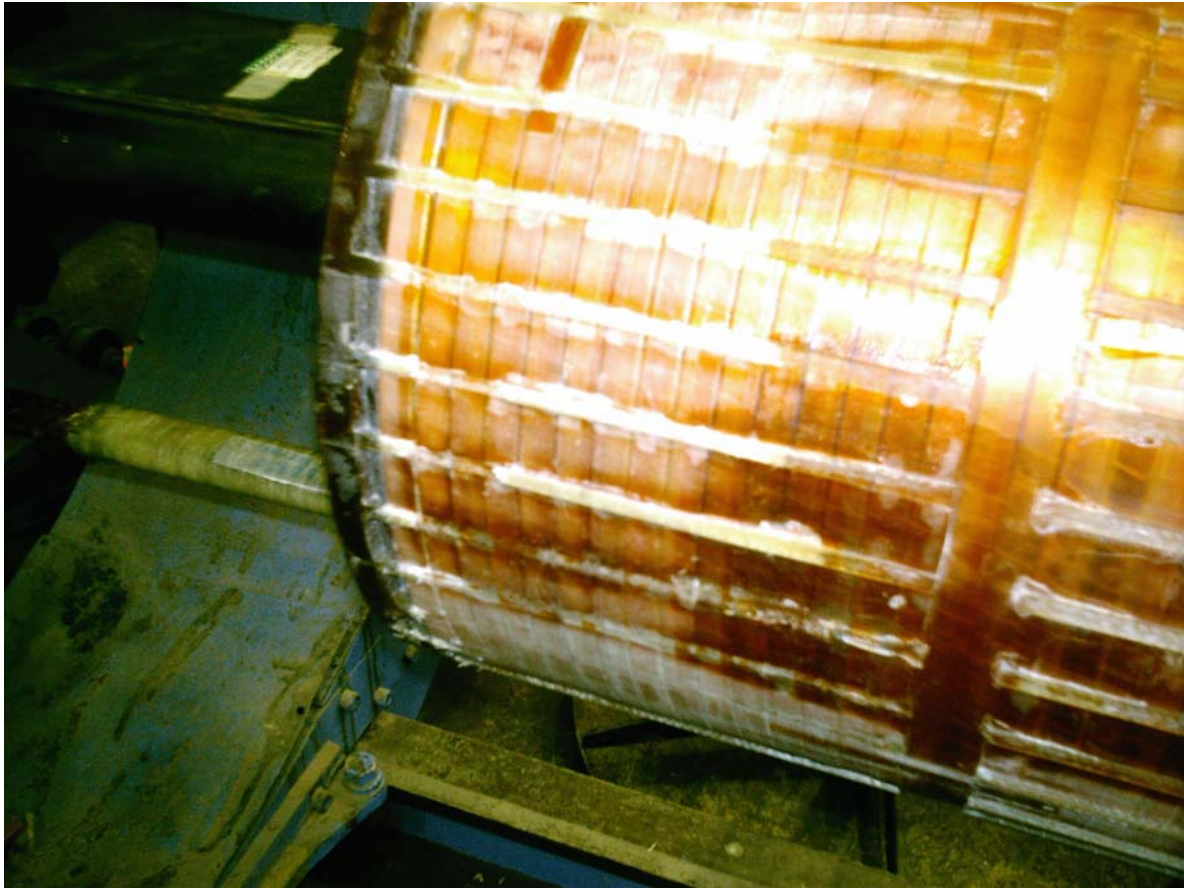
Segment #1 out of the mold. External Silicon fillers that form the “waffle” pattern have not yet been removed





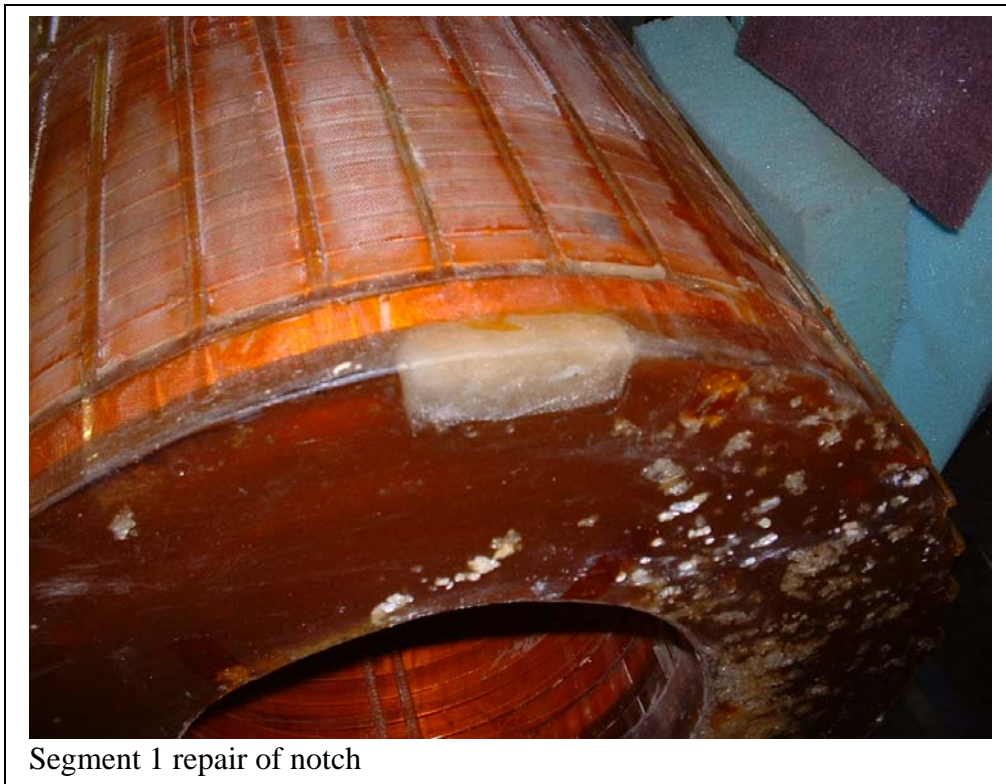
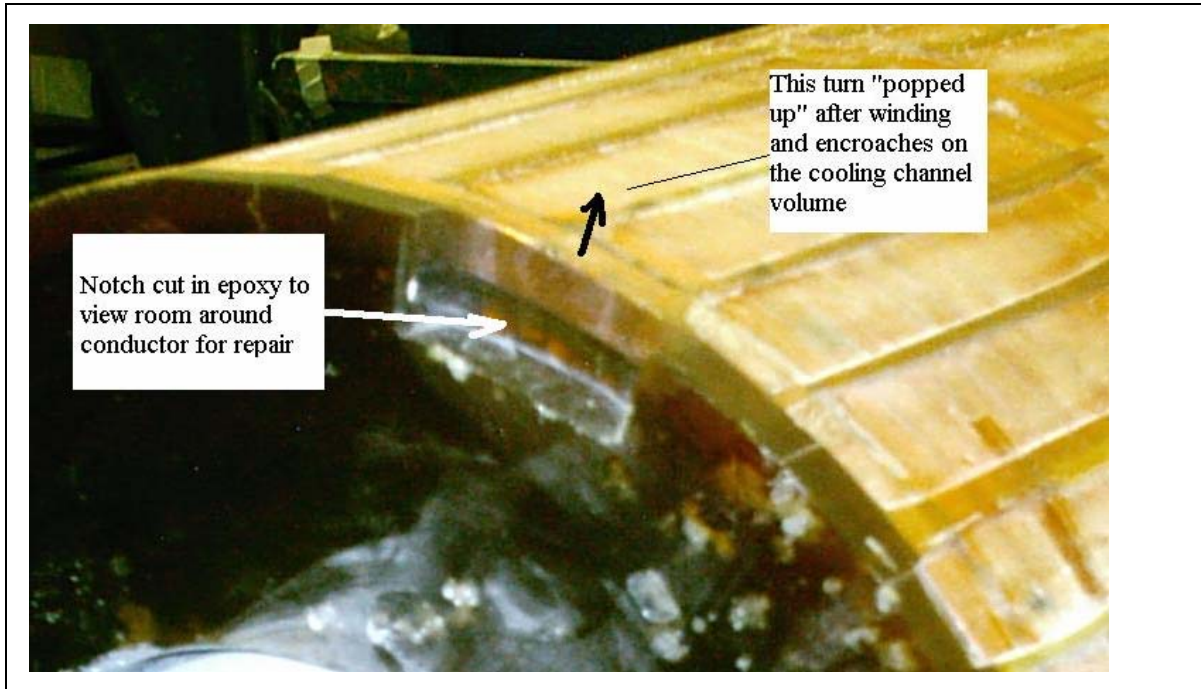
Segment #1 Bore – Will be fitted to CVIP's Bore tube

OD Machining Operations

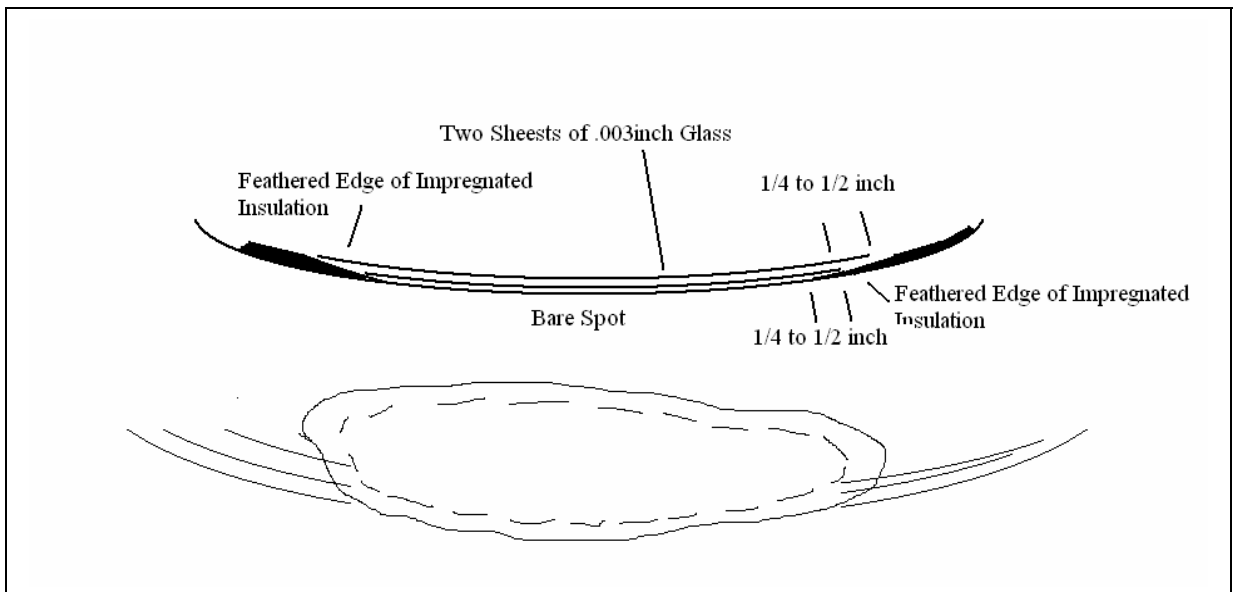


The initial OD Machining operation used by Everson was to turn the coil on a lathe. The impacts of tool against the ribs knocked off some of the ribs. The machining operation was changed to a vertical mill. This made the ribs actually flats but the chord error was small.

Damaged Turn



Failed Impregnation and Repair



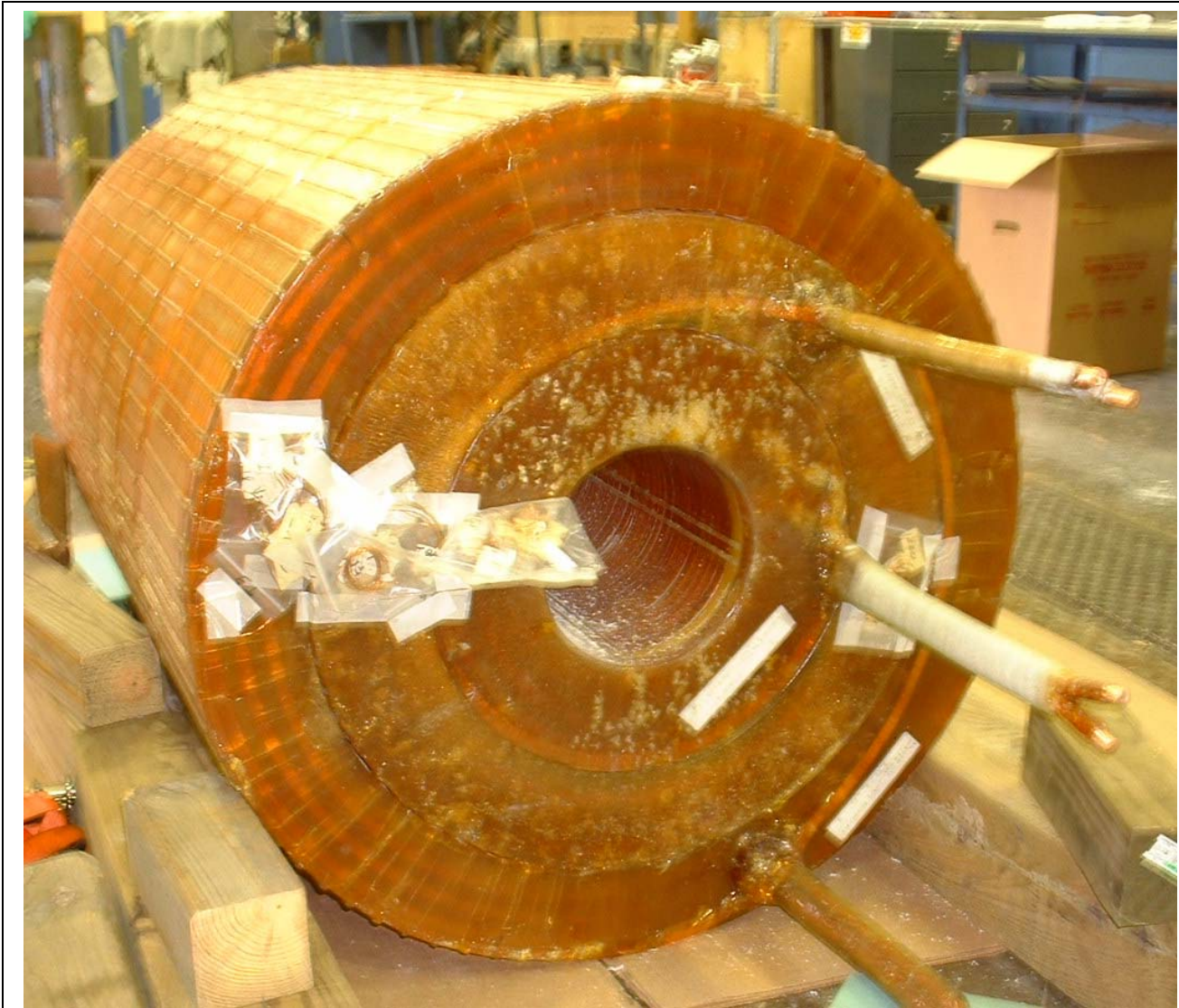
Nesting

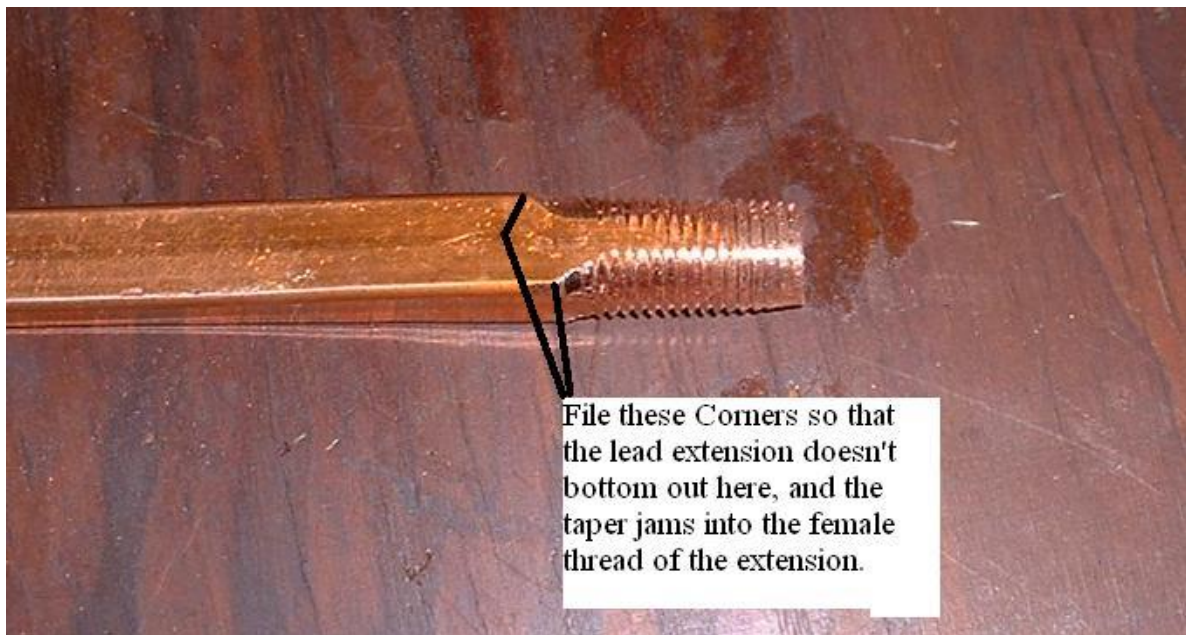


Successfully nested coils – Segment 1 in Segment 2



Three Nested Coils with Instrumentation





File these Corners so that the lead extension doesn't bottom out here, and the taper jams into the female thread of the extension.

Lead Pipe Thread



Leads threaded with the coils being prepared for shipping



17.4 Vessel Manufacturing Experience



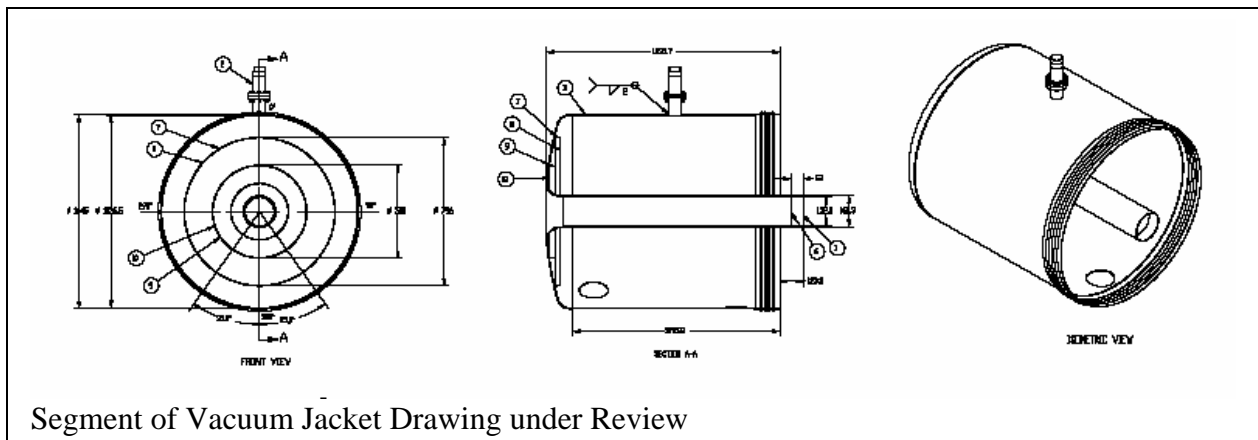
The Inner Cold Vessel at CVIP

Status of Vessel Drawing Submittal by CVIP

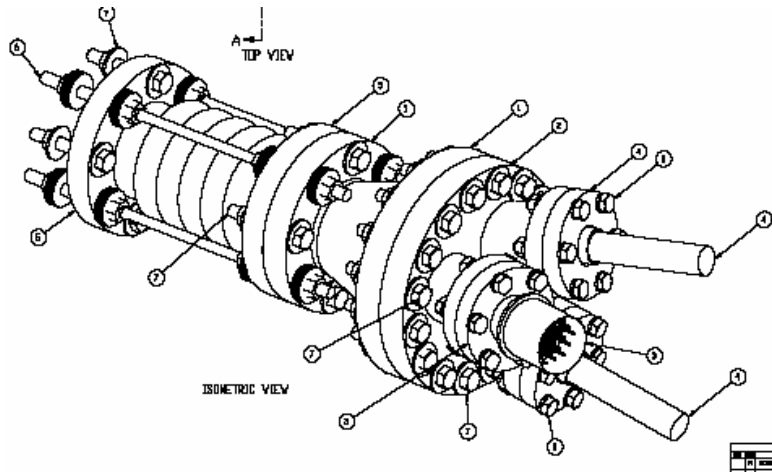
Inner Vessel drawings complete and nearly all approved.

Outer vessel drawings under review

Final manufacturing procedure is approved, but there are some final additions.



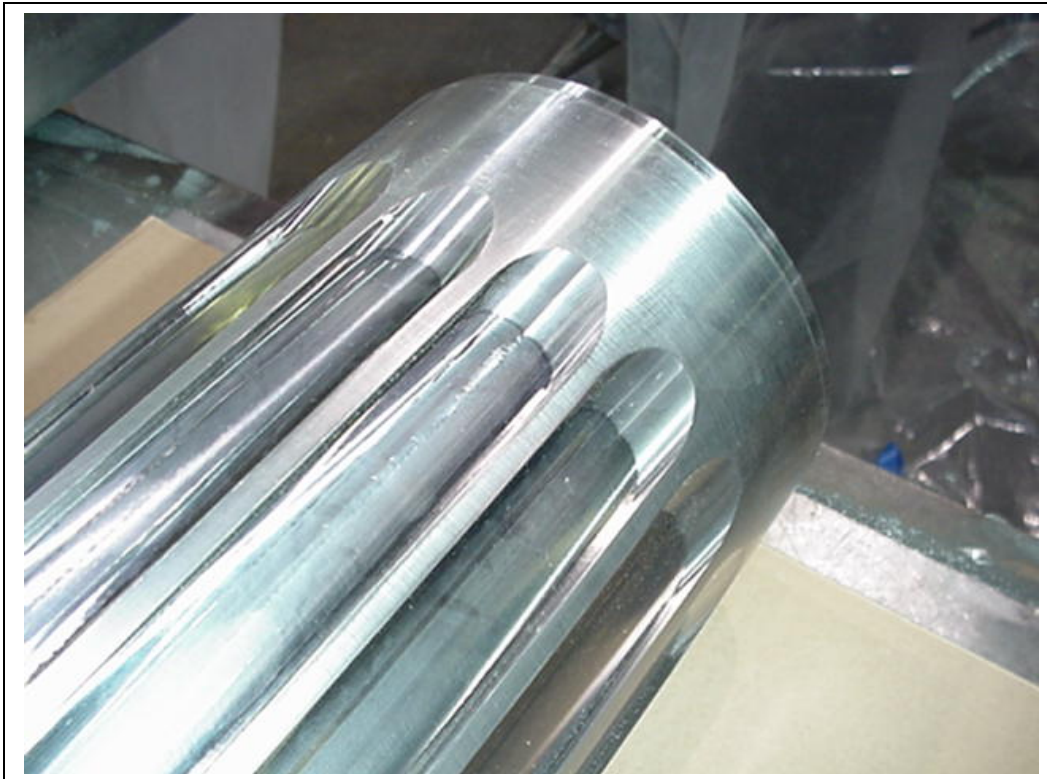
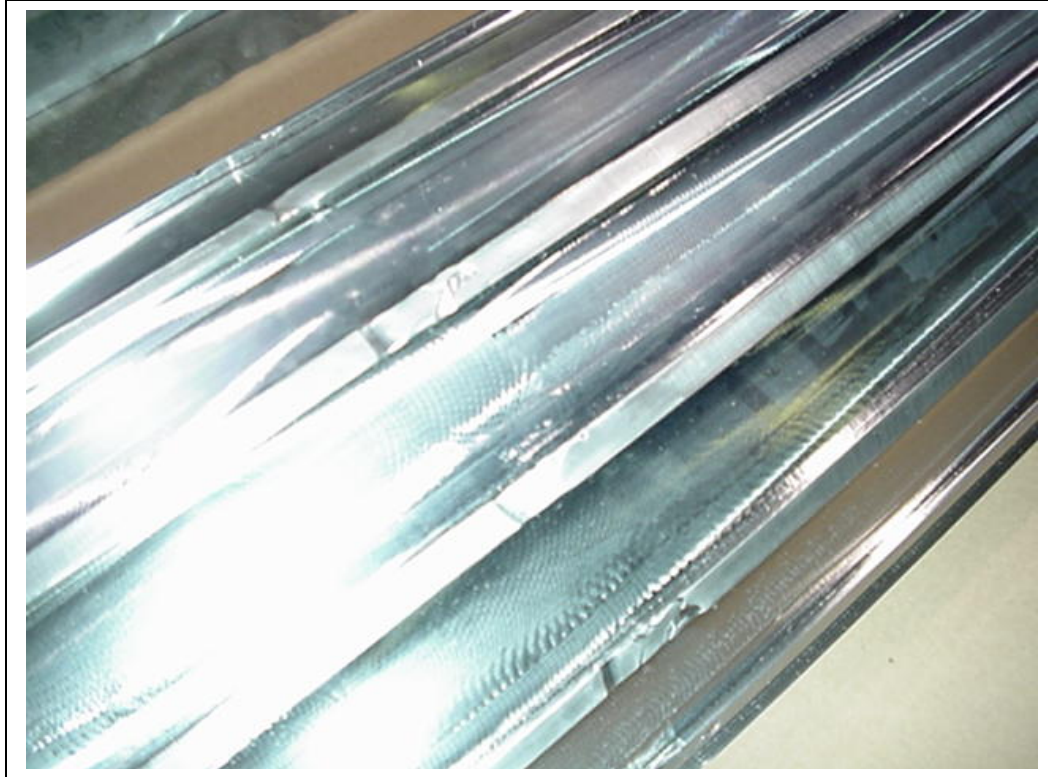
Segment of Vacuum Jacket Drawing under Review



Metallic seals for the cover flange



Outer Vacuum Shell at CVIP



OK- The only problem with deleting the glass bead is that the contraction of the epoxy is about twice that of steel. With the glass bead, the contraction is close to steel. As long as the fillers are not bonded to the steel it will be OK, otherwise we will have some scary noises when we cool down for the first time. In our larger experiment across the street they use quartz sand for filler, and haven't had problems with fouling valves, but they do have screens in the sump. Even pure epoxy has flakes and chips that can clog valves – especially when it cracks on cooldown. I think these should be flushed out during the tests at MIT. Eliminating the glass filler eliminates a concern that I didn't talk to you about: activation of any boron containing glass. So leaving out the filler eliminates one more type of material that might activate. "Epoxy only" filler in the dished head will shrink about 3 mm on the diameter with respect to the steel. You had talked about welding some studs or tabs to the head to hold the epoxy block to the head. The contraction could shear off studs. If they are closer to the ID they would probably just bend.



Attached are pictures of Epikure 3140 + Epon 815C epoxy sample. We did not use glass bead as filler because we think it may come loose and damage the valve seat of N2 flow control. It's also not recommended by the manufacturer. We are in process of filling the inner vessel head and forming the strips for filling void area between the cooling shroud and inner vessel wall,



Casting the fillers



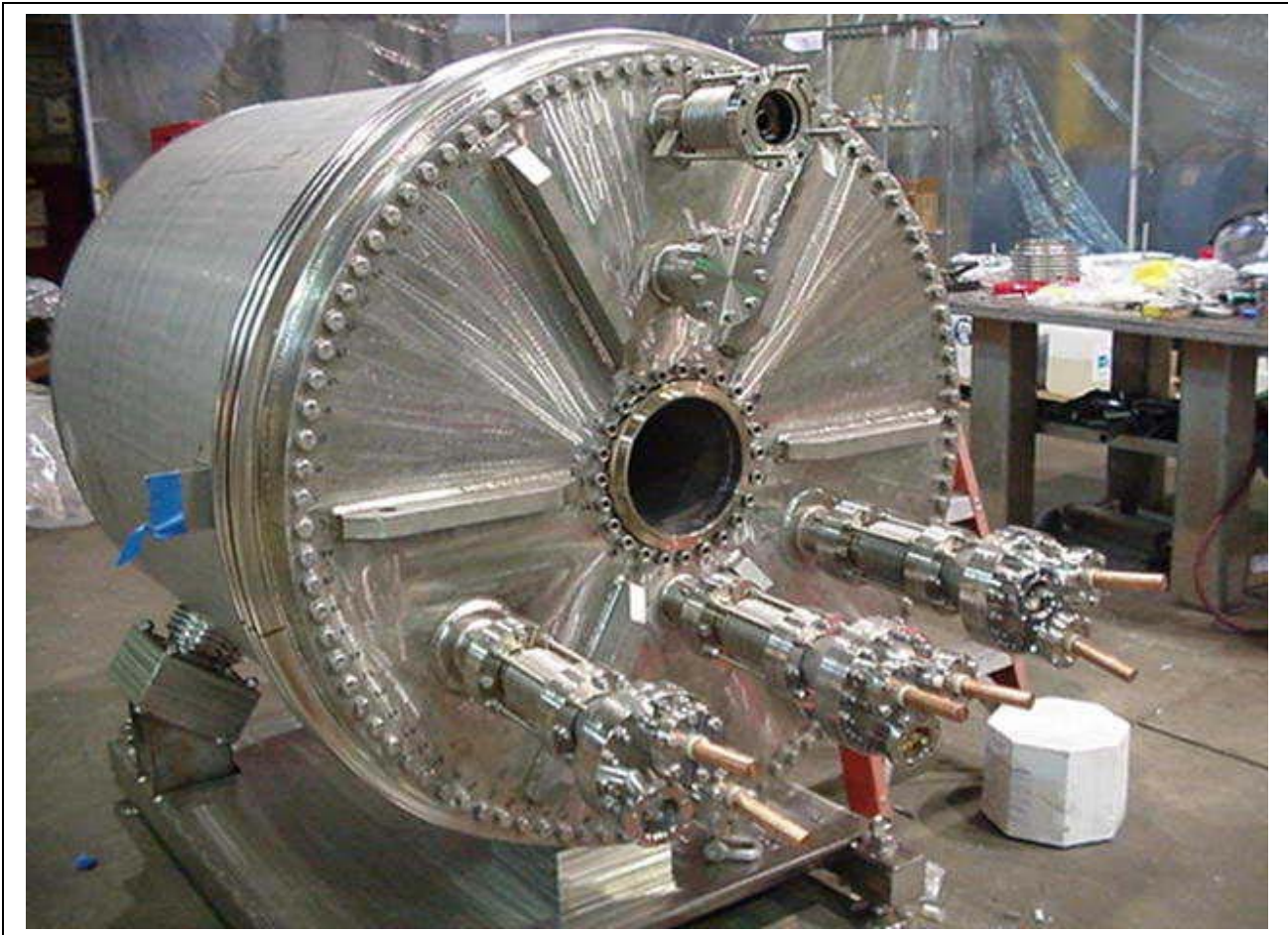


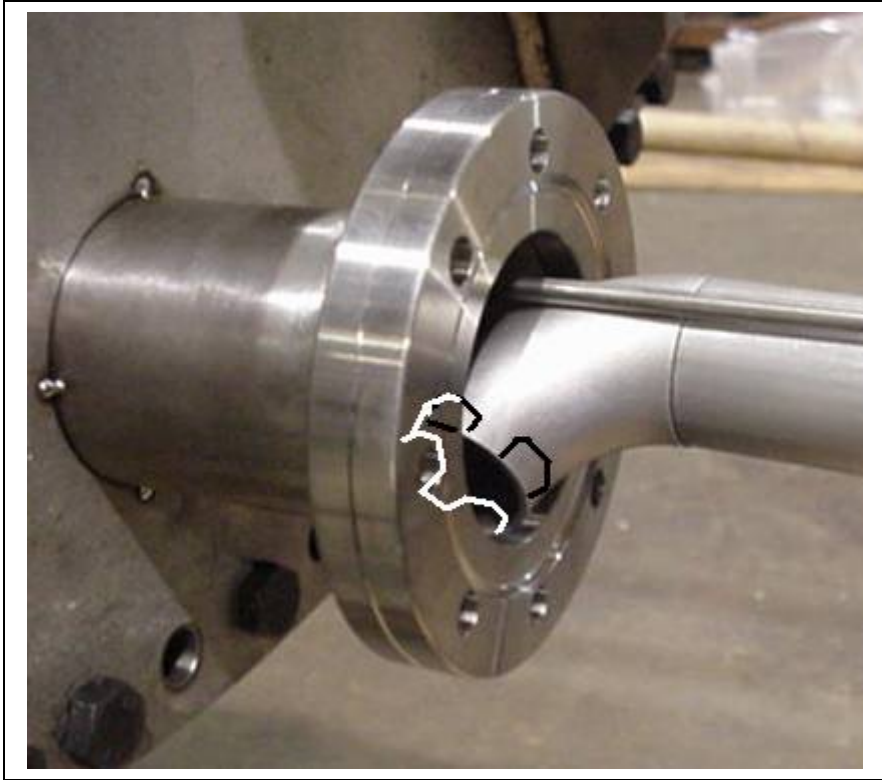
OK We will solder the connections. Please make sure they are bundled carefully, with the labeling tags intact. Tape them to the inside wall of the triport well enough so they don't shift when the magnet is shipped.

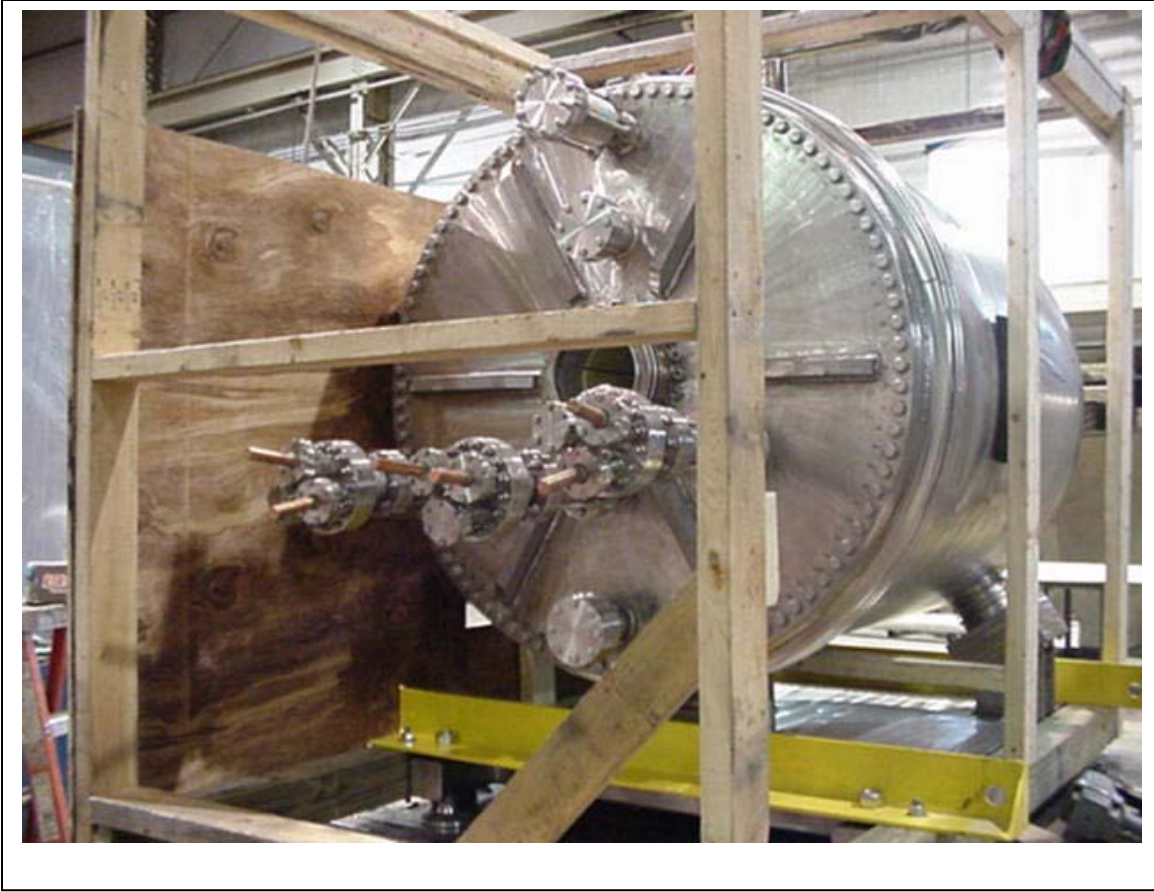
For documentation/contractual reasons, let me add some reasons why it is not as appropriate for CVIP to do the soldering:

The CERNOX sensors have 4 leads each. Two are to apply a current, and two are to measure a voltage. The CERNOX sensor current leads could be wired in series. However, we are not sure we have a current source that can supply the series voltage. We may have to apply current in groups of two or four. The pin count assumed two pins for all the current leads and then two pins for each CERNOX voltage tap. – However we would have instructed CVIP at this point, we might have had to redo the connections during the final configuration of the instrumentation. The discrete sensor was wired without a single lead for the common ground. Each diode has two leads. The common ground will have to be connected inside the instrumentation port to meet the pin count. This was my mistake. Friedrich has sent us the electronics, and we are going to test our understanding with the extra diodes and then wire the diode leads here.

I talked to Jim Zapp when I was at CVIP and he indicated that he hadn't bought the second 20 pin connector. CVIP should be supplying the second 20 pin connector. – This was in the change order. CVIP can test and ship with a blind flange for the second instrumentation port, but we will need the connector. Shipped with the final shipment or later.









19.0 Safety and Fault Analyses

Postulation of Safety Issues:

Failure of Bore Heater

Joint Failure

- Excessive motion

- Omission of a Force Component

Insulation Failure

Leaks

- He/LN2 Cryostat Leak

 - Mechanical Seal Failure

 - Bellows Crack

 - Ceramaseal Break

Over Pressure

- Hotter than expected Magnet

- Loss of Vacuum in Jacket

- Vacuum Jacket Volume Pressurization

- Quick charge of LN2 with warm cryostat

Thermal Shock

- Quick charge of LN2 with warm cryostat

Accident

- Fire

- Seismic

Failure of Bore Heater

There is no safety consequence – Only frosting of the experimental boundary

Joint Failure - The most common kind of magnet failure

- Excessive Motion

Joints are cantilevered, but hoop tension and Lorentz force compensated as in a coax. Bellows allow motion. The Joints will be insulated and wrapped with epoxy Glass

= Omission of a Force Component

George Mulholland helped with a reminder of the pressure force. This is about 1400 lbs, and is taken by tension in the joint. G-10 guides have been added to reduce the moment on the ceramaseal connections.

Insulation Failure

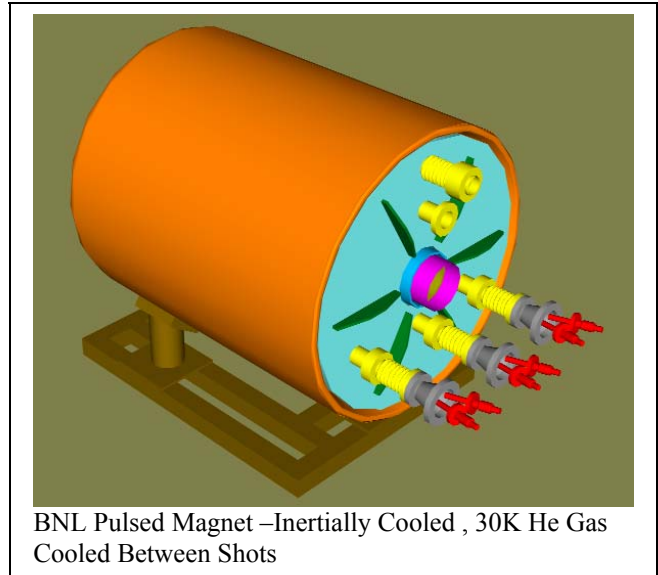
Conservative insulation design is employed. All insulation planes above one volt are insulated with Kapton. Regions where thermal contraction cracks are possible are insulated with Kapton to produce a reliable parting plane, and provide insulation against tracking behavior.

Specification Content:

Electrical Testing

The Seller shall assign a trained personnel and provide all necessary test equipment including digital multimeters for resistance measurement and DC hipot testers for ground insulation testing, during assembly process and at the completion of the prototype cryostat. Electrical testing of the electrical connection and component, including pulsed coil and bus connection, sensors, and diagnostic wiring, shall be performed after a component becomes inaccessible for service unless the enclosure is disassembled. The checkpoints and the type of electrical testing during assembly stage shall be defined by Seller in the fabrication plan and approved by the Purchaser's Representative.

Insulation test



No measurable electrical connection at mega-Ohm range shall be allowed between the ground and any diagnostic component / connection, between different sensors, and between sensor and the coil circuit.

DC Hipot Testing

The initial DC hipot testing shall be performed on the pulsed magnet coils and lead connections before they are installed. All subsequently measured leakage currents shall be compared with the initial value for verification.

The coils and lead connections shall be tested at 1 kV for 1 minute with the limiting current of a DC hipot tester set to 10 micro Amp. The allowable leakage current shall be no more than 5 micro Amp.

Caution:

- Do not perform hipot testing with the coil / lead in evacuated enclosure.
- Isolate the voltage tap wires when the coil / lead is tested with hipot tester.
- Do not use hipot tester on any sensor.

Leaks

Probably Frostbite is the most significant danger. Even a small leak would fill the test cell with cold He.

He/LN2 Cryostat Leak

The Cryostat and vacuum jacket are designed in accordance with ASMEVIII, including proof tests, (but not stamped?)

Mechanical Seal Failure

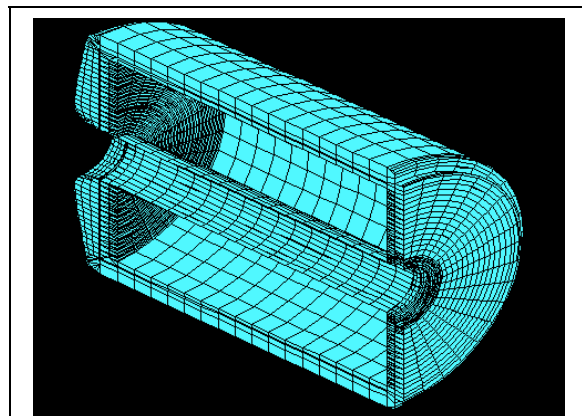
A redundant welded seal is available as a back-up. It can be applied if there are problems in-service? Or in the shop?

Bellows Crack

These need to be specified conservatively with respect to displacement, and pressure rating. Axial displacement of the bore bellows, and lateral and radial displacements of the bellows support feet are both around 3mm.

Ceramaseal Break

The ceramaseal joint penetration is loaded in compression and restrained by it's connection with the conductor break-out. It is intended to be either immersed in LN2 or traced with LN2 cooling tubes – This is probably an argument for tracing . Mechanical qualification is in Process (We have one and dunk tests, and load tests are needed)



Elements modeling Helium total $2 \cdot 167 \text{ m}^3$ Or $.335 \text{ m}^3$. This is 5 m^3 at 30 K and 49 m^3 of Helium at RT and 1 atm.

The dished head and annulus can be filled to reduce the He inventory. The dished head volume is .044

Leak Related Specification Content:

Sniffing Tests During Manufacture

The following shall be performed in the prescribed order:

- 1) Confirm the sensitivity of sniffer is better than 5×10^{-5} std atm-cc/second helium with a calibrated helium source.
- 2) Use an appropriate temporary cover to close the vessel / volume, and introduce helium gas into the test volume without cracking the temporary seal.
- 3) Confirm the background helium reading is below the sensitivity of the sniffer.
- 4) Spot leak checking shall be performed by inserting the sniffer in the envelop, which covers the outer surface of the joint area.
- 5) At high background helium count, consider isolating the first envelop from the background with a second envelope, which is flushed with nitrogen gas.
- 6) Remove leak checking attachments and clean up the surface. Flush out helium gas if necessary.

Over Pressure

Hotter than expected Magnet

Magnet insulation damage occurs above 100C. Our operating range is 30 to 100K

Loss of Vacuum in Jacket

Cryogenic Foam limits heat gain to less than 1000 watts. The magnet stored energy is of order $1e7$ Joules
The Cryogenic Foam planned for use on the cryostat is fire retardant, and will limit heat gain and resulting pressurization during an external fire.

For Liquid N₂ operation, we can vent liquid to the "shuttle tank" via pressure relief? The pressure relief valve be positioned at the bottom so we vent liquid and reduce the probability of overpressure due to relief valve pressure drop? . For He operation, since there is only a small volume of He gas - not liquid, loss of vacuum accident cannot be of much consequence in terms of relief valve performance. This is an issue with the LHe superconducting magnets not in resistive magnets. In our magnet, the loss of vacuum would produce a small heat gain with respect to the mass inventory of He. Much less time rate of expansion than pulsing the magnet.

Vacuum Jacket Volume Pressurization

A relief disk is provided. Pressure beyond one atmosphere loads the bellows and can damage them. The "Bumpers" that take the net lateral vacuum load do not work in tension if the vacuum jacket is pressurized.

Quick charge of LN₂
with warm cryostat

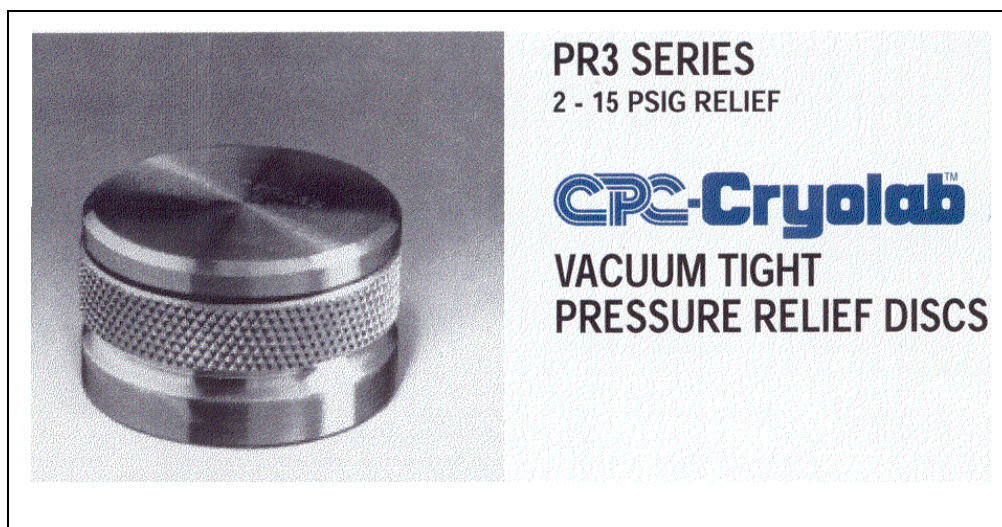
Thermal stress in the
magnet?

Bore Heater Strip Vendor
and Specifications (Willie
Burke?)
Accident

Fire

Foam has good fire
retardant properties.
CryoCoat™ UltraLight™
provides the robust
mechanical properties

associated with syntactic foam technology in a low-density insulation system (specific gravity from 0.08 to 0.11) with excellent thermal properties. CryoCoat™ UltraLight™ can be used as a mold-in-place insulation system on large, complex and uneven surfaces, or blown into closed molds to form near-net-shape components. CryoCoat™ UltraLight™ UL79 withstands liquid hydrogen temperatures and the **elevated temperatures of re-entry from space.**



Application of the foam is intended to occur at BNL. Application procedures and use of solvents etc. will have to meet BNL safety procedures.

Seismic

The magnet is supported on three legs. One is fixed and two are sliding. The sliding block feet will have to have some tensile capacity to ensure magnet stability against side loads.
Supports are robust otherwise – not as delicate as superconducting magnets supports.