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Engineering Analysis of the Reference Design for the CLAS Toroid

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1 Overview

The design of the CEBAF Large Acceptance Spectrometer (CLAS) is based on a large superconducting toroidal magnet with six coils. The magnetic field allows the determination of particle momenta to about one percent over a large acceptance.

CEBAF has generated a reference design for the CLAS magnet. [1] To demonstrate the feasibility of this design we have made choices which we believe are appropriate, but not unique. These include a cable-in-conduit-conductor (CICC). The reference design has been reviewed by an external Technical Advisory Panel [1] and by an independent consulting firm (Powers and Associates). While these reviews did not delve deeply into the design, they approved of the general approach. The purpose of this note is to furnish prospective bidders with some of the analytical work we have done in support of the reference design. While we believe this work to be appropriate and accurate, the vendor is cautioned to use this information at his own risk.

The choice of CICC conductor was made after considering several options. For example, in our judgement, an indirectly cooled or "intrinsically stable" design does not present ample energy margin for local thermal disturbances. We have also considered a pool boiling "Stekley stable" design. We concluded this design would not meet our space constraints.

Detailed calculations of the conductor, cooling system and structural analysis of the magnet are given in Appendices A, B and C. The sections that follow give the important conclusions of the analysis.

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[1] The magnet design is shown on CEBAF dwg 66100-D-00442 and associated drawings.

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2 Conductor

The coil of the reference design is shown in Figure 1, and a cross section of the coil and its bobbin plate is shown in Figure 2. Each of the six coils has 80 turns and operates at 10420 A. Figure 3 shows the operating load line for the conductor.

We have analyzed the response of the CICC conductor to a quench. Without a dump resistor, the coil hot spot would reach an unacceptable temperature of 420K. Hence reliable dump circuitry is required. With a 0.1Ω dump resistor the maximum temperature in the coil can be restricted to 60K. In these circumstances the maximum pressure within the CICC does not exceed 11 atmospheres.

The results of our quench analysis are:

- 97% of the stored energy will be deposited in the .1Ω dump resistor in the event of a quench.
- The coil will operate with a temperature margin of 1.4K.
- The energy margin per centimeter of coil length is .377 Joules.
- The coils are stable for all credible conductor motions and epoxy cracking scenarios.

3 Cooling System

The reference design uses a forced flow cooling scheme (see CEBAF dwg 66100-D-00459). The 2.8 atm supercritical helium flows through all six coils in series. This flow is recooled after a passage through each coil by the subcooler/re-cooler bath. Our analysis indicates that a recirculating flow of 15 gram/sec is satisfactory. A liquid helium pump is required to maintain this forced flow.

SURA will install a helium refrigerator to service the End Stations. This End Station Refrigerator (ESR) requires the return of boil-off helium. The return pressure at the End Station is 1.2 atm. This establishes a temperature of 4.5 K in our helium bath. Liquid nitrogen is also supplied by the ESR at a maximum pressure of 2.5 atm.

We have sized all vessels and piping that operate at a nominal pressure of 2.8 atm to withstand at least 130 psi. All other vessels are designed for 60 psi.

Our calculations show a theoretical heat load of 80W to the 4.4K structure. Including the heat load from the power leads, this corresponds to a 7.4 g/s flow of 4.42K liquid helium required from ESR. We have added a 50% contingency to this number to establish our requirements from the ESR.

4 Structural Analysis

We have used CEBAF's internal electromagnetic code SHOFORCE to calculate the magnetic forces on the coil structure. In general, these forces tend to force each coil into a circular shape. In addition each coil is attracted toward the other five coils. These centering forces are largest in the forward region. We have used analytical as well as finite element (FE) techniques to determine stresses and deflections. [2]

Under perfect symmetry the coil does not experience out-of-plane electromagnetic forces. However, the structure is designed to withstand a permanent 0.2° out-of-plane condition and an upset condition where one coil is off and the remaining five are at full current.

The electromagnetic forces used in the analysis are shown in Figures 1s, 6s and 7s of Appendix C. The results of our finite element analysis are shown in Figures 2s, 3s, 4s and 5s. The analysis shows the following peak stresses:

- Bobbin plate — 13,000 psi
- Forward support rings — 32,000 psi
- Aft support ring — 12,500 psi

References

[1] "A Large Acceptance Spectrometer for CEBAF," Report to the LAS Technical Advisory Committee, CEBAF, November 17-19, 1988.

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[2] CEBAF uses the FE code IDEA's.

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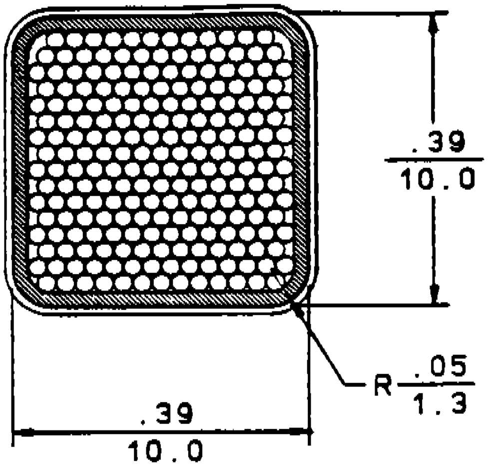

CONDUCTOR DATA

OPERATING CURRENT 10.4 kA
CRITICAL CURRENT DENSITY IN SC (at 3.5 T) 3060 A/mm
OPERATING CURRENT DENSITY IN SC (at 3.5 T) 1590 A/mm

WIRE (68% BY VOLUME)
-225 STRANDS .54 mm DIA
-2 (OF 3) STRANDS 100% CU
-1 (OF 3) STRANDS 63% CU, 37% NbTi
-7.5 : 1 OVERALL CU/SC RATIO

JACKET :
.64 mm WALL
304L STAINLESS STEEL

INSULATION :
.10 mm THICK (HALF-LAPPED, 2 WRAPS, .25 um KAPTON)

LENGTH OF ONE TURN 460 INCHES ~~(1.16 METERS)~~
TOTAL LENGTH REQ'D 73 600 INCHES ~~(185.6 METERS)~~

NOTE: DIMENSIONS ARE IN INCHES / MILLIMETERS.

Figure 1: Conductor for the Reference Design.

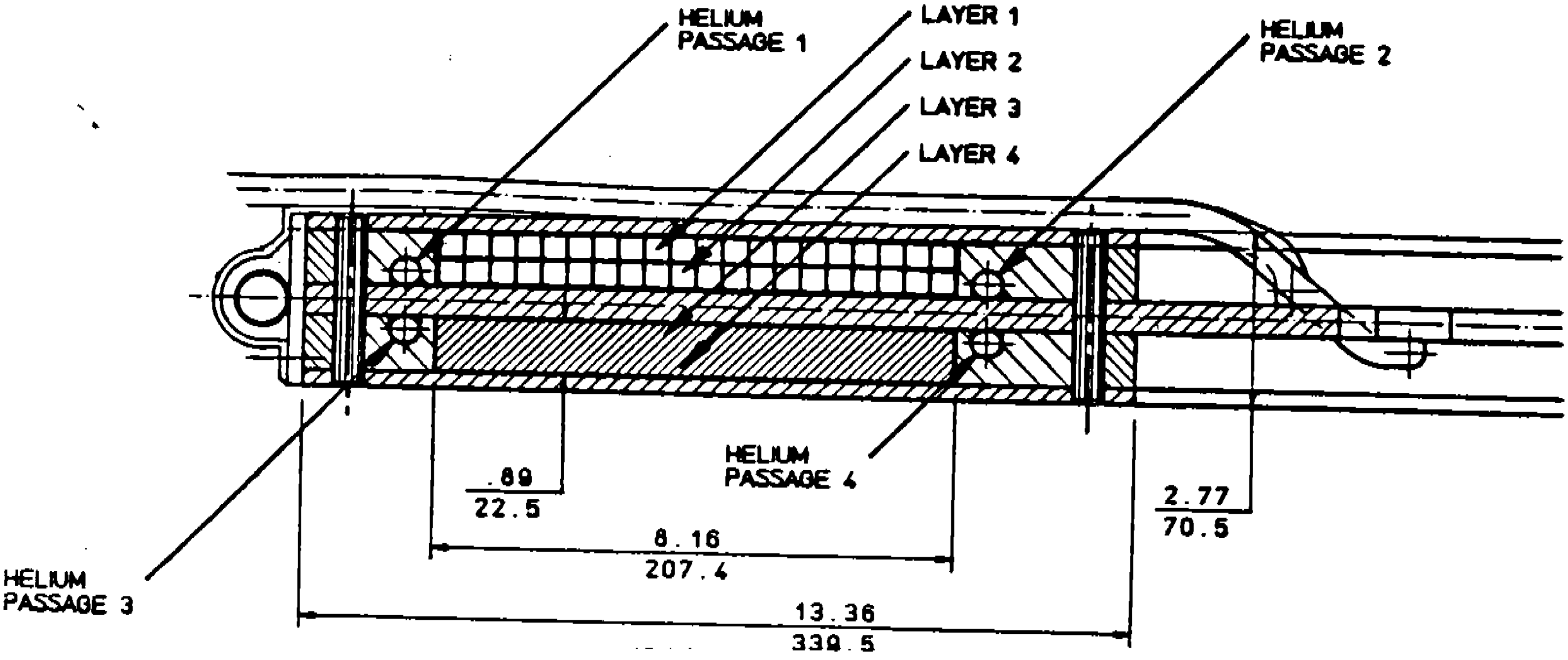

Figure 2: Cross section of the coil on its bobbin plate.

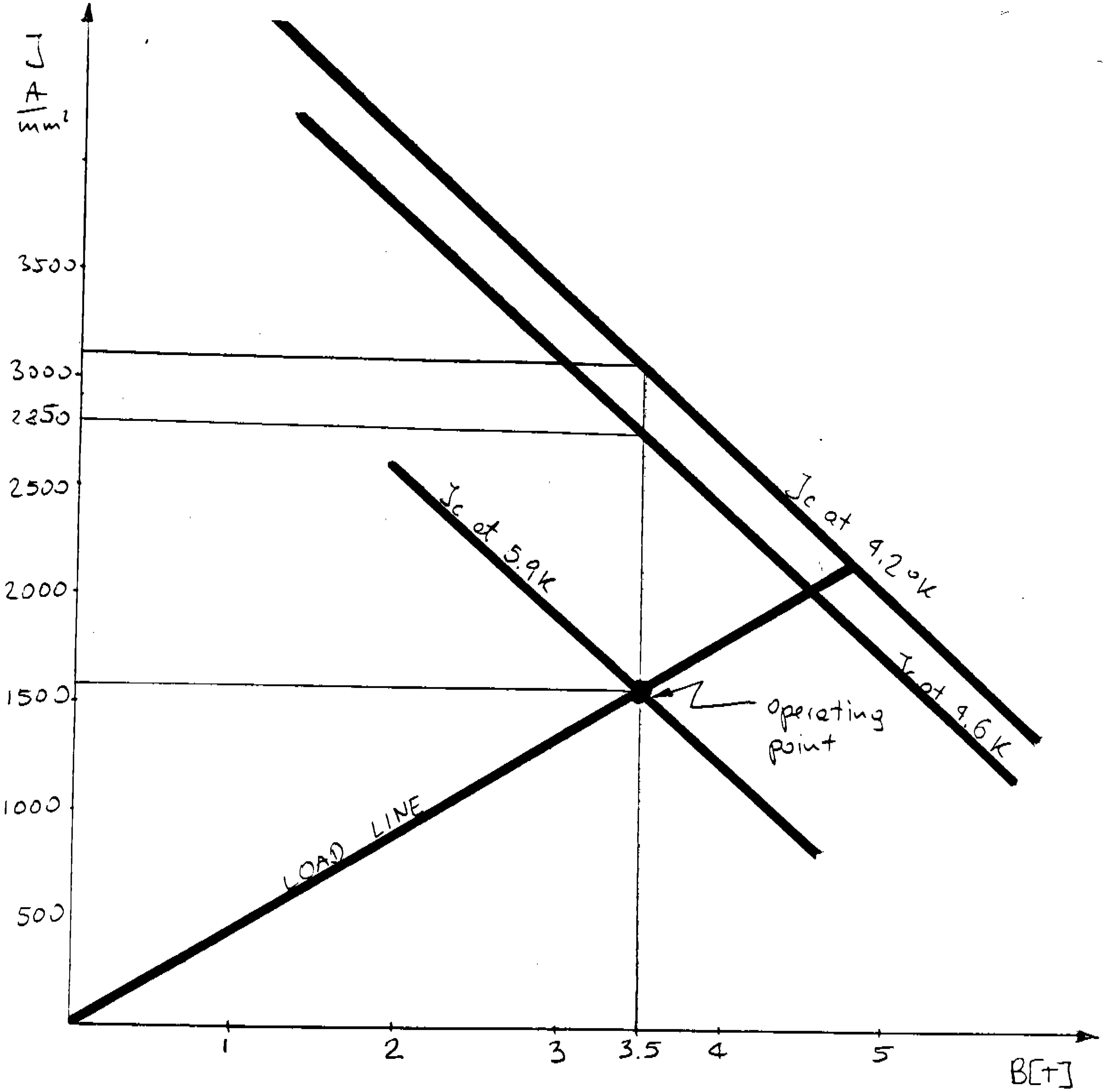

Figure 3: Operating load line for the conductor.

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APPENDIX A

CONDUCTOR DESIGN FOR LAS MAGNET

1 Cable-in-conduit design (see fig 2,1)

1.1 Operating condition:

- max operating temp. – 4.8°K
- cooling by trickle flow of super critical LHe at 2.8ata
- operating current – 10420A
- overall current density – $100\frac{A}{mm^2}$
- critical current density in SC – $3100\frac{A}{mm^2}$ at 3.5T
- $\frac{J_o}{J_c}$=.5 (see fig 3)
- insulation–kapton 2 layers half-lapped, .2mm total thickness per side
- conductor size 10.4 × 10.4mm
- hydraulic dia 3.6×10^{-4}m

The limiting current, providing stable operation against small disturbances (conductor motion, epoxy cracking etc) according to Lue and Miller (1) is I=14000 A.

I_{lim}, evaluated according to the formula with zero LHe flow:

$$J = 1.1\left[f_{cu}\left(1-f_{co}\right)/f_{co}\right]^{1/2}\left(T_c - T_b\right)^{1/2}\rho^{1/2}D^{-1}$$

where:

f_{cu}
f_{co} - metallic fraction of cable cross section, .68
f_{cu} - copper fraction of cable metallic cross section, .88
T_c - critical temperature, 7.8K
T_b - "bath" temperature, 4.5K
ρ - resistivity of copper @ 4.3K and 3.5T (assuming RRR 100), $3.1 \times 10^{-10}\Omega$m
D - hydraulic diameter, 2.6×10^{-4}m

The quench characteristics of the magnet has been analyzed using the code "QUENCH". This analysis was based upon a 0.1 ohm external dump resistor and quench detection voltage of 40mV. In order to account for the presence of helium, the quench velocity was corrected by use of equation 9.17 from reference 13. This correction reduces the quench by a factor of 0.3 compared to the "dry" coil. Based upon this, we predict a maximum hot spot temperature of approximately 60K (5).

In the event the dump circuit failed, our analysis shows that the hot spot temperature would reach 420K. In our judgement this is unacceptable and, therefore, we would require reliable quench detection and dump circuits.

External dump circuit should remove about 99.7% of the total stored energy (18MJ). The max pressure rise within the coil can be estimated according to Wechi (2)

$\Delta p = .95\rho\ c\ \left(\frac{4f}{D}\right)^{3/2}\left(\frac{D}{4f}\right) t^{1.1} \approx 11atm$

where:

ρ=129 $\frac{kg}{m^3}$ density of He at 4.5K and 2.8atm

c=120.4 $\frac{m}{s}$ velocity of sound

D-2.6×10^{-4}m hydraulic diameter

f=.005 friction factor (assumed)

t=7 sec time to discharge the magnet if .1 or resistor is used

1.2 Conductor stability analysis for CICC conductor

1.2.1 ΔT available within the highest field region:

- critical temp. at zero current (ref. 12)

$$T_c(3.5T, 4.2K) = T_c(OT)\left(1 - \frac{B}{B_{c2}(O)}\right)^{.59}$$

where:

$T_c(OT) = 9.2°K$ critical temp. at zero field

$B_{c2}(O) = 14.5T$ critical field at zero current

$B = 3.5T$ max field

$$T_c(3.5T, 4.2K) = 9.2\left(1 - \frac{3.5}{14.5}\right)^{.59} = 7.8°K$$

- current sharing temp.

$$T_{sc} = T_c - (T_c - T_o)\frac{I_o}{I_c}$$

$\frac{I_o}{I_c} = .52$ of operating to critical current ratio

$$T_{cs} = 7.8 - (7.8 - 4.2).52 = 5.9°K$$

and

$$\Delta T = T_{cs} - T_{op}$$

T_{op}=4.5 K operating temperature at high field

$$\Delta T = 5.9 - 4.5 = \underline{1.4K} \text{ available temperature margin}$$

1.2.2 Energy margin available per 1cm of conductor

a) Energy margin available from He for ΔT=1.4K

- LHe volume

  V=.24cm^3 (calculated from fig 2)

  $Q_1 = \sum V\rho\Delta h = .24\mathrm{cm}^3\,(+.1154 \cdot 2.91|_{h\rightarrow 4.5-5}$
$+\ .1112 \cdot 1.87|_{h\rightarrow 5-5.2} + .0995 \cdot 2.95|_{h=5.2-5.4} + .05715 \cdot 7.15|_{h\rightarrow 5.4-5.6}$
$+.04878 \cdot 5.92|_{h\rightarrow 5.9-5.6})=$ .3682 J/per 1cm of conductor length

b) Energy margin available from SS for ΔT=1.4K

$Q_2 = mc_p\Delta T$

C_p(at 5° K)=$2.8\frac{J}{k_gK}$

m=Vρ=.256cm^3: .008 $\frac{kg}{cm^3}$ =2.13$\cdot 10^{-3}$ kg

$Q_2 = 2.13\cdot 10^{-3} 2.8 \cdot 1.4 = 8.3 \cdot 10^{-3}$ J

c) Energy margin available from Cu for ΔT=1.4K

C_p(at 5K) = .15 $\frac{J}{kgK}$

m= 3.8$\cdot 10^{-3}$ kg

$Q_3 = .15 \cdot 3.8 \cdot 10^{-3} \cdot 1.4 = 7.9 \cdot 10^{-4}$J

d) Energy margin available from NbTi (4)

C_p(at 5 K)=.4$\frac{mJ}{kg}$

m=5.3 $\cdot 10^{-4}$ kg

$Q_4 = .4 \cdot 5.4 \cdot 10^{-4} \cdot 1.4 = 3.5 \cdot 10^{-4}$J

Total energy margin available

$$Q = Q_1 + Q_2 + Q_3 + Q_4 = .377 \text{ J}$$

1.2.3 Possible disturbences

a) Epoxy cracking

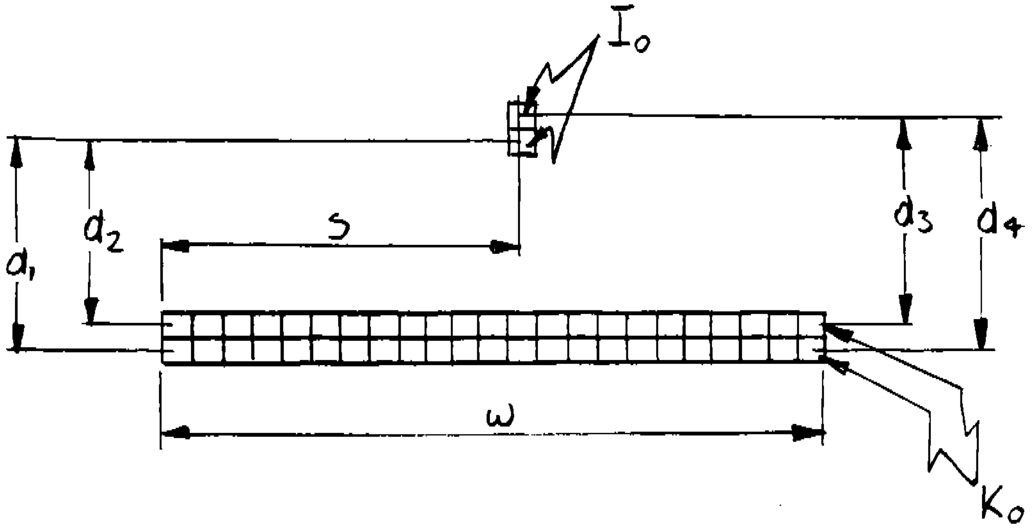
The energy released by cracking epoxy due to excessive stress has been measured to be less than <u>18k J/m^3</u> (3, 9, 10, 11). Assuming thickness the conductor insulation and the ground insulation to be about .2 and .8mm respectively at the bobbin plate, the released energy can be .0162 $\frac{kJ}{m^3}$ (assuming that epoxy release energy uniformly) or $1.6 \cdot 10^{-3}$J per cm of conductor

b) Debonding energy

Debonding energy has been reported (3) to be 3 $\frac{KJ}{m^3}$ which gives $Q_v = 3 \cdot 10^{-4}$ J per 1cm of conductor

c) Stick-slip conductor motion

Force between the conductor and the current sheet



$K_o = 9.9 \cdot 10^5 \frac{A}{m^2}$ current density per conductor layer

I=10420 A current within the conductor

w=.21 m width of the current sheet

S-distance between projection of conductor on the current sheet a it's outer edge

d_1= .023m distance between conductor and the layer attached to the central plate on the other side

d_2=.033m distance between the conductor and the outermost layer on the other side

d_3=.033m distance between the outermost conductor with the innermost layer

d_4=.043 distance between the outermost conductor with the outermost layer

Force per unit length of conductor (11):

$$\frac{F}{l} = \frac{I_o \mu_o k_o}{2\pi} \left(tan^{-1} \frac{w-s}{d} + tan^{-1} \frac{s}{d} \right)$$

and force per the conductor at the edge (s=0) per unit length is.

for d_1

$$F_1 = \frac{10420 \cdot 4\pi \cdot 10^{-7} \cdot 9.9 \cdot 10^5}{2\pi} \left(tan^{-1} \frac{.21}{.023} \right)$$

$$F_1 = 3.0 \cdot 10^3 \frac{N}{m}$$

for $d_2 = d_3$

$$F_3 = F_2 = 2.9 \cdot 10^3 \frac{N}{m}$$

for d_4

$$F_4 = 2.8 \cdot 10^3 \frac{N}{m}$$

Total force $F_T = \sum_1^4 F = 1.16 \cdot 10^4 \frac{N}{m} = 116 \frac{N}{cm}$

For a coefficient of Friction μ=.4 (7) and in-plane loading per coil block F=$7.5 \cdot 10^3 \frac{N}{m}$ the frictional force is not large enough to suppress coil motion.

The work done by frictional force per cm of coil length:

$$W = T\Delta = F_T\mu\Delta = (116N)(.4)\Delta Q$$

where Δ is the displacement in meters, therefore the motion required to produce .37 Joules per centimeter of length is:

$$\Delta = \frac{.37}{116 \cdot .4} = .008m$$

Conclusion:

We should not encounter heat impulses that could drive the coil normal. In order to quench, the conductor has to move .8cm. This is not possible with the present design. Other motions are impossible because at 4K conductor is held tightly within the Aluminum enclosure. The energy released by the epoxy cracking can be easily absorbed by the coil.

REFERENCES:

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5. Wilson Quench program
6. National Bureau of Commerce, NBS Fermi P.O. NO. 9860
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11. Joseph A. Edminister "Electromagnetic" McGraw-Hill, Inc 1979
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APPENDIX B

COOLING REQUIREMENTS FOR CLAS

1) HEAT LOAD TO THE 4.5° K STRUCTURE

1.1 Radiation

a) coil only (from LN_2 shield to 4.5K surface)

$$\frac{Q}{A} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{1}{\xi_1} + \frac{1}{\xi_2} - 1}$$

T_1=100K assumed LN_2 shield temperature

T_2=4.5K assumed cold mass temperature

ξ_1=.06 Al with .25μ oxide (1)

ξ_1=.03 Cu with commercial polish (1)

$\sigma = 56.59 \cdot 10^{-9} \frac{W}{m^2 k^4}$ Boltzman constant

$$\frac{Q_1}{A} = \frac{56.69 \cdot 10^{-9}\left(100^4 - 4.5^4\right)}{\frac{1}{.06} + \frac{1}{.03} - 1} = .116\frac{W}{m^2}$$

Surface area of the one coil ("CLAS COIL" drwg 66100-E-00434):

- both sides $A_s = 11760 in^2 \times 2 = 23520 in^2 = 15.2 m^2$
- edge $A_e = 458 \cdot 2.1 = 962 in^2 = .62 m^2$

$$\text{total surface area} \quad A_T = As + Ae = 15.8m^2$$

$$\text{radiation heat load for one coil:} \quad Q_R = 15.8 \cdot .116 = 1.84W$$

b) Stainless steel support rings

$$\xi_1 = .048 \quad \text{ss (1)}$$

for:

$$\xi_2 = .03 \quad \text{cu (1)}$$

and surface area of the large ring ("CLAS cold rings drwg 66100-E-00446):

$$A_{SR} = 2.36m^2$$

surface area of the small ring:

$$A_{SR} = 3.66m^2$$

$$\text{total surface area} \quad A_T = A_{LR} + A_{SR} = 2.43 + 3.66 = 6.1m^2$$

$$\frac{Q_2}{A} = .107\frac{W}{m^2}$$

$$Q_2 = .65W$$

c) Subcooler recooler and feed can (dwg 66100-E-00451 and 66100-E-00450)

A= 7.4 m^2
Q$_3$ = .107 · 7.4 =8W

Total radiation heat load per magnet Q$_R$ = 6 · 1.84 + .65 + .80=<u>12.5W</u>

1.2 Conduction

- <u>out of plane support</u>-10 per coil ("CLAS 4K to 300K support ass'y" dwg 66100-B-00464)

  cross section A=.6 in^2

  cold length (4→ 100° K) L=1.0in

  $\int_4^{100}$ kdt=.51$\frac{W}{in}$ for G-10

  Q$_1$ = 6 · 10 · $\frac{1.37}{1.0}$ · .51=19W for 6 coils & 10 supports per coil

- <u>in plane support;</u> 3 per coil ("CLAS AFT in plane radial suspension ass'y" dwg 66100-E-00472 and "CLAS forward in plane radial suspension ass'y" dwg 66100-E-00473)

  cross section $A_1 = .32in^2(2), A_2 = .11in^2(1)$

  cold length(4→ 100°K)L=12in

  $\int_4^{100}$kdt=13.4$\frac{W}{in}$ for SS304
  Q$_2$ = 6$\frac{.32\cdot2+.11}{12}$· 13.4=5.2W for 6 coil and 3 supports per coil

- <u>LN$_2$ support standoffs;</u> supporting shield against cold structure ("CLAS LN$_2$ shield" drwg 66100-E-00455)
  Q (central block)= $\frac{\frac{\pi}{4}(1.04^2-.36^2)}{.63} \cdot \int_4^{77} kdT = 2.37 \int_4^{77}$ kdT
  Q (within center of the bobbin) =40$\frac{\frac{\pi}{4}\cdot.32^2}{.63} \int_4^{77} kdT = 5.10 \int_4^{77}$ kdT
  Q (along perimeter) = 36$\frac{\frac{\pi}{4}.25^2}{.34} \int_4^{77} kdT = 5.19 \int_4^{77}$ kdT
  $\int_4^{77} kdT = .508\frac{W}{in}$ for G-10
  $Q_T = 6(2.37 + 5.10 + 5.19)\cdot$ .508= 38.6W per magnet

- "Feed can" cold to warm support assume 4W

1.3 Total heat load to the 4.3K structure

Q$_T$=12.5+19+5.2+39.6+4≈80W

<u>Assumed Q$_T$=100W</u>

APPENDIX B

2) COOLING REQUIREMENT AND PRESSURE DROP ACROSS THE COIL

CLAS torus is kept cold with supercritical 2.8atm He subcooled to 4.5K by a heat exchanger (called the subcooler-recooler) before is sent to the coil. The Helium temperature rises as it travels around the coil. We have limited the maximum temperature to 4.8K. The required flow rate can be evaluated according to formula:

$$\overset{o}{m}= \frac{Q}{6\Delta H_{4.5\div 4.8^{\circ}K}} = \frac{100W}{6\cdot 1.74\frac{J}{g}} = 9.5g/sec$$

Assume 15g/s=.033 $\frac{lb}{s}$

2.1 Max ΔT across the coil block ("CLAS single coil" drwg 66100-E-00460)

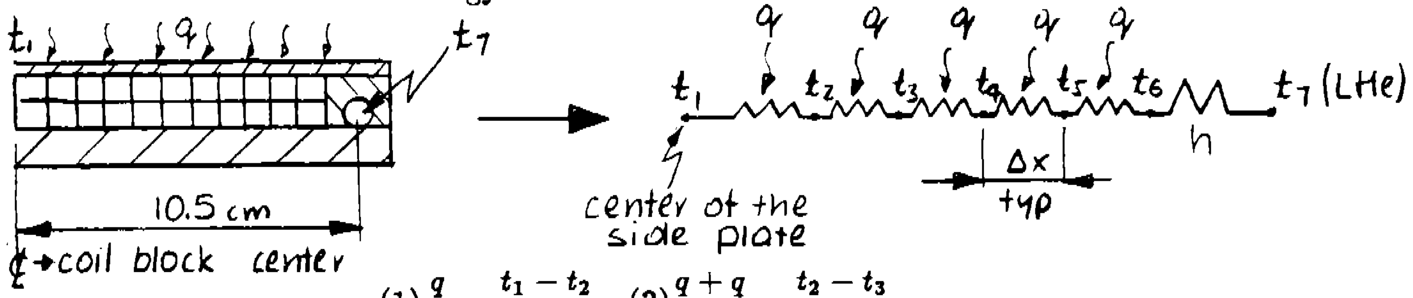
Assumption:

-heat is not transferred across the coil and from side plate to the coil.

-due to the symmetry, one-half of the side plate is considered

-heat flow only in one direction

Use Electrical Circuit Analogy:



$$(1)\frac{q}{A} = \frac{t_1 - t_2}{\frac{\Delta x}{K}} \quad (2)\frac{q+q}{A} = \frac{t_2 - t_3}{\frac{\Delta x}{K}}$$

$$(3)\frac{q+q+q}{A} = \frac{t_3 - t_4}{\frac{\Delta x}{K}} \quad (4)\frac{q+q+q+q}{A} = \frac{t_4 - t_5}{\frac{\Delta x}{K}}$$

$$(5)\frac{q+q+q+q+q}{A} = \frac{t_5 - t_6}{\frac{\Delta x}{K}}$$

$$(6)5q = A_1 h(t_6 - t_7) \rightarrow \frac{5q}{A_1 h} = t_6 - t_7$$

$$(1)+(2)+(5) \rightarrow \frac{15q}{A} = \frac{t_1 - t_6}{\frac{\Delta x}{k}} \rightarrow \frac{15q\Delta x}{KA} = t_1 - t_6 ...(7)$$

$$(6)+(7) \rightarrow \frac{5q}{A_1 h} + \frac{15q\Delta x}{KA} = t_1 - t_7 = \Delta T$$

Δx=2.1cm-subsection lenght

A=.32cm^2-cross section of unit length of side plate

A$_1$=πD=π1.1=3.43cm^2 heat transfer area per cm length of cooling tube

q= $\frac{Q}{A_o}$Q-radiative heat load per 2.1 × 1 cm surface

$\frac{Q}{A_o}$=.116$\frac{W}{m^2}$=1.16$\cdot$10^{-5}$\frac{W}{cm^2}$ radiation heat load per unit area from sec 1.1

q=1.16 $\cdot$ 10^{-5} $\cdot$ 2.1 = 2.4 $\cdot$ 10^{-5}W/per subsection length

k=.4$\frac{W}{cm^o K}$-thermal conductivity of Al

h-heat transfer coefficient evaluated as follow:

h=$\frac{N_u k_{he}}{D}$

N_{nu}=.023(Re)$^{.8}$(Pr)$^{.4}$ -Nusselt number

Pr=.747 Prandtl # for He at 2.8atm and 4.5K

k$_{he}$=.213$\frac{mW}{cm^o K}$=5.4$\cdot$10^{-4}$\frac{W}{inK}$ thermal conductivity of He at 2.8 atm and 4.5K

Re=$\frac{D\vartheta\rho}{\mu e}$ Reynold # of flowing fluid

$v = \frac{\overset{o}{V}}{A_{tube}} = \frac{\frac{\overset{o}{m}}{4\rho}}{\frac{\pi D^2}{4}} = \frac{\overset{o}{m}}{\pi\rho D^2}$ flow velocity per cooling tube (4 tube used)

$\overset{o}{m}$=15g/s=.033$\frac{lb}{s}$ mass flow rate

ρ=.1291$\frac{g}{cm^3}$=8.04$\frac{lb}{ft^3}$ average fluid density

D=.43in =.036ft tube inner dia

μ=2.4$\cdot$10^{-6}lb/fts absolute fluid viscosity

Re=$\frac{\overset{o}{m}}{\pi D\mu e} = \frac{.033}{\pi\cdot .036\cdot 2.4\cdot 10^{-6}} = 1.2\cdot 10^5$

$N_{NU} = .023\left(1.2\cdot 10^5\right)^{.8}(.747)^{.4}$=237

h=$\frac{237\cdot 5.4\cdot 10^{-4}}{.43} = .30\frac{W}{in^2 K} = .046\frac{W}{cm^2 K}$

and temperature gradient across the side plate is:

$\Delta T = \frac{5\cdot 2.4\cdot 10^{-5}}{3.43\cdot .046} + \frac{15\cdot 2.4\cdot 10^{-5}\cdot 2.1}{.4\cdot .32} = 7.5\cdot 10^{-4} + 5.9\cdot 10^{-3} = \underline{6.6\cdot 10^{-3}K}$

Conclusion: Expected temp gradient across the side plate is negligible

2.2 Pressure drop across one cooling tube (coil cooling passages)

$$\Delta P = .001294\frac{fL\rho v^2}{D}\text{ “Crane” catalog (2)}$$

where:

$f = .184 \cdot Re^{-.2} = .0154 -$ for Re>50000 - friction factor at Re=1.2·10^5 (4)

$L = 12m = 472in \approx 39.4ft -$ tube length per single coil

$v = \frac{\overset{o}{m}}{\pi \rho D^2} = \frac{.033}{\pi \cdot 8.04 \cdot .036^2} = 1.0\frac{ft}{s}$ fluid velocity per cooling passage

$\rho = 8.04\frac{lb}{ft^3}$average fluid density at 4.5K and 2.8atm

$D = .43in$ cooling tube inner diameter

$$\Delta P = .001294\frac{.0154 \cdot 39.4 \cdot 8.04^{.2}}{.43} = .017psi/coil$$

$\underline{\Delta P_T = 6 \cdot \Delta P = .10psi}$ per magnet excluding subcooler-recooler

APPENDIX B

3) SUBCOOLER-RECOOLER DESIGN

LIQUID REQUIREMENT FROM END STATION REFRIGERATOR

He flow within the CLAS is driven by a LHe pump. A Welter-Meisner-Institute pump (3) is used with characteristic presented in publication DF-02 CEC 89. For flow schematic see "Flow schematic diagram CLAS torus superconductive magnet" drwg 66100-D-00459

3.1 Temperature of the fluid leaving the pump assuming He in temp. 4.5K.

From fig. 4 pub. DF-02 CEC

- total efficiency-η_T=.2 for flow 15g/s=420 l/h –assumed flow rate (sec 2)
- hydraulic efficiency η_h=.6

Isotropic efficiency is $\eta_T = \eta_h \cdot \eta_s \rightarrow \eta_s = \frac{\eta_T}{\eta_h}$

and $\eta_s = \frac{.2}{.6}$=.34

defined as: $\eta_s = \frac{he-hf}{he'-hf} \rightarrow he = \frac{1}{\eta_s}(he-hf)$+hf

h_f -enthalpy of the fluid at inlet h_f(4.5K, 2.7ata)=11.335J/g and s_f=3.514 $\frac{J}{gK}$

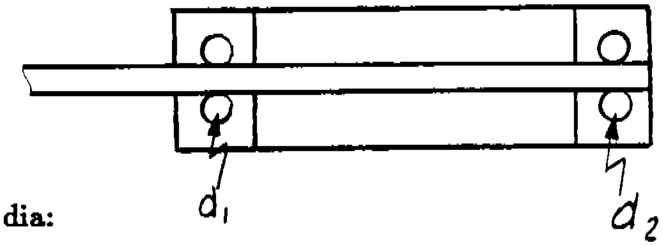
h_e -enthalpy of the fluid at the outlet if compression was ideally isentropic h_e(4.52K, 2.8ata)=11.476J/g and s=3.514 $\frac{J}{gK}$

h'_e -actual enthalpy of the fluid leaving the pump.

$$h'_e = \frac{1}{.34}(11.476 - 11.335) + 11.335 = 11.75J/g$$

what is equivalent of $T_{out} \approx$ 4.6°K

3.2 Heat transfer coefficient within the recooler-subcooler system



for tube dia:

$$D = \left[2\left(d_1^2 + d_2^2\right)\right]^{1/2}$$ equivalent subcooler tube size

where

d_1 = .41in coil inner cooling tube

d_2 = .44in coil outer cooling tube

D=.85in; assume OD=1″, ID=.875in (standard tube size)

Heat transfer coefficient is evaluated as follows:

$$\overset{o}{V} = \frac{\overset{o}{m}}{\rho} = \frac{15\left[\frac{g}{S}\right]}{3655\left[\frac{g}{ft^3}\right]} = 4.1 \cdot 10^{-3}\frac{ft^3}{s} \text{volumetric flow rate}$$

$$v = \frac{\frac{4.2 \cdot 10^{-3}}{\pi\left(\frac{.875}{12}\right)^2}}{4} = 1\frac{ft}{s} \quad \text{flow velocity}$$

$$Re = \frac{D \cdot v\rho}{\mu e} \quad \text{Reynold number}$$

where:

$$D = .0729ft \quad \text{inner tube dia}$$

$$\rho = 7.95\frac{lb}{ft^3} \quad \text{fluid density}$$

$$\mu_e = 2.4 \cdot 10^{-6}\frac{lb}{fts} \quad \text{absolute fluid viscosity}$$

$$Re = \frac{.0729 \cdot 1 \cdot 7.95}{2.4 \cdot 10^{-6}} = 2.4 \cdot 10^5$$

$$N_{nu} = .023 \cdot Re^{.8}Pr^{.4} - \quad \text{Nusselt number}$$

$$P_r = .747 \quad \text{for He at 2.8ata, 4.5K}$$

$$N_{nu} = .023(2.4 \cdot 10^5)^{.8}(.747)^{.4} = 405$$

$$h = \frac{N_{nu} \cdot k}{D}$$

$$k = .213\frac{mW}{cm\ K} \quad \text{thermal conductivity of He at 2.8ata 4.5K}$$

$$D = .875" = 2.22cm \quad \text{tube diameter}$$

$$h = \frac{405 \cdot .213 \cdot 10^{-3}}{2.22} = .039\frac{W}{cm^2K}$$

3.3 Pressure drop per linear foot of subcooler-recooler:

$$\frac{\Delta P}{l} = .001299\frac{f\rho v^2}{d} \quad \text{pressure drop per 1ft ("Crane" cat ref 2)}$$

$$f = .184Re^{-.2} \quad \text{for} R_e > 50000$$

for data from sec. 3.2

$$\frac{\Delta P}{l} = .001299\frac{1.54 \cdot 10^{-2} \cdot 7.95 \cdot 1^2}{.875} = 1.8 \cdot 10^{-4}\frac{psi}{ft}$$

3.4 Required tube length for subcooler to cool fluid from 4.6→ 4.5K. ("Subcooler-recooler ass'y" drwg 66100-E-00451)

Property of the fluid entering subcooler:
- flow rate 15g/s (sec 2)
- pressure p=2.8ata
- temp. T=4.6°K (sec 3.1)

Heat balance between flowing fluid and the He both (assumed to be at 4.42K and 1.2atm)

$$Q_i = h_i A_i \Delta T_i - \quad \text{heat transferred from fluid to the boundry}$$
$$Q_o = h_o A_o \Delta T_o - \quad \text{heat transferred from the boundry to liquid.}$$
$$h_i = .039 \frac{W}{cm^2 K} - \quad \text{heat transfer coefficient within the tube}$$
$$A_i = \pi D_i L = 6.97 cm^2 - \quad \text{inner heat transfer area/per 1cm tube length}$$
$$A_o = \pi D_o L = 7.97 cm^2 - \quad \text{outer heat transfer area/per 1cm tube length}$$

Δ T- temp. gradient across the interface (ignore wall resistance)

Assume interface temp. 4.46K, temperature of boiling LHe within the dewar T=4.42K

For $\Delta T = .04$°K within the bath, $H_o = h_o \Delta T = .0035 \frac{W}{cm^2}$(read from the nucleate boiling heat transfer table for p=1atm)

Assumed average flowing fluid temp $\frac{4.6+4.5}{2} = 4.55$°K

and heat transfer rate:

$$Q_i = 6.97 \cdot .039\,(4.55 - 4.46) = .025 \frac{W}{cm}$$
$$Q_o = 7.97 \cdot .0035 \approx .025 \frac{W}{cm}$$

Total heat to be removed (for subcooler):

$$Q_T = \overset{o}{m}\,(h(2.8ata, 4.6^\circ K) - h(2.8ata, 4.5^\circ K) =$$
$$= 15(11.94 - 11.36) = 8.7W$$

Required tube length for subcooler:

$$L_s = \frac{Q_T}{Q_i} = \frac{8.7[W]}{.025\left[\frac{W}{cm}\right]} = 340cm = 3.4m$$

3.5 Required tube length for recooler.

In order to achieve higher degree of accuracy divide cooling region in the three stages.

G.P in plane loading

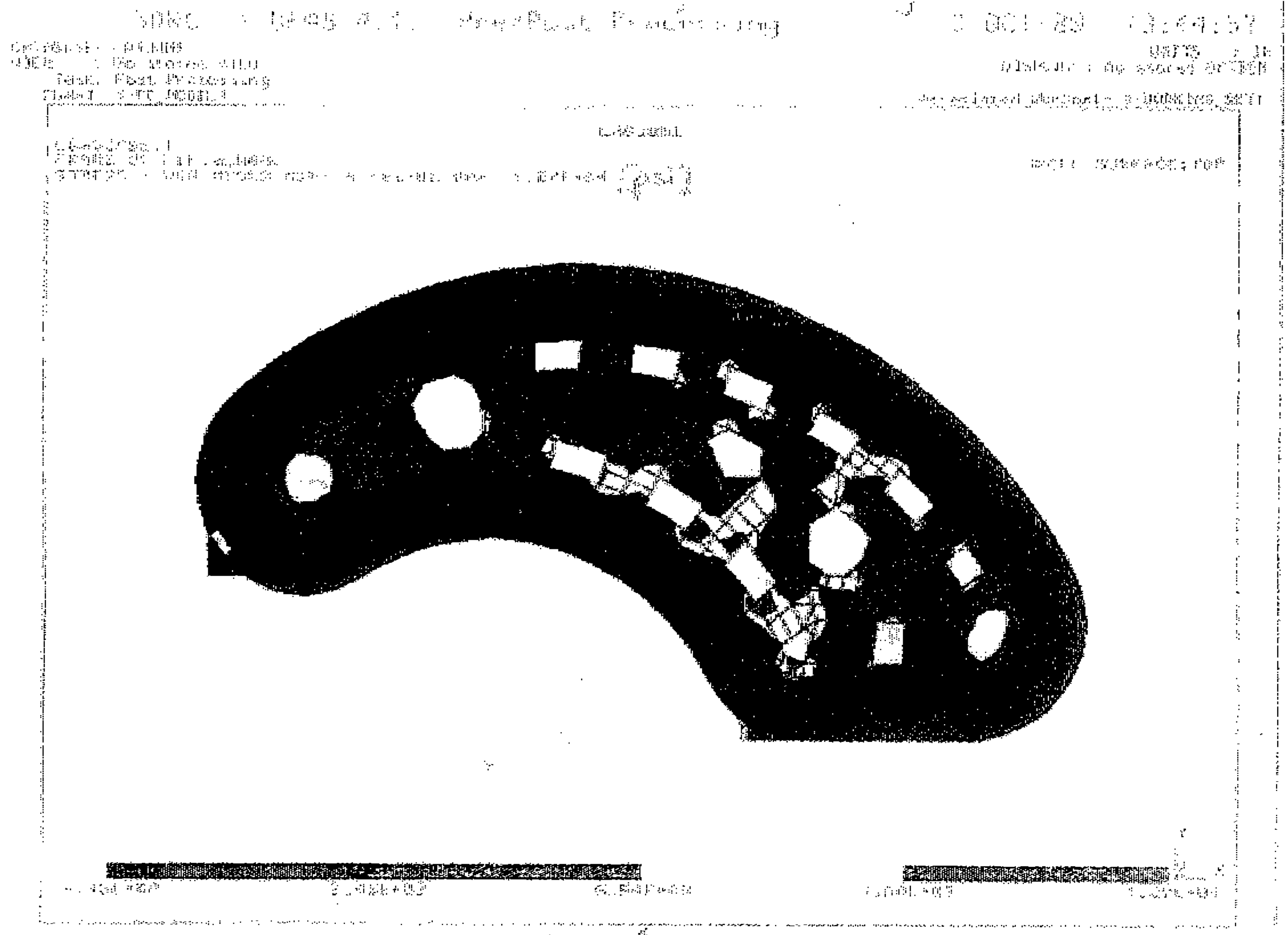


Fig 23

a) <u>cooldown the fluid 4.8→ 4.7°K</u>

$$T_{ave} = \frac{4.8 + 4.7}{2} = 4.75K$$

assumed interface temp 4.49°K:

$$h_o \Delta T (for \Delta T = 4.49 - 4.42 = .07^\circ K) = .0085 \frac{W}{cm^2}$$

$$Q_i = .039 \cdot 6.97\,(4.75 - 4.49) = .071 \frac{W}{cm}$$

$$Q_o = .0085 \cdot 7.977 = .068 \frac{W}{cm}$$

Energy removed:

$$Q_{4.8 \to 4.7} = \overset{o}{m}\ (\ h\,(2.8ata, 4.8^\circ K) - h\big(2.8ata, 4.7K\big) =$$

$$= 15 \cdot .582 = 8.73W$$

Required tube length: $L_{4.8 \to 4.7K} = \dfrac{8.73}{.008} = 128cm$

b) <u>cooldown the fluid 4.7→ 4.6°K</u>

$$T_{ave} = 4.65K$$

assumed interface temp 4.48°K; for $\Delta T = .06^\circ K \to h_o \Delta T = .0063 \frac{W}{cm^2}$

$$Q_i = .039 \cdot 6.97\,(4.65 - 4.48) = .046 \frac{W}{cm}$$

$$Q_o = .0063 \cdot 7.97 = .050 \frac{W}{cm}$$

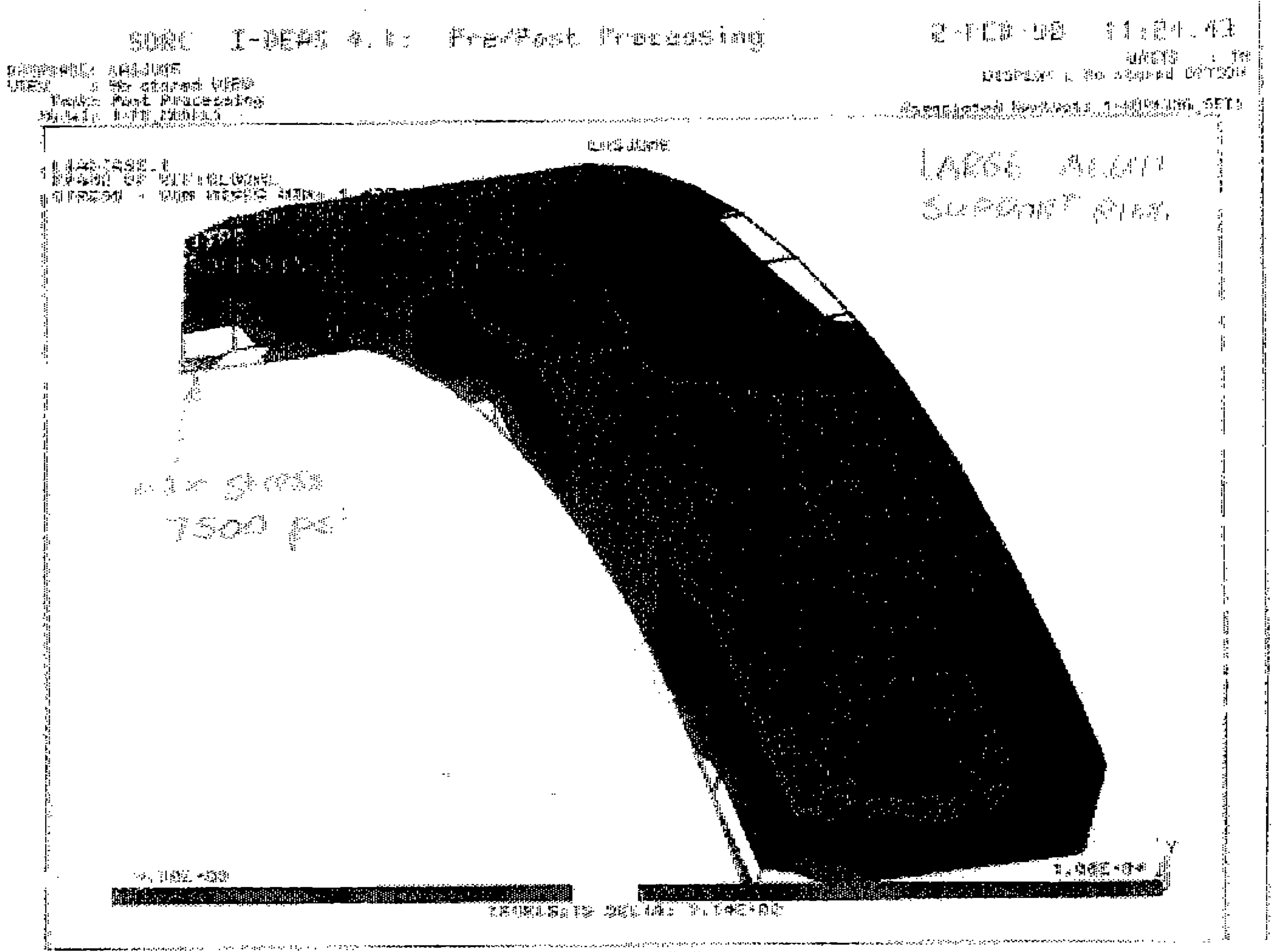
Energy removed:

$$Q_{4.7 \to 4.6k} = \overset{o}{m}\ \Delta h_{4.7 \to 4.6} = .15 \cdot .582 = 8.73W$$

Required tube length: $L_{4.5 \to 4.4^\circ K} = \dfrac{8.73}{.046} = 180cm$

c) <u>cooldown the fluid 4.6→ 4.5° K</u>

$$T_{ave} = 4.55^\circ K$$



SDRC I-DEAS 4.1: Pre/Post Processing

2-FEB-98 11:24:43

Task: Post Processing

LARGE ALUM SUPPORT RING.

max stress 7500 psi

fig 3

assumed interface temp 4.46K,

$$(for\ \Delta T = .04°K)h_o\Delta T = .0035\frac{W}{cm^2}$$

$$Q_i = .039 \cdot 6.97(4.55 - 4.46) = .025\frac{W}{cm}$$

$$Q_o = .0035 \cdot 7.97 = .026\frac{W}{cm}$$

Energy removed:

$$Q_{4.6-4.5k} = \overset{o}{m}\ \Delta h_{4.6-4.5k} = 8.73W$$

$$L_{4.6-4.5k} = \frac{8.73}{.025} = 340\text{cm}$$

Total tube lenght per one recooler:

$$L_R = L_{4.8\rightarrow 4.7} + L_{4.7\rightarrow 4.6} + L_{4.6\rightarrow 4.5} =$$
$$= 128 + 180 + 349 = 657cm$$

Assume 6 m per one recooler since boiling heat transfer rate of LHe at p=1.2atm is higher of about 20% than for p=1atm

Total tube length per assy:

$$L_T = L_S + 6L_R = 3.4 + 6 \cdot 6. \approx 39.4m = \underline{128ft}$$

3.6 Pressure drop across the cooling system.

3.6.1 Pressure drop per one coil.

ΔP_c= .017 psi/12m. (from section 2.2)

3.6.2 Pressure drop across the recooler-subcooler.

Δ P(per ft)=$1.8 \cdot 10^{-4}\frac{psi}{ft}$ (from sect 3.3)

$$\Delta P_r = 128[ft] \cdot 1.8 \cdot 10^{-4}\left[\frac{psi}{ft}\right] = .023psi$$

3.6.3 Estimation of pressure drop within the joints between recooler-coil and subcooler tube bends.

3.6.3.1 Pressure drop due to sudden enlargement and contraction (recooler tube-coil cooling tubes)

- sudden contraction:

$$K_{2c} = \frac{.5(1-\beta^2)\sqrt{sin\frac{\theta}{2}}}{\beta^4} \quad \text{“Crane cat” p.A-26 (2)}$$

-sudden enlargment:

$$K_{2e} = \frac{1-\beta^2}{\beta^4}$$

where

$$\begin{aligned}
\beta &= \frac{d_1}{d_2} \\
d_1 &= .43in \quad \text{coil cooling tube} \\
d_2 &= .87in \quad \text{recooler tube} \\
\theta &= 180^\circ \quad \text{angle of contraction} \\
K_2 &= K_{2c} + K_{2e} = 15 \\
\Delta P &= .0001078 K_2 \rho v^2 \quad \text{“Crane cat” p3-4 (2)} \\
\rho &= 7.95 \frac{lb}{ft^3} \quad \text{fluid density} \\
v &= 1 \frac{ft}{s} \quad \text{flow velocity}(sec2.2) \\
\Delta P &= .0001078 \cdot 15 \cdot 7.95 \cdot 1 = .012psi \quad \text{pressure drop per one jont} \\
\Delta P_T &= 12\Delta P = .14psi \quad \text{total pressure drop per 12 sudden enlargement}
\end{aligned}$$

3.6.3.2 Pressure drop for 1″ dia tube bend

$$K_b = (n-1)\left(.25\pi f_T \frac{r}{d} + .5K\right) + K \quad (\text{“Crane” cat p.A-29 (2)})$$

$$\begin{aligned}
r &= 4d \quad \text{bend radii} \\
d &= .875'' \quad \text{tube diameter} \\
f_T &= .025 \quad \text{for steel pipe “crane” p.A-26 (2)} \\
K &= 14 f_T = .35 \quad for \ \frac{n}{d} = 4 \\
n &= 6 \cdot 2 = 12 \quad \text{number of bend per recooler} \\
K_b &= (12-1)(.25\pi \cdot .025 \cdot 4 + .5 \cdot .35) + .35 = 3.1 \\
\Delta P_b &= .0001078 \cdot .3.1 \cdot 7.95 \cdot 1^2 = .003psi
\end{aligned}$$

TOTAL PRESSURE DROP THRU ENTIRE SYSTEM:

$$\begin{aligned}
\Delta P &= \Delta P_r + \Delta P_c + \Delta P_T + \Delta P_b \\
\Delta P &= .023 + 6 \cdot .14 + .072 + .003 \approx \underline{.3psi}
\end{aligned}$$

$$\text{assume } .1atm = 1.5psi$$

3.6 Liquid requirement from the central liquifier for assumed max heat load of 100 W to the CLAS and flow rate of LHe 15g/s (sec 2)

3.7.1 The vapor generated in the subcooler-recooler box:

- heat removed from the magnet

$$Q_1 = 100W \text{ (sec 1.3)}$$

- heat removed from the "hot" incoming fluid from the pump

$$Q_2 = 8.7W(seeQ_T \text{ for subcooler sec 3.4})$$

$$\overset{o}{m}_{v1} = \frac{Q_1 + Q_2}{h_{fg}(at1.2atm, 4.42K)}$$

$$\overset{o}{m}_{v1} = \frac{100 + 8.7}{29.94 - 10.8} = 5.7g/s$$

3.7.2 The vapor generated within the pump dewar

Heat generated by the pump(for assumed WALTHER-MEISSNER PUMP ref 3).

For pressure drop across the system:

$$\Delta p = 1.5psi = 760cmLHe = 216\frac{lb}{ft^2}$$

and for flow rate of $\overset{o}{m}$=420$\frac{l}{h}$ and hydraulic efficiency about $\eta_h = .6$

Work done by the pump:

$$W = \frac{\Delta p \overset{o}{V}}{550} \frac{1}{\eta_h}$$

where:

$$\overset{o}{V} = 420l/\ h = .0042\frac{ft^3}{s} \quad \text{volumetric flow rate}$$

$$\Delta p = 216\frac{lb}{ft^2} \quad \text{head added}$$

$$W = \frac{216 \cdot .0042}{550 \cdot .6} = 2.8 \cdot 10^{-3}hp = 2.2W$$

- Heat load thru the pump connection- assume 2W
- Heat load into the pump dewar- assume 4W

Total Q_3=2.2+2+4=8.2W⇒ $\overset{o}{m}_2 = \frac{Q_3}{\Delta h_{fg}} \approx .4g/s$

Conclusion: The vapor rate generated within the pump dewar $\overset{o}{m} \approx .4$g/s

3.6.3 Flow requirement to cool power-loads

Liq He requirement to cool 2×10000A power leads is eq to 28 l/h=$7.8 \cdot 10^{-6} \frac{m^3}{s}$. (American Magnetic data)

For liquid at 1atm and 4.4K it's equivalent of mass flow:

$$\overset{o}{m}_3 = 7.8 \cdot 10^{-6} \frac{m^3}{s} \cdot 124 \frac{kg}{m^3} = .97 g/s$$

assume $\overset{o}{m}_3 = 1.3$ g/s

3.6.4 Required flow rate from ESR $\overset{o}{m} = m_1 + m_2 + m_3$

$$m = 5.7 + .4 + 1.3 = \underline{7.4 g/s}$$

4. LIQUID NITROGEN COOLING SYSTEM

The shield for each CLAS coil is cooled is series by liquid nitrogen delivered from the end station refrigerator. For reference see "Flow schematic diagram CLAS torus superconductive magnet" drwg 66100-D-00459

4.1 Cooling requirements

a) Heat load to the shield thru MLI

- Surface area of the coil shield
  $A_c = 90 m^2$
- Surface area of the rings shield (See "CLAS torus ass'y" drwg 66100-E-00402 for surfaces area)
  $A_r = 6 m^2$
- Surface area of the R-S and feed can
  $A_f = 15 m^2$

For 3/4" of MLI with layer density 50 layer/inch, coefficient of thermal conduction

$$k = 1 \cdot 10^{-9} \frac{W}{mK}$$

and: $Q_1 = \frac{A_T}{t} k \Delta T$

where:

$A_T = 117 m^2$ heat conduction area

$t = 3/4" = 1.9 \cdot 10^{-2} m$ thickness of MLI

$h = 1 \cdot 10^{-4} \frac{w}{mK}$ thermal conductivity of MLI

$\Delta T = 220 K$ temperature gradient shield-vacuum vessel

$$Q_1 = \frac{117}{1.9 \cdot 10^{-2}} \cdot 1 \cdot 10^{-4} \cdot 220 = 135W$$

b) Heat load thru coil suspension rods (“CLAS radial support rod” drwg 66100-E-00439)

Q=$\frac{A}{L} \int_{80}^{300} k_{ss}$ dT -per coil
$\int_{80}^{300}$ k dT = 27.1 $\frac{W}{cm}$ for SS
A=2A$_1$ + A_2 -total cross section area of the suspension rods
$A_1 = .32 in^2 = 2.07 cm^2$ radial rod
$A_2 = .11 in^2 = .71 cm^2$ axial rod
L=6in = 15.2cm length experiencing gradient 80→ 300K

$$Q = \frac{2 \cdot 2.07 + .71}{15.2} \cdot 27.1 = 8.7W$$

Total per magnet $Q_2 = 6 \cdot Q = 6 \cdot 8.7 = 52W$

c) Heat load thru out-of-plane suspension system (“CLAS 4K to 300K support ass'y” drwg 66100-B-00464)
Q=10$\frac{A}{L} \int_{80}^{300} k_{G-10} dT$ - per coil/10 suspension
$\int_{80}^{300} k_{G-10} dT = 1.28 \frac{W}{cm}$ for G-10
A= 1.2 in^2, L= 2.3in length experiencing gradient 80→ 300 K
Q=10 $\frac{1.2}{2.3}$1.28 = 6.7W
Total per magnet $Q_3 = 6 \cdot Q = 40W$

Total heat load to the LN$_2$ system

$$Q = Q_1 + Q_2 + Q_3 = 135 + 52 + 40 = 227W$$

Assuming 2× Q for unknown heat load, total load for calculation

$$Q \approx 700W$$

d) Flow requirement

If exit fluid quality be 1, required mass flow rate to remove 700W is

$$\overset{o}{m} = \frac{Q_T}{h_{f-g}} = \frac{700W}{199 \cdot 10^3 \frac{J}{kg}} = 3.5 \cdot 10^{-3} \frac{kg}{s} = 7.8 \cdot 10^{-3} \frac{lb}{s}$$

4.2 Pressure drop across the system if all cooling passages are connected in series.

a) Fricitional pressure drop is calculated according to Barron “Cryogenic system” (4)

$$\Delta p_f = \frac{L \left(\frac{dp}{dL} \right)_o}{x_2 - x_2} \int_{x_1}^{x_2} (1 - x)^{2-n} \phi_L^2 dx$$

$\left(\frac{dp}{dL}\right)_o$ - pressure drop per unit length if pure liq would flow

$\left(\frac{dp}{dL}\right)_o = 43.5\frac{f\rho_l q^2}{d^5}$ "Crane" catalog

Re=6.31 $\frac{\overset{o}{W}}{d\mu}$ Reynolds number for fluid only

where:

W=29$\frac{lb}{h}$ flow rate

d= .43in internal tube diameter

$\mu_f = 158 \cdot 10^{-3}$ centipoise fluid viscosity

$\mathrm{Re}_f = 6.31\frac{29}{.43 \cdot 158 \cdot 10^{-3}} = 2692$

and

f=$\frac{64}{Re}$ for Re $<$ 2800 "Barron" p.115

f= $\frac{64}{2692}$ = .023

q=$\frac{\overset{o}{m}}{\rho_f}$ —volumetric flow rate

where:

$$\overset{o}{m} = 7.8 \cdot 10^{-3}\frac{lb}{s} \quad \text{required flow rate}$$

$$\rho_f = 50.4\frac{lb}{ft^3} \quad \text{fluid density}$$

$$q = \frac{7.8 \cdot 10^{-3}}{50.4} = 1.6 \cdot 10^{-4}\frac{ft^3}{s}$$

and

$$\left(\frac{dp}{dL}\right)_o = 43.5\frac{.023 \cdot 50.4(1.6 \cdot 10^{-4})^2}{.43^5} = 8.8 \cdot 10^{5}\frac{psi}{ft}$$

—$\phi_L = \left(X^2 + CX + 1\right)^{1/2}/X$

$X^2 = \frac{C_L}{C_G}\frac{(Reg)^m}{(Ref)^n}\frac{\rho g}{\rho f}$

where:

Reg = 6.31 $\frac{W}{d\mu}$ Reynold nu if pure gas would flow thru the system

W=29 $\frac{lb}{h}$ flow rate

d=.43 tube diameter

$\mu_\rho = 6 \cdot 10^{-3}$ centipoise

Reg=6.31 $\frac{29}{.43 \cdot 6 \cdot 10^{-3}} = 7.1 \cdot 10^4$

Ref = 2692 Reynold nu if pure liquid would flow

$\rho_g = .36\frac{lb}{ft^3}$ gas density at 1.2 atm and 80K

$\rho f = 50.4\frac{lb}{ft^3}$ liq density

$$\left.\begin{array}{l} C_L = 64 - \text{for liq} \\ C_G = .184 - \text{for vapor Re} > 50000 \\ \text{n} = 1 - \text{for liquid} \\ \text{m} = .2 - \text{for vapor Re} > 50000 \end{array}\right\} \quad \text{table 7-19 Barron (4)}$$

$$X^2 = \frac{64(7.1 \cdot 10^4)^{.2}.36}{.184(2692)^1 \cdot 50.4}\left(\frac{1-x}{x}\right)^2$$

$$X = .093\left(\frac{1-x}{x}\right)$$

and for laminar liquid and turbulent gas C=12 (page 416 Barron)

so:

$$\phi_L = \frac{\left\{\left[.093(\frac{1-x}{x})\right]^2 + 12\cdot .093(\frac{1-x}{x}) + 1\right\}^{1/2}}{.093(\frac{1-x}{x})} =$$

$$= \frac{(-12.43x^2 + 127.8x + 1)^{1/2}}{1-x}$$

for condition as follows:

x_1=.01 fluid quality at inlet
x_2=.99 fluid quality at outlet
L=6L$_c$ + $L_s + L_B$ – tube length per CLAS
L_c = 60ft tube length per coil
L_s = 20ft tube length per subcooler
L_B = 20ft tube length per feed box and connection
L = 6·60 + 20 + 20 total tube length per ass'y
$\left(\frac{dp}{dL}\right)_o = 8.8\cdot 10^{-5}\frac{psi}{ft}$

$$\Delta p_f = \frac{400\cdot 8.8\cdot 10^{-5}}{1}\int_{.01}^{.99}\frac{-12.43x^2 + 127.8x + 1}{1-x}dx$$

$$= .0352\int_{.01}^{.99}\frac{-12.43x^2 + 127.8x + 1}{1-x}dx$$

$$= .0352\left[-\int_{.01}^{.99}\frac{-12.43x^2}{1-x}dx + \int_{.01}^{.99}\frac{127.8x}{1-x}dx + \int_{.01}^{.99}\frac{1}{1-x}dx\right]$$

$$= .0352\left\{\ 12.43\left(\frac{1}{2}(1-x)^2 - 2(1-x) + ln(1-x)\right)\Big|_{.01}^{.99}x\right.$$

$$\left. +127.8\left[-x - ln(1-x)\right]\Big|_{.01}^{.99} - ln(1-x)\Big|_{.01}^{.99}\right\} =$$

$$= 14.7 \quad \text{psi}$$

b) pressure drop due to change of momentum

$$\Delta pm = \phi_M(\overset{o}{m}_L + \overset{o}{m}_g)^2/g_c\rho_L A^2 \quad \text{Barron p. 417}$$

$$\phi_M = \frac{(1-x_2)^2}{R_{L2}} - \frac{(1-x_1)^2}{R_{L1}} + \left(\frac{x_2^2}{1-R_{L2}} - \frac{x_1^2}{1-R_{L1}}\right)\left(\frac{\rho_l}{\rho_\rho}\right)$$

$$R_L = \frac{X}{(X^2 + cX + 1)^{1/2}} = \frac{X}{(X^2 + 12X + 1)^{1/2}}$$

for inlet x_1=.01
$X_1 = 92.91$ and $R_{L1} = .941$
for outlet x_2 =.99
$X_2 = 9.31 \cdot 10^5$ and $R_{L2} = 9.3 \cdot 10^{-5}$
and
$\phi_M = \frac{(1-.99)^2}{9.3 \cdot 10^{-5}} - \frac{(1-.01)^2}{.941} + \left(\frac{.99^2}{1-9.3 \cdot 10^{-5}} - \frac{.01^2}{1-.941}\right)\frac{50.4}{.36} = 139$

$$m_L = m_g = 7.8 \cdot 10^{-3}\frac{lb}{s}$$

$$g_c = 32.2\frac{lbft}{s^2 lbf}$$

$$\rho_L = 50.4\frac{lb}{ft^3}$$

$$A = \frac{\pi\left(\frac{.44}{12}\right)^2}{.4} = 1.06 \cdot 10^{-3} ft^2 \quad \text{tube area}$$

$$\Delta pm = \frac{139\left(7.8 \cdot 10^{-3}\right)^2}{32.2 \cdot 50.4\left(1.06 \cdot 10^{-3}\right)^2} = 4.7\frac{lbf}{ft^2} = .03psi$$

c) Total pressure drop across the LN_2 system

$$\Delta p = \Delta pf + \Delta pm = 14.7 + .03 \approx \underline{15psi}$$

4.3 Max shield temperature

a) heat transfer coefficient within the tube h=$\frac{N_{Nu}k}{d}$

$$N_{Nu} = .023(Re)^{.8}(Npr)^{.4}$$

- for liquid
Re=2692 sec. 4.2
k= $139.6 \cdot 10^{-3}\frac{W}{mh}$ thermal conductivity of liquid
$N_{pr} = 2.14$ Prandtl number
D= .44in = $1.73 \cdot 10^{-3}$ m
h=$\frac{.023(2692)^{.8}(2.14)^{.4} \cdot 139.6 \cdot 10^{-3}}{1.73 \cdot 10^{-3}} = 1.94 \cdot 10^3\frac{W}{m^2K} = 1.25\frac{W}{in^2K}$
- for gas
Re= 71000 sec. 4.2
N_{pr} =.8 Prandtl number for gas at 80K
k=$7.23 \cdot 10^{-3}\frac{W}{mK}$ thermal conductivity of gas
h= $\frac{.023(71000)^{.8}(.8)^{.4} \cdot 7.23 \cdot 10^{-3}}{1.73 \cdot 10^{-3}} = 6.68 \cdot 10^2\frac{W}{m^2K} = .043\frac{W}{in^2K}$

b) Max shield temperature for worst heat transfer coefficient (for gas) h=.043 $\frac{W}{in^2K}$
Use electrical circuit analogy according to schematic (see "CLAS LN_2 shield" drwg 66100-E-00455)

$$\frac{28q_1^2}{A}\frac{\Delta x}{k} + \frac{7q_1^2}{A_1 h} = \Delta T_{max}$$

q_1-heat load per 1in Δ x surface

$q_1 = \frac{Q}{A} \cdot \Delta x$

where:

$\frac{Q}{A} = 1.15\frac{W}{m^2} = 7.5 \cdot 10^{-4}\frac{W}{in^2}$ heat load per unit area

Δx=4in

$q_1 = 7.5 \cdot 10^{-4} \cdot 4 = 3 \cdot 10^{-3}$W

A=$\frac{1}{16}in^2$ cross section of 1/16" thick shield per unit length

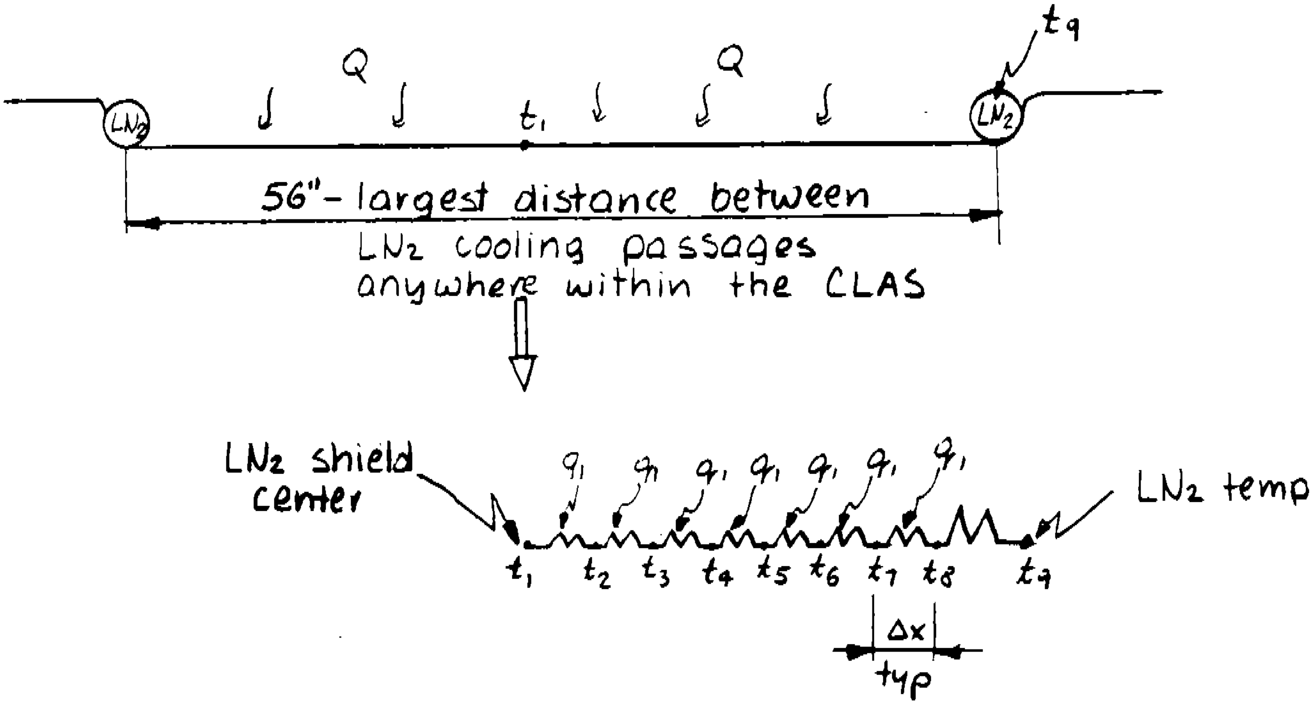
$A_2 = \frac{1}{4}\pi d = \frac{1}{4} \cdot 3.14 \cdot .44 = .34in^2$ heat transfer area of the cooling tube (assumed $\frac{1}{4}$ of total)

k=5.1$\frac{W}{inK}$ thermal conductivity of Cu at 80K

h=.043$\frac{W}{in^2K}$ heat transfer coefficient

$$\Delta T = \frac{28 \cdot 3 \cdot 10^{-3} \cdot 4}{1/10 \cdot 5.1} + \frac{7 \cdot 3 \cdot 10^{-3}}{.34 \cdot .043} = 1.1 + 1.4 = \underline{2.5K}$$

Max temperature gradient across the shield is 2.5K



APPENDIX C

MAGNET STRESS ANALYSIS

1 Stress within the bobbin plate and the vacuum vessel

1.1 In plane loading, perfect symmetry assumed.

When magnet is energized to the full field, coil is loaded by the magnetic loading shown on Fig. 1^* calculated by internal CEBAF computer program. Due to symmetry out-of-plane coil load is eq. to zero, only large in plane forces are present, forcing the coil to get round.

Stress analysis has been performed with FE code IDEAS. Due to the lack of memory space only coil enclosure (bobbin) have been created with loading attached to the outer periphery.

Aluminum bobbin plate is assumed to be held in front end by the Al ring and is free to translate along the back rings. Bobbin plate is free to rotate around the magnet axis.

Maximum expected Von Mises stress: within the bobin plate is $1.3 \cdot 10^4$psi(Fig 2^*), within the front Al ring 7500 psi and the back SS rings $3.2 \cdot 10^4$psi(Fig 3^*,4^*)

1.2 Out-of-plane loading.

Because of evident lack of symmetry during manufacturing process, coil have to be held in transverse direction to not get in contact with the vacuum vessel. In order to take this load, 5 spring loaded spacers attached to the vacuum vessel are used (on each coil side). Each spacer has spring constant of 54000 lb/in to balance magnetic spring constant of 4000 lb/in in most loaded spacer. Maximum Von-Mises stress under one coil misplacement in reference to rest of the magnet of .2° (see Fig 6^* for loading for this case) is 2800 psi and is very local (see Fig 5^*).

Most severe loading case, can be imagined for LAS is lack of current within one coil when huge out-of-plane force within the system would exist (fig 7). This load would be transfered thru mentioned above spacers and array of secondary spacers (located about .25" from the bobbin) to the vacuum vessel. Assuming that vacuum vessel is held in place by array of rods keeping six coils in symmetry, max expected Von-Mises stress within the vessel is about 37kpsi (max local see Fig 8^*). Location of this rods can be probably optimized to bring this stress down, however Div 2, ASME code allow primary plus secondary stress intenstiy to be lower than $2Sy = 2 \cdot 30000 = 60000$psi

1.3 Stress within the vacuum vessel

During the study state operation vacuum vessel is loaded by the external atmospheric pressure. In order to decrease stress level due to this load, array of vacuum spacers is provided. Max stress level calculated according to ASME sec. VIII boiler code is about 8000psi well checked with IDEAS.

2 CLAS suspension system

CLAS suspension system contain: ("CLAS torus ass'y" drwg 66100-E-00462)

- 10 out of plane support block (per coil) to take any out of plane loading during the coil operation
- 1 links restraining the coil in horizontal direction (per coil)
- 2 links restraining the coil in radial direction (per coil)
- 5 links restraining the coil cases for the dead weight and fault load (per coil pair)

2.1 Out-of-plane support ("CLAS single coil ass'y" drwg 66100-E-00460)

If coil within CLAS magnet would be kept in the ideal position, all out-of-plane force due to magnetic field would cancel. However any distortion of the symmetry would cause magnetic out-of-plane forces which have to be taken care of.

Magnitude of this force can be calculated with internal CEBAF program (5).

With FE IDEAS, force per support can be calculated for various coil displacement in out-of-plane. Largest ratio force/displacement was found to be 4000 lb/in. In order to suppress run away condition support member have to have spring constant larger than k_m = 4000 lb/in.

Our design use fiberglass block, loaded with Belleville washers with spring constant about k_s=54000 lb/in.

Initially at room temperature this support block have to be preloaded to the following value:

- load due assumed misplacement

$$P_1 = \Delta x K_m$$

where
Δx=.1 in coil displacement at the heaviest loaded support for equivalent angular rotation of .1°

$$P_1 = .1 \cdot 4000 = \underline{400 lb}$$

- load required to make up for differential thermal contraction:

$\Delta l = 2 \cdot h(\frac{\Delta h}{h}) + t(\frac{\Delta t}{t})$
h=2.6in support height
$\frac{\Delta h}{h}$ for G-10 with 4 → 300 K gradient$=1.6 \cdot 10^{-3} \frac{in}{in}$
t=.5in thickness of Al bobbin plate
$(\frac{\Delta t}{t})_{Al} = 4.2 \cdot 10^{-3} \frac{in}{in}$ thermal contraction of Al

$$\Delta l = 2 \cdot 2.6 \cdot 1.6 \cdot 10^{-3} + .5 \cdot 4.2 \cdot 10^{-3} = 1.04 \cdot 10^{-3} in$$

$$P_2 = \Delta l \cdot ks = 1.04 \cdot 10^{-3} \cdot 54000 = \underline{56 lb}$$

Total preload at room temperature

$$P = P_1 + P_2 = 400 + 56 = 456lb$$

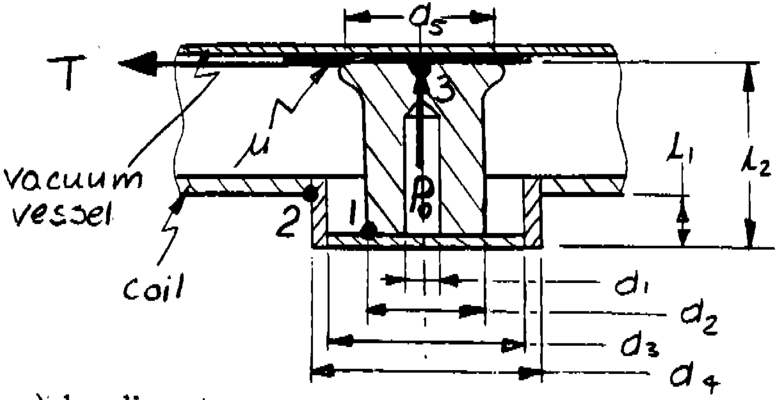
Max expected load should occur during the cooldown and is eq to:

$$P_o = P + P_g$$

where: P=456 lb preload
$P_g = \frac{W}{5} = \frac{3500}{5}$=700lb dead weight per support

$$P_o = 456 + 700 = 1156lb$$

Max expected stress for the model shown below:



$l_1 = 1.07$
$l_2 = 2.32$
μ=.2

d_1=5
d_2 =1.25
d_3=2.87
d_4=3.0
d_5=1.5

a) bending stress

$$\sigma_{b1} = \pm\frac{Tl_2\frac{d_2}{2}}{\frac{\pi}{4}\left(\left(\frac{d_2}{2}\right)^4 - \left(\frac{d_1}{2}\right)^4\right)} = \pm\frac{P_o\mu l_2 d_2}{\frac{\pi}{2}\left(\left(\frac{d_2}{2}\right)^4 - \left(\frac{d_1}{2}\right)^4\right)} = \pm\frac{1156\cdot .2\cdot 2.32\cdot 1.25}{\frac{\pi}{2}\left(\left(\frac{1.25}{2}\right)^4 - \left(\frac{.5}{2}\right)^4\right)}$$

$\sigma_{b1} = \pm 2872$ psi

$$\sigma_{b2} = \pm\frac{T(l_1 + l_2)\frac{d_4}{2}}{\frac{\pi}{2}\left(\left(\frac{d_4}{2}\right)^4 - \left(\frac{d_3}{2}\right)^4\right)} = \pm\frac{P_o\mu\left(l_1 + l_2\right)d_4}{\frac{\pi}{2}\left(\left(\frac{d_4}{2}\right)^4 - \left(\frac{d_3}{2}\right)^4\right)} = \pm\frac{1156\cdot .2(1.07 + 2.32)\cdot 3}{\frac{\pi}{2}\left(\left(\frac{3}{2}\right)^4 - \left(\frac{2.87}{2}\right)^4\right)}$$

$\sigma_{b2} = 1821$ psi

b) compressive (tension) stress

$$\sigma_1 = \frac{P_o}{A_1} = -\frac{P_o}{\frac{\pi}{4}\left(d_2^2 - d_1^2\right)} = \frac{1156}{\frac{\pi}{4}\left(1.25^2 - .5^2\right)} = 1122 \text{ psi}$$

$$\sigma_2 = \frac{P_o}{A_2} = \frac{P_o}{\frac{\pi}{4}\left(d_4^2 - d_3^2\right)} = \frac{1156}{\frac{\pi}{4}\left(3^2 - 2.87^2\right)} = 1873 \text{ psi}$$

$$\sigma_3 = \frac{P_o}{A_3} = \frac{P_o}{\frac{\pi}{4}d_5^2} = \frac{1156}{\frac{\pi}{4}\cdot 1.5^2} = 655 \text{ psi}$$

c) shear stress

$$\tau_1 = \frac{T}{A_1} = \frac{P_o\mu}{\frac{\pi}{4}\left(d_2^2 - d_1^2\right)} = \frac{1156 \cdot .2}{\frac{\pi}{4}\left(1.25^2 - .5^2\right)} = 224 \text{ psi}$$

$$\tau_2 = \frac{T}{A_2} = \frac{P_o\mu}{\frac{\pi}{4}\left(d_4^2 - d_3^2\right)} = \frac{1156 \cdot .2}{\frac{\pi}{4}\left(3^2 - 2.87^2\right)} = 386 \text{ psi}$$

$$\tau_3 = \frac{T}{A_3} = \frac{P_o\mu}{\frac{\pi}{4}d_5^2} = 130 \text{ psi}$$

d) principal stresses

$$\sigma_{1(1.2)} = \frac{\sigma_{b1} + \sigma_1}{2} \pm \sqrt{\left(\frac{\sigma_{b1} + \sigma_1}{2}\right)^2 + \tau_1}$$

$$\sigma_{1(1.2)} = -\frac{2872 + 1122}{2} \pm \sqrt{\left(\frac{2872 + 1122}{2}\right)^2 + 224^2}$$

$$\sigma_{1.1} = -4006 \text{ psi}$$

$$\sigma_{1.2} = -12 \text{ psi}$$

$$\sigma_{2(1.2)} = \frac{\sigma_{b2} + \sigma_2}{2} \pm \sqrt{\left(\frac{\sigma_{b2} + \sigma_2}{2}\right)^2 + \tau_2}$$

$$\sigma_{2(1.2)} = \frac{1872 + 1821}{2} \pm \sqrt{\left(\frac{1872 + 1821}{2}\right)^2 + 386^2}$$

$$\sigma_{2.1} = 3733 \text{ psi}$$

$$\sigma_{2.2} = -40 \text{ psi}$$

$$\sigma_{3(1.2)} = \frac{\sigma_3}{2} \pm \sqrt{\left(\frac{\sigma_3}{2}\right)^2 + \tau_3^2}$$

$$\sigma_{3(1.2)} = \frac{655}{2} \pm \sqrt{\left(\frac{655}{2}\right)^2 + 386^2}$$

$$\sigma_{3.1} = 833 \text{ psi}$$

$$\sigma_{3.2} = -179 \text{ psi}$$

e) Von Mises stress

$$\sigma_i^1 = \sqrt{\sigma_{i1}^2 + \sigma_{12}^2 - \sigma_{i1}\sigma_{i2}}$$

$$\sigma_1^1 = \sqrt{4006^2 + 12^2 - 4006 \cdot 12} = 4000 \text{ psi} \;\; < \text{Sy for G} - 10$$

$$\sigma_2^1 = \sqrt{3733^2 + 40^2 + 3733 \cdot 40} = 3753 \text{ psi} \;\; < \text{for G} - 10$$

$$\sigma_3 = \sqrt{833^2 + 179^2 + 179 \cdot 833} = 935 \text{ psi} \;\; < \text{ Sy for}$$

Teflon slidding pad (recommended value 1400 psi)

2.2 Horizontal link ("CLAS AFT in-plane horizontal suspension ass'y" drwg 66100-E-00471)

Horizontal link is designed to take:

- cooldown frictional force within out-of-plane support blocks eq to:

$$P_f = n \cdot P_o \mu$$

where:
n=10 number of out-of-plane support per coil
P_o=1156 lb expected load per out-of-plane support (sec. 2.1)
μ =.2 assumed coefficient of friction of interface
$P_f = 10 \cdot 1156 \cdot .2 = 2032 lb$

- acceleration load of 1.5g

$$P_s = 1.5W$$

where:
W=3500 lb weight of one coil

$$P_s = 1.5 \cdot 3500 = 5250 lb$$

Total max expected load:

$$P_T = P_f + P_s = 2032 + 5250 = 7282 lb$$

Required rod diameter:

$$\sigma = \frac{P_T}{A} = \frac{2}{3}\text{Sy}(= 130000 \text{ psi for Nitronic 50})$$

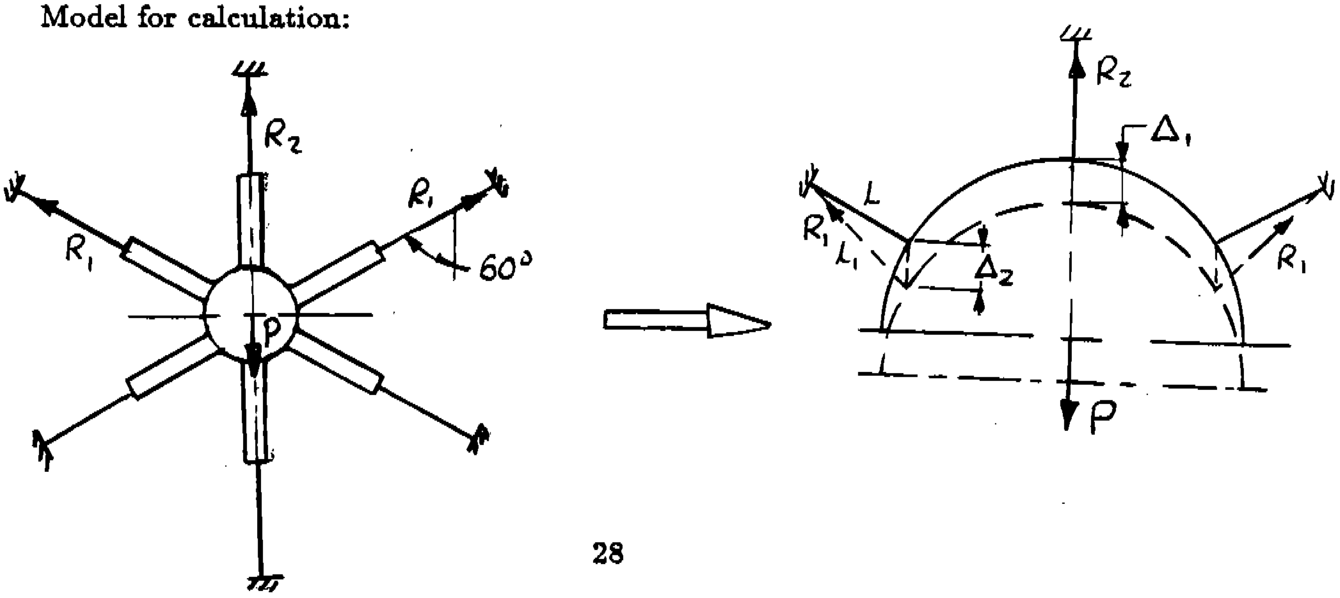
$$A = \frac{7282}{2/3 \cdot 130000} = .084 in^2$$

$$d = .32 in \quad \text{assumed } d = 3/8$$

2.3 Radial links ("CLAS AFT in-plane radial suspension ass'y" drwg 66100-E-00472 "CLAS forward in-plane radial suspension ass'y" drwg 66100-E-00473)

Design load: P=2W=2 · 21000=42000lb (magnet weight + 1g)

Model for calculation:



$$2R_1 cos60^\circ + R_2 = P \Rightarrow R_1 + R_2 = P$$

$$\Delta_1 = \frac{R_2 l}{EA}$$

$$y = l sin30 = .5l$$

$$x = l cos30 = .866l$$

$$l_1 = (x^2 + (y + \Delta_z)^2)^{1/2} = (.75l^2 + (.5l + \Delta_z)^2)^{1/2} =$$

$$= (l^2 + l\Delta_z + \Delta_z)^{1/2}$$

$$l_1 - l = \Delta l = \frac{R_1 l}{EA} = (l^2 + l\Delta_z + \Delta_z^2)^{1/2} - l$$

$$\Delta_1 = \frac{R_2 l}{EA}$$

since $\Delta_1 = \Delta_z$

$$\frac{R_1}{EA} = \left(l^2 + \frac{R_2}{EA} + \frac{Rl_2^2}{E^2 A^2}\right)^{1/2} - l$$

$$\frac{R_1}{EA} + 1 = \left(1 + \frac{R_2}{EA} + \frac{R_2^2}{E^2 A^2}\right)^{1/2}$$

$$\frac{R_1^2}{E^2 A^1} + 2\frac{R_1}{EA} + 1 = 1 + \frac{R_2}{EA} + \frac{R_2^2}{E^2 A^2}$$

$$\frac{R_1^2}{EA} + 2R_1 = R_2 + \frac{R_2^2}{EA}$$

but $R_1 = P - R_2 = 42000 - R_2$

$$\frac{(42000 - R_2)^2}{EA} + 84000 - 2R_2 = R_2 + \frac{R_2^2}{EA}$$

$$\frac{1.76 \cdot 10^9 - 8.4 \cdot 10^4 R_2}{EA} + \frac{R_2^2}{EA} + 84000 = 3R_2 + \frac{R_2^2}{EA}$$

for assumed d=5/8 dia, A=.31 in^2
and E=28$\cdot$10^{6}psi

$328 - 1.58 \cdot 10^{-2} R_2 + 84000 = 3R_2$ (weak dependence to diameter)

$$R_2 \approx 28000 lb/per2$$

$R_1 = 42000 - 28000 = 14000 lb/per$ two links - assume the same load for front and back links since fill angle is small

Required rod diameter;
for Sy=130ksi

$$A = \frac{28000 \cdot 1/2}{2/3 \cdot 130000} = .17$$

$$d = .47dia \qquad \text{use } 5/8''$$

2.3.1 Because of large magnet thermal contraciton, helical springs are used to suspend the magnet in radial direction.

Spring design for the front radial links:

Thermal contraction of the cold mass in radial direction up to 300K link connection:

$\Delta l_{ss} = 36.8 \cdot 4.2 \cdot 10^{-3}$ =.1544in contraction of Al from ring from 300→ 4K

$\Delta l_{AL} = 45.3 \cdot 4.2 \cdot 10^{-3}$ = .1905in contraction of Al portion from ring to attachment point from 300→ 4K

$\Delta l_r = 1.6 \cdot 10^{-3} \cdot 18$ = .0288in contraction of tension rod (experience gradient 300 → 4K)

$\Delta = \Sigma\Delta l = .3297in \approx$ .33in

Max load of the spring during the operation:

$$P = \frac{1}{4}R_2 = 7000lb(\text{with no g load})$$

Assume that cooldown spring load won't be larger than P_c=2500 lb.

Required spring constant should be:

$$k_{sp} = \frac{2500}{\Delta} = \frac{2500}{.33} = 7580\frac{lb}{in}$$

Required min spring deflection:

$$y = \frac{P + P_c}{k_{sp}} = \frac{7000 + 2500}{7580} = 1.25in$$

Assume helical spring.

Wire diameter is calculated from formula

$$d = \left(\frac{2.55PD}{s}\right)^{1/2} \quad \text{“Machinery handbook” (6)}$$

where:

$$P = P + P_c = 9500lb$$

$$D = 4.0in \text{ assumed mean spring diamter}$$

$$S = 70000psi \text{ working stress of 316 LN}$$

$$d = \left(\frac{2.55 \cdot 9500 \cdot 4}{70000}\right)^{1/2} = 1.12in$$

assume d=1.125

Number of active coils:

$$N = \frac{Gd^4F}{8PD^3}$$

where:
G=$11 \cdot 10^6$ psi shear modulus of 316LN
d= 1.125 in wire diameter
F= 1.5in required deflection
P=9500lb operating load
D= 4" mean spring diameter

N=$\frac{11\cdot10^6\cdot1.125^4\cdot1.5}{8\cdot9500\cdot4^3}$=5.4 assume 6

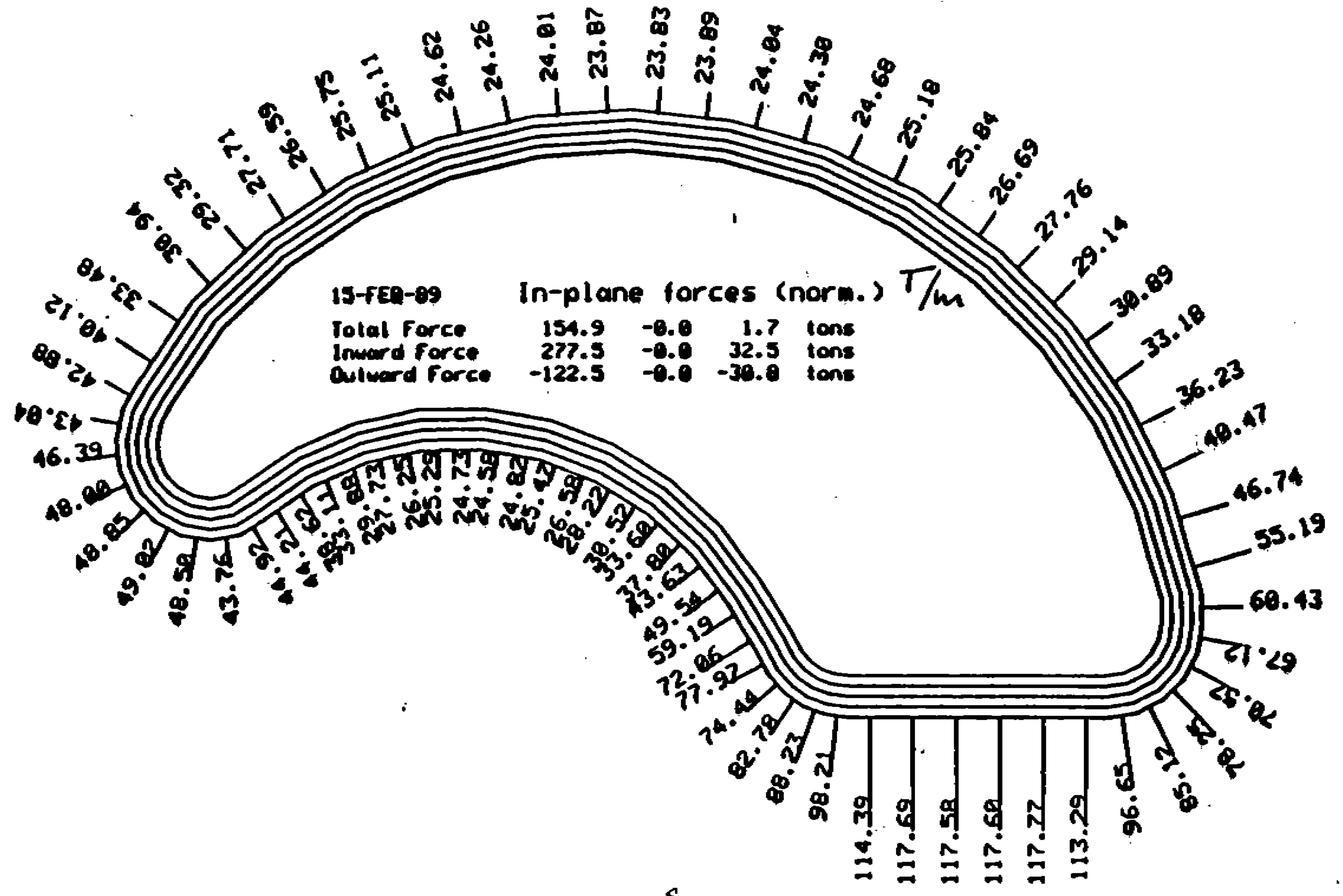
Spring length (closed ends and ground)
- pitch $p = \frac{F}{N} + d = \frac{1.5}{6} + 1.25 = 1.375$
- free length FL=pN + 2d = $1.375\cdot6 + 2 \cdot 1.125 = 10.5$

Forward links (drwg 66100-E-00473) are lighter loaded (thermal contraction to the point of attachment is smaller) so safety margin will be higher.

2. CLAS coil-coil support (drwg 66100-E-00433)
   Max expected load per link for fault condition (one coil off) has been calculated with IDEAS and is eq to 22000 lb. Expected material used for this member is carbon fibre composite Sy(min)=100000 psi and requried dia should be:

$$d = \sqrt{\frac{4}{\pi}\frac{P}{2/3Sy}} = \sqrt{\frac{4}{\pi}\frac{22000}{2/3\ 100000}} \approx \underline{.64in}$$

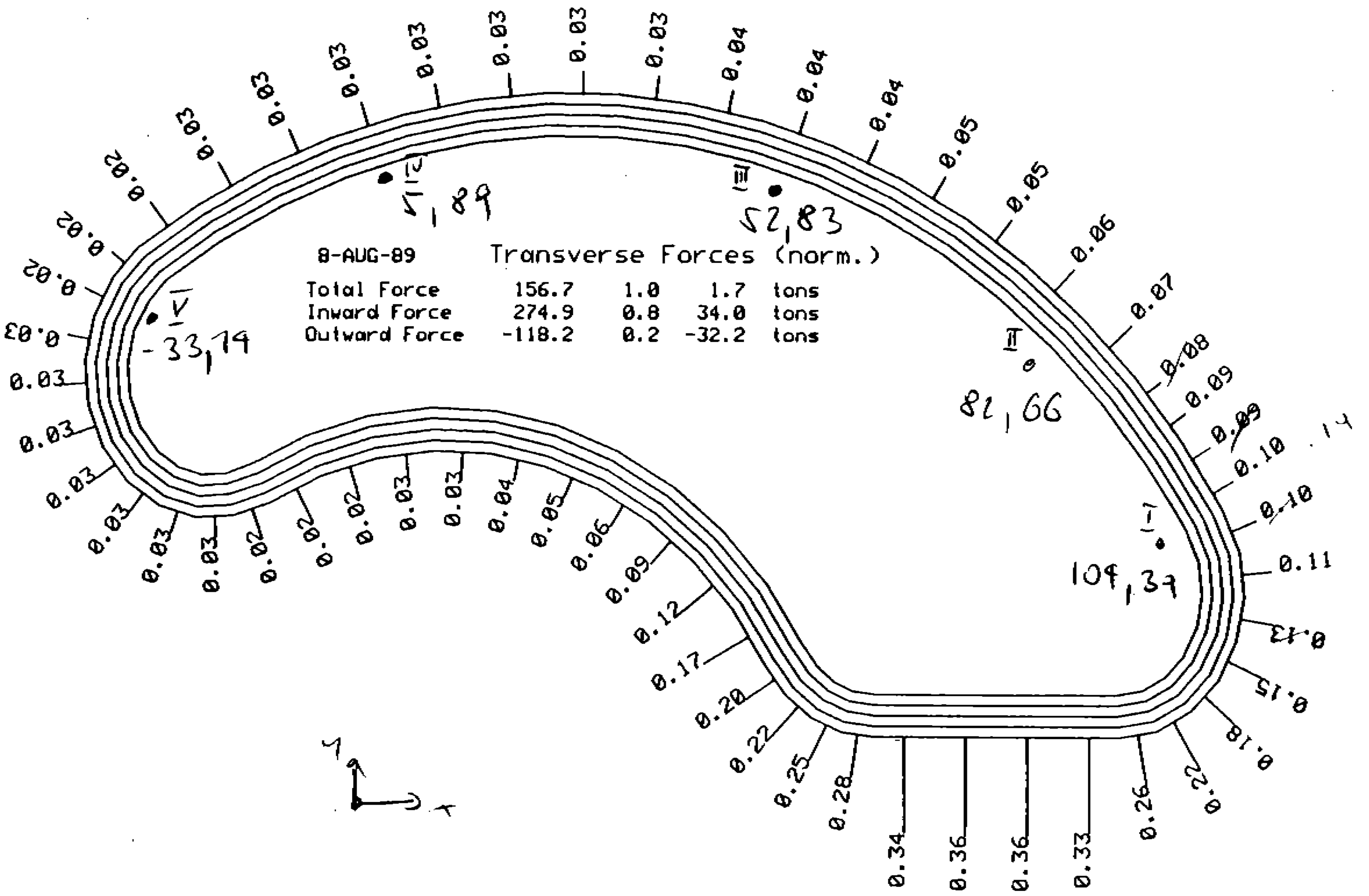
Coil - in-plane loading



Total Force	154.9	-0.0	1.7	tons
Inward Force	277.5	-0.0	32.5	tons
Outward Force	-122.5	-0.0	-30.0	tons

fig 1^s

COIL .2° OUT



8-AUG-89 Transverse Forces (norm.)

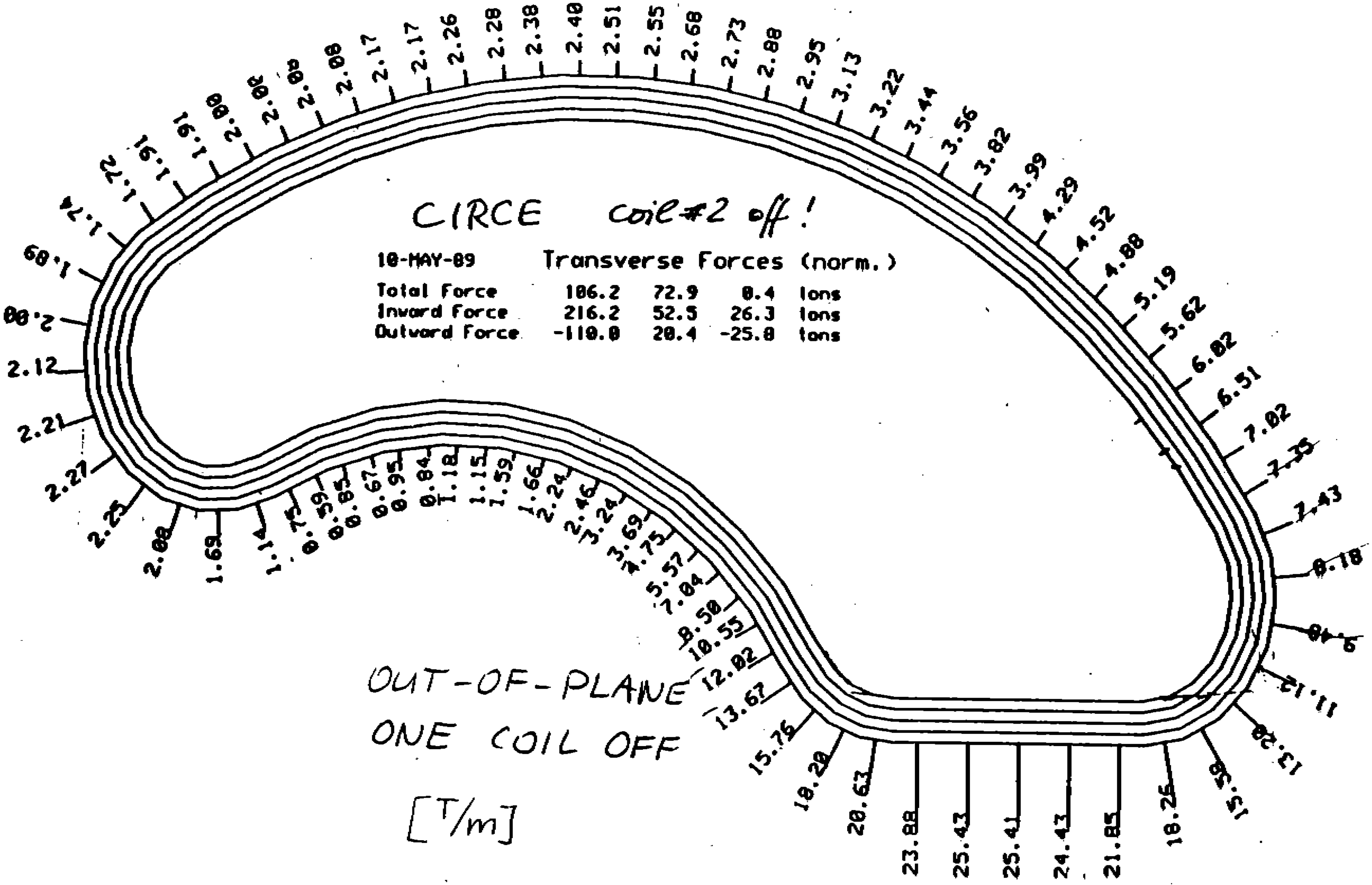
Total Force	156.7	1.0	1.7	tons
Inward Force	274.9	0.8	34.0	tons
Outward Force	-118.2	0.2	-32.2	tons

Forces $[\frac{T}{m}]$

I.... V Out-of-plane support locations

Fig 6s

CIRCE coil#2 off!

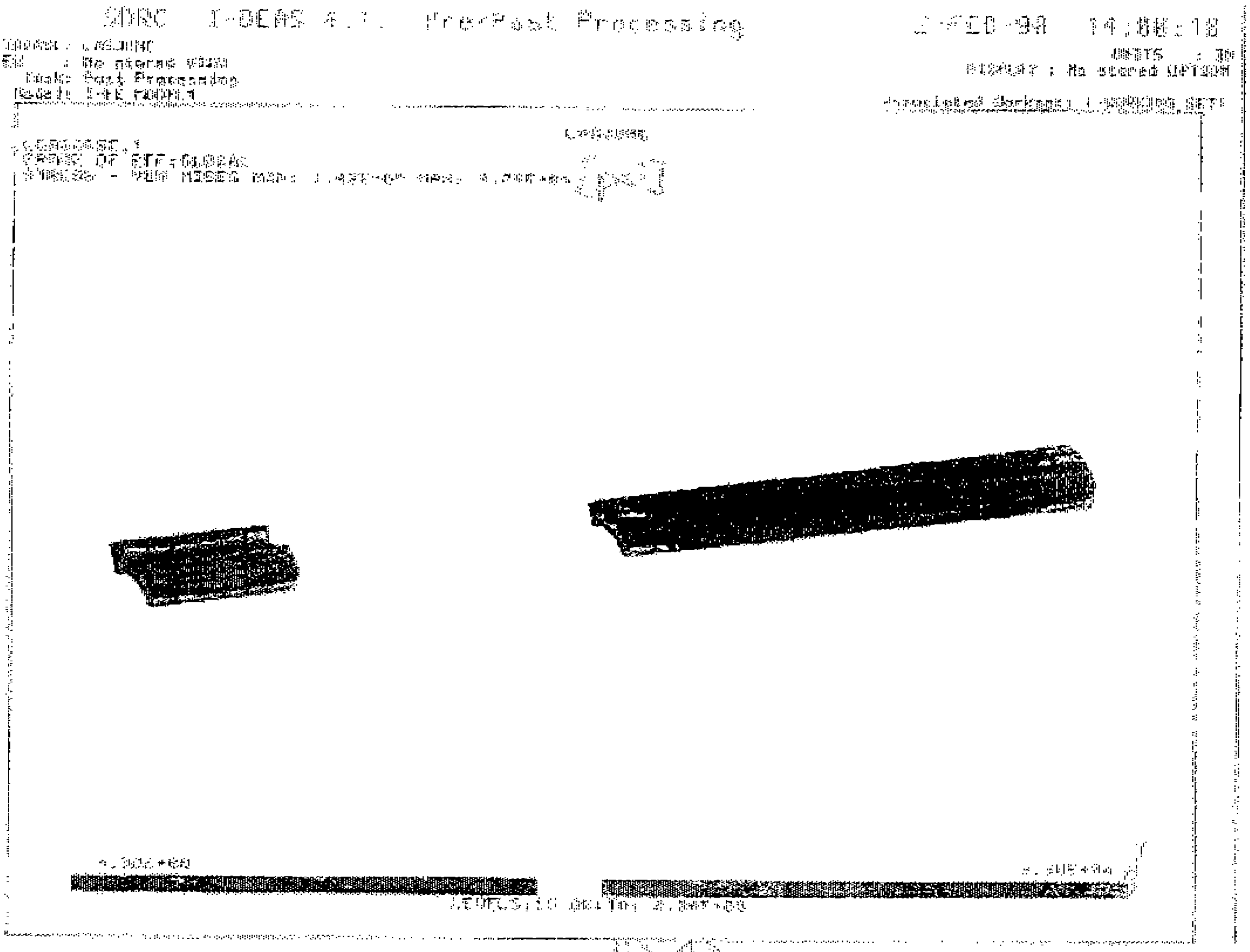

10-MAY-89 Transverse Forces (norm.)

Total Force	106.2	72.9	0.4	tons
Inward Force	216.2	52.5	26.3	tons
Outward Force	-110.0	20.4	-25.8	tons

OUT-OF-PLANE
ONE COIL OFF

[T/m]

Fig 7s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14:08:18
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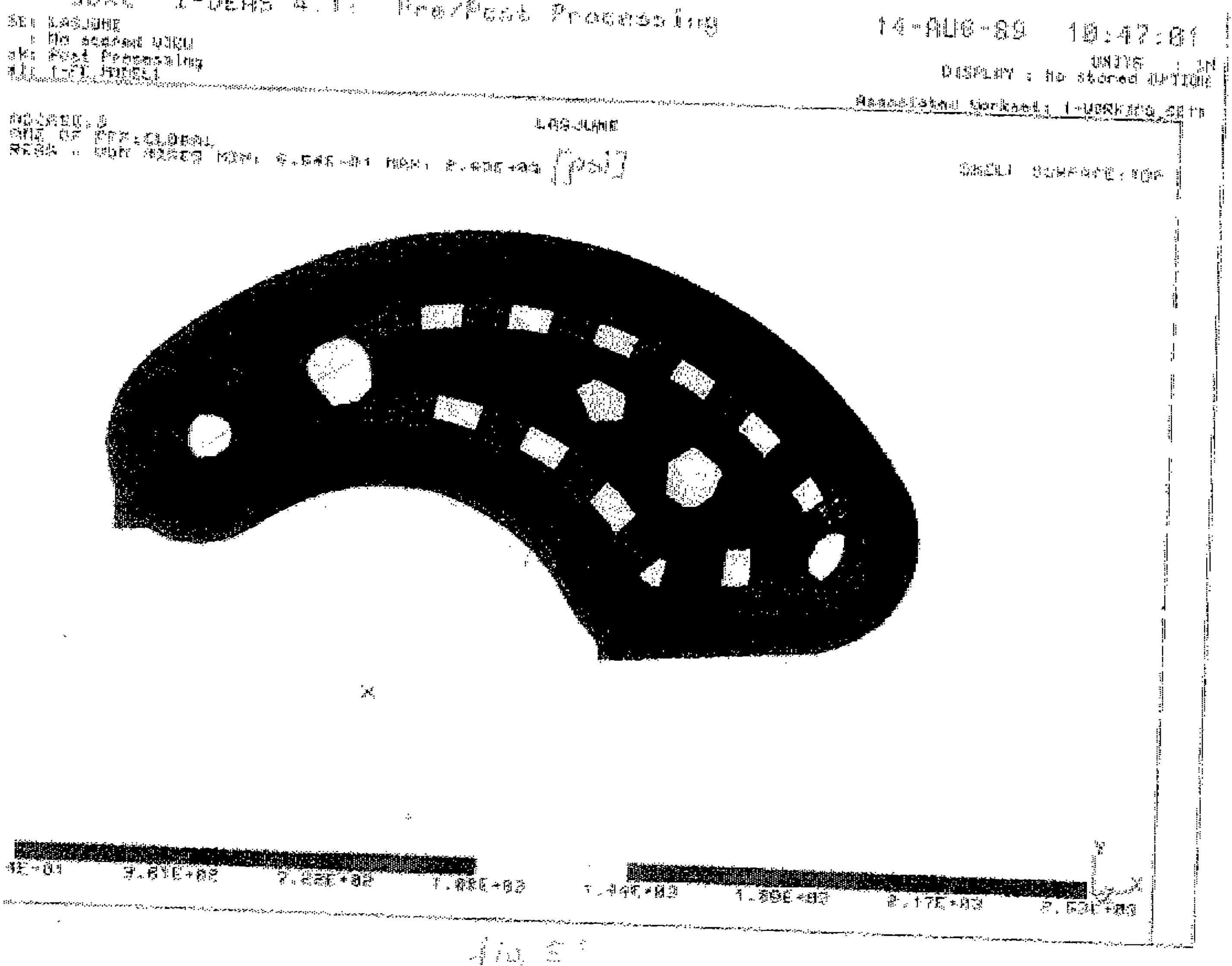
Task: Post Processing
UNITS : IN
DISPLAY : No stored OPTION
Associated Workset: [illegible]
FRAME OF REF: GLOBAL
STRESS - VON MISES MIN: [illegible] MAX: [illegible]
LEVELS: 10 DELTA: [illegible]

Fig 4

COIL .2° out



SDRC I-DEAS 4.1: Pre/Post Processing

14-AUG-89 10:47:01

SE: LASJUNE
: No stored VIEW
sk: Post Processing
ML: 1-FE MODEL1

UNITS : IN
DISPLAY : No stored OPTION

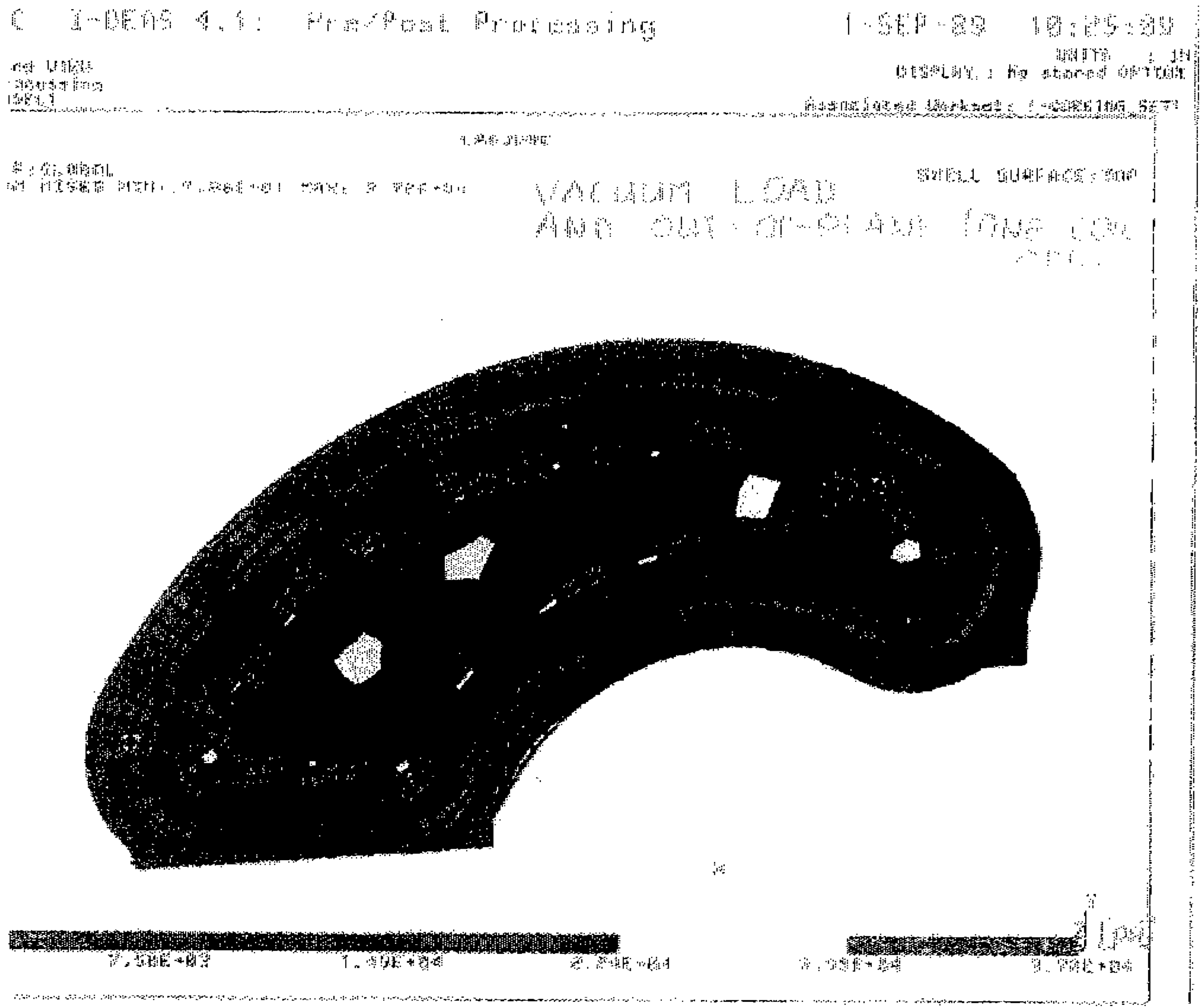
Associated Workset: 1-WORKING SET1

LOADCASE, 1
SYS OF REF: GLOBAL
STRESS - VON MISES MIN: 9.64E-01 MAX: 2.53E+03 [psi]

LASJUNE

SHELL SURFACE: TOP

fig 5



C I-DEAS 4.1: Pre/Post Processing 1-SEP-89 10:25:39

UNITS : IN
DISPLAY : No stored OPTION

Associated Workset: [illegible]

SHELL SURFACE: TOP

VACUUM LOAD
AND OUT-OF-PLANE [illegible]

Fig 25

References:

1. Russel B. Scott "Cryogenic Engineering" Edition 1988 by Met-Chem Research Inc
2. Crane "Flow of fluids through valves, fittings and pipe" Technical paper no 410
3. H. Berndt, R. Doll, W. Wiedemann "Two years experience in liquid helium transfer with a maintenance free contrifugal pump" publ. DF-02 presented at CEC89 Los Angeles
4. Randal F. Barran "Cryogenic systems" Oxford 2nd edition
5. "Shoforce" computer program by CEBAF
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