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RESPONSE AND PERFORMANCE DESIGN CORRELATION

G. R. Jones
E. H. Christensen

General Dynamics Convair Division
P. O. Box 83577
San Diego, California 92138

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ABSTRACT

Through General Dynamics Convair Division's experience in the design, detail analysis, and manufacturing of structural supports for superconducting magnet cryostats suspended in a vacuum enclosure, a data base, well suited for the development of correlations of pertinent thermal performance criteria for stainless steel supports, has been created. The thermal requirements of these supports in fusion applications are well defined for the Mirror Fusion Test Facility (MFTF) and have been analyzed in detail for cool-down response and steady-state performance, using Convair's THERMAL ANALYZER computer program.¹ From the output of these thermal conditioning simulations, correlations were developed for magnet LHe heating from supports in terms of strut geometric parameters.

The correlations enable the user to conservatively estimate transient and steady-state heat loads and the approximate time that the steady-state operation conditions are reached during magnet cool-down. The dependent variables employed in these correlations are primarily the geometry of the support and location of the radiation shield intercept(s) along the support.

Steady-state heat load is predicted by equations 1 or 2. Equation 3 predicts the time history of the midpoint (warmest section) of the strut between cold-end pin and LN_2 intercept during cool-down. Equation 4 predicts the LHe heating history from the support during the terminal portion of the cool-down and equation 5 predicts the total cool-down time for the strut to reach steady-state heat rate. These correlations can take the place of the expensive, time consuming computer simulations previously required.

In addition to capabilities of determining thermal response, optimization of thermal performance through adjusting location of the LN_2 -cooled heat intercepts, LN_2 -cooled liners, and radiation shields is also discussed.

INTRODUCTION

A widely accepted method of supporting large superconducting magnets suspended in a vacuum vessel is through the use of steel cold mass supports. These supports take the form of cylindrical struts with threaded rod ends. The rod ends contain a ball and race bearing assembly that facilitates smooth motion and are pinned to triangular plate clevises which are welded to the magnet case. Figure 1 shows a typical magnet and support arrangement.

Conduction down the strut is inevitable and if left unchecked will lead to excessive capacity requirements for the helium refrigeration system. For this reason, the design includes some features to reduce the load from ambient environments. First, on the cylindrical tube that makes up the strut, there is an LN_2 intercept which is a collar with liquid nitrogen piped through it and is constrained to approximately 80K. This intercept picks up 75 percent of the conduction load. Additionally, the portion of the support between the intercept and the magnet is shielded from radiation by an active liquid-nitrogen-cooled shield also constrained to 80K. Figure 2 shows a typical steady-state temperature profile for a support strut. Radiation heat gain that would be transmitted down the hollow cylinder of the strut turnbuckle is almost entirely eliminated by interspersing a series of disc radiation shields along its length supported by a thin low-conductivity rod. The upper portion of the turnbuckle between the intercept and the vacuum vessel is passively shielded from radiation by a stainless steel foil shield supported from the strut on low thermal conduction mounts.

These are the features included in the simulation of supports of the MFTF magnet system using General Dynamics' THERMAL ANALYZER Computer Program¹. This program is used to calculate and document detail thermal parameters for the given input during transient or steady-state operation modes. A finite element model is formulated according to guidelines required by the program. A typical individual support model has approximately 70 nodes which represent the clevis and strut components, size, shape, and materials that are thermally interacting. They interact with the aid of approximately 100 resistors that replicate any mode of heat transfer, conduction, convection, or radiation and combinations of these modes. All of the significantly different magnets of the MFTF A-Cell and Axicell configurations including their respective support systems were thermally simulated for cool-down, warm-up, and steady-state operation. These analyses provide a thermal response and performance data base of approximately thirty different supports. When each of the total individual magnets was simulated, designs had not been finalized. With redesign as an ever present possibility and the data base existing from earlier simulations of both the MFTF A-Cell and Axicell configurations, it appeared to be a significant aid to develop a more austere approach to generating estimates of thermal performance rather than recalculating thermal resistances for THERMAL ANALYZER each time the support dimensions were changed.

STEADY STATE CORRELATION

One of the initial steps in developing the correlations was to determine which support design parameters would significantly influence and correlate

the heat leakage of the supports. The earliest consideration assumed distance and cross-sectional area which can be combined with thermal conductivity to form the thermal resistance expression:

$$R = \frac{x}{kA}$$

where x = distance, length dimension

A = cross-sectional area

k = thermal conductivity of material

The shield, which covers the end of the strut closest to the magnet, is constrained to 85°K. Actual design-space constraints placed the LN_2 intercepts at from 40 percent to 54 percent of the pin-to-pin length from the cold end. This is referred to as L^* , the characteristic length from magnet pin to the LN_2 intercept. Typical long clevises on these magnet cold masses in a vacuum chamber actually place all intercepts near 50 percent of the total support length from magnet LHe cooled surface to the room temperature vacuum vessel. From a practical standpoint, this is about as close to a 33 to 40 percent length from cold end considered optimum a typical refrigeration cost optimization that can be achieved.²

Another dependent variable chosen for the study is the turnbuckle diameter, D_{TB} . There is a general, direct relationship between the turnbuckle diameter and, individually, the thickness of the turnbuckle, the diameter of the end shank, clevis thicknesses, and the pin diameters. As one increases, so do the others because their size is determined by longitudinal structural load. The turnbuckle diameter variable is directly related to the cross-sectional area of the turnbuckle in the form of:

$$A = \frac{\pi}{4} (D_{TB,o}^2 - D_{TB,i}^2)$$

where

$D_{TB,o}$ = outer turnbuckle diameter

$D_{TB,i}$ = inner turnbuckle diameter

This cross-sectional area was earlier related to the conductive thermal resistance expression, $R = x/kA$.

The turnbuckle diameter, D_{TB} also has an additional factor of consideration in the analysis. As previously stated, the LN_2 shield over the cold end of the strut is constrained at 85°K. The magnet, at steady-state, operates near 5°K and, by conduction, the strut at steady-state has a temperature profile that increases from a minimum of 5°K to 85°K at the intercept to 300°K at the vacuum vessel. Because of this differential in temperatures, there is radiative heat transfer between the shield and the turnbuckle. This heat energy, Q_r , is expressed in the form

$$Q_r = A_s \epsilon \sigma (T_s^4 - T_{tb}^4)$$

where ϵ = emittance

A_s = surface area of support below intercept

T_s = LN_2 shield temperature

T_{tb} = mean turnbuckle temperature

σ = Stefan-Boltzmann constant

The surface area, A_s , is approximately given by the equation:

$$A_s = \ell \pi D_{TB}$$

where ℓ is a pseudo-length of the support from the magnet to intercept including clevises and rod end permitted by the tendency for the sizes of all these components to be proportional to D_{TB} . Thus, D_{TB} has two-fold influence in flow of heat in the cold end of the strut, i.e. radiative heat transfer related to D_{TB} to the first power and conductive heat transfer related to D_{TB} to the second power.

A third independent variable was examined as to its enhancement of correlation. The pin diameter, D_p , of the strut assembly and the contact resistances in the race/ball/pin interfaces of the rod end influence the total heat transfer of a given support design. A contact coefficient in a vacuum for these sizes of interfaces and loadings of 124 Btu/hr-ft²-°R (703 W/m²-°K) was used.³

MATC1, a statistical analysis program was used for the logarithmic regression analysis. The resulting correlation equation for steady-state heat rate dependence on D_{TB} and L^* only is:

$$Q = 6.9997 D_{TB}^{1.3774} L^{*0.9566} \quad (1)$$

and for dependence on D_{TB} , L^* , and D_p is:

$$Q = 8.3295 D_{TB}^{0.6049} L^{*0.8665} D_p^{0.7406} \quad (2)$$

L^* is distance from magnet pin to LN₂ intercept

Figure 3 shows the computer simulated calculated heat rate plotted versus the right hand power functions for the 3-parameter version of the correlation equation 2. The slope of the solid line is the correlation constant of the equation and displays the degree of correlation with the heat rate data point. The resulting correlations could be used for nominal design analysis of a support system with many struts resulting in accurate overall heat rate.

If only a few struts are being design analyzed and individual strut heat loads are therefore significant, a more conservative equation with a higher constant is desired. General experience and a review of these data recommends a constant which predicts nominal plus one standard deviation heat rates. This is shown as the dotted line in Figure 3 and is seen to under-predict less than 15 percent of the data points.

TRANSIENT PERFORMANCE CORRELATIONS

This analysis results from computer simulation data obtained in the design phase of the Mirror Fusion Test Facility-B (MFTF-B) system for Lawrence Livermore National Laboratory. The system consists of 26 magnets identified as either transition (T1/T2) coils, yin-yang (Y-Y), barrier (A1/A2), or solenoid coils, with supports totaling approximately 80 in number and labeled as either hanger, lateral, or axial supports.

By ramping down the helium coolant, which is the support fin root conductive driving temperature, in 80 hours, the magnet case cools nominally in 110 hours and the support continues to cool toward steady-state for an additional period of time predominantly dependent on the length parameters of the strut.

The transient temperature correlation equation below the intercept may be expected to be of the form of the closed form solution to the diffusion equation for a uniform rod cooled at the end surfaces⁴. Numerous adjustments to this solution are required to fit the strut data because the diffusivity varies considerably with temperature, the support is not of uniform cross-section and has a high resistance monoball/pin interface, the end surfaces of the supports below the LN₂ intercept are cooled to two different temperatures (LN₂ and LHe), and the LHe-cooled end follows a ramp-down rather than step. Furthermore, early in the first 25 hours of the transient, additional strut cooling is by radiation to the LN₂ liner. The temperature data at the midpoint of the cold end of the strut for cooling simulations are shown in Figure 4. The full range of cold end lengths is covered by four selected struts. The data are characterized by the 25-hour exponential transient followed by a more linear and slower temperature decay. A piecewise correlation of these data leads to the following equation that describes cooling of temperature of midpoint of strut between the cold-end pin and the LN₂ intercept:

$$T = \left[(T_a - T_b) e^{\frac{-16.0 \theta_2}{L^{1.30}}} + T_b \right] e^{-0.0054\theta} \quad (3)$$

$$T \geq T_b$$

for $\theta_2 = \theta$ if $\theta \leq 25$ hrs.

and $\theta_2 = 25$ hours for $\theta > 25$ hrs.

where T_a = starting temperature, 300K

T_b = steady-state temperature at midpoint of strut
between cold end pin and LN_2 intercept ($\approx 63K$ for
MFTF-B struts)

θ_2 = time in hours during rapid initial cooling

θ = total cooling time in hours

L = distance from magnet case to LN_2 intercept, cm.

The degree to which this equation fits the simulated strut response is shown by the solid lines in Figure 4. In general the equation-predicted response is slower than the detail simulation and therefore slightly conservative. This relationship can be used to approximate when the strut temperatures will reach steady-state. Because of the exponential asymptotic approach to steady state, practical definition precludes this time from exactly coinciding with the time when steady-state heat load is reached. Steady-state temperature is important from a magnet system alignment standpoint. The midpoint between the pin and LN_2 intercept is in the vicinity of the end of the rod end shank is inserted in the turnbuckle. The exception to this is in the cases of exceptionally long struts. Experience with cool-down simulations shows this zone to be the last place to stabilize to a steady-state temperature.

The equation that describes LHe heating history of strut after the time at which the magnet coolant reaches LHe temperature, i.e. for MFTF-B $\theta_3 = 80$ hrs., is

$$Q = Q_{ss} e^{0.012 L \left(\frac{\theta}{\theta_3} \right)^{-2.67}} \quad (4)$$

for L and θ as defined above,

Q_{ss} = steady-state heating (Figure 3).

This relationship is vital when considering refrigeration requirements. It may be possible to start operation before steady-state conditions are reached. This relationship will allow an estimate of the possible refrigeration penalties for such operation. Figure 5 is an actual heat rate history of a typical strut with correlated results superimposed. As can be seen in Figure 5, the correlation generally yields conservatively high transient heat rates during the terminal portion of the cool-down. This is preferable when sizing the capacity of helium refrigeration required. A slightly excessive capacity system is more desirable than one that is undersized.

The heat rate reaches steady-state, $Q = Q_{ss}$ when

$$e^{0.012L} = \left(\frac{\theta_{ss}}{\theta_3} \right)^{-2.67} \quad (5)$$

θ_{ss} $\approx$ $\frac{\text{total cooling time in hours to reach steady-state}}{\text{heat rate}}$

INTERCEPT OPTIMIZATION

LN_2 -cooled intercept locations were originally optimized based on minimizing total refrigeration costs of the cryogens LN_2 and LHe. This places the intercept 33 to 40 percent of the total support distance from the magnet case to the vacuum vessel. For long supports this does not optimize cool-down time. Two relationships were developed that would establish time to reach

steady-state refrigeration conditions based on distance from the cold-end pin to the intercept. The relationship of time to reach steady-state versus length below the intercept is

$$\theta_{s1} - \theta_4 = 0.064 L^*{}^{1.40} \quad (6)$$

where θ_4 = cooling time in hours when the inlet coolant reaches LHe temperature (80 hours for MFTF-B).

θ_{s1} = cooling time in hours to steady state for lower part of strut

For the section of strut from the intercept to the vacuum vessel, the following relationship was developed:

$$\theta_{su} = 0.0046 (L_T - L^*)^{1.92} \quad (7)$$

where θ_{su} = total cooling time in hours to steady state for upper part of strut

L^* = length from cold-end pin to intercept, cm

L_T = pin-to-pin length of strut, cm

These relationships provide a means for determining the value of changing the location of the intercept along the strut to improve cooling response. It has been observed that both sections of the longest struts may not reach steady-state temperature conditions simultaneously. These equations will allow the user to position the intercepts either to have both sections reach steady state together and thereby minimize cooling time and alignment problems, or have the lower end reach steady state with regard to refrigeration requirements and disregard the contraction still possibly going on in the upper strut section. Figure 6 indicates how a strut's temperature profile can be altered by relocation of an intercept plus adding an extra intercept.

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Figure 1. Typical Support Strut Thermal Control Features

Figure 2. Typical Steady-State Temperature Profile of Cold Mass Support Strut

Figure 3. Correlation and Design Heat Rate Equation
with Three Strut Parameters

Figure 4. Selected Curve Fits vs. Data Points of Strut Cold-End Midpoint Temperature Histories

HEAT RATE FROM HANGAR(01) WARM/COLD STRUT
DURING COOLDOWN WITH 31.2 G/SEC TOTAL HE FLOW

Figure 5. T2 Hanger Heat Rate History Plus Projection
Using Correlation

Figure 6. Temperature Profile of Longest T2 Support Strut
As-Modeled and with Two LN₂ Intercepts