1 Thermal Study of a Cryogen-less MgB₂ Cavity

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5 Abstract

6 Recent efforts towards production of high-quality magnesium diboride
7 (MgB₂) coatings have raised the possibility of producing usable accelerating
8 cavities. Work continues to reliably produce films of sufficient quality over
9 the large, complex surface area of an accelerating cavity, but this technology
10 would open many interesting technical opportunities. One of these is to re-
11 place the traditionally required liquid helium cryogenic systems with a dry
12 system based on cryocoolers. This is made possible by the much higher T_c of
13 MgB₂, allowing operation closer to 30 K where cryocooler efficiency becomes
14 competitive with alternative systems. This removes the need for pressure
15 vessels in the cryomodule as well as internal distribution systems, greatly
16 simplifying cryomodule design and fabrication. The lack of uniform cooling
17 over the cavity surface, however, complicates behavior by coupling RF losses,
18 heat leak, and cooling design in a way not seen in traditional SRF cavities.
19 In this paper, these complexities are explored, including realistic cryocooler
20 performance, temperature dependant RF losses, and standard thermal man-
21 agement challenges.

22 *Keywords:* Superconducting RF, Magnesium Diboride

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24 1. Introduction

25 The ultimate goal of MgB₂ research is the creation of large surface area
26 films of very high quality and uniformity. As this work progresses towards
27 reality, it has sparked interest in novel applications for MgB₂. One of these
28 applications is to use a MgB₂ coated RF cavity with cryocoolers instead of the
29 more traditional cryogenics plant. The much higher transition temperature
30 for MgB₂ means that this cavity could be operated in a high temperature

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31 range to take advantage of a cryocooler's increased efficiency over competing
 32 systems. This would also allow the cooling system to be designed dry, without
 33 any liquid cryogens, greatly simplifying the overall design and completely
 34 removing the need for pressure vessel certification. The goal of this study is
 35 to begin to explore the relationship between the film quality, cavity material,
 36 and cryocooler-based cooling system in this new application.

37 2. Magnesium Diboride

38 The surface resistance of a superconducting material is generally given in
 39 two components: R_{res} and R_s^{BCS} . The first term is the Residual Resistance [1]
 40 and is thought to be dominated by trapped magnetic flux and the quality
 41 of the RF surface itself (purity, grain boundaries, etc). While there is a
 42 significant amount of MgB₂ research ongoing, most films produced to date
 43 have a significant R_{res} , generally dominating performance. The work in this
 44 paper will consider both the state of the art as it currently exists (R_{res} on
 45 the order of 1 $\mu\Omega$), and more ideal performance expected once the coating
 46 techniques are perfected.

47 The second term is given by BCS theory [2] and takes the form:

$$R_s^{BCS}(n\Omega) = \frac{1}{T} 10^5 f_{GHz}^2 e^{-\Delta_o/kT} \quad (1)$$

48 Where Δ_o is the superconducting gap energy. Niobium has a transition
 49 temperature of 9.25 K and gap energy of 1.55 meV, giving a BCS surface
 50 resistance of 3.4 $n\Omega$ at 350 MHz and 4 K. MgB₂ is a two-gap superconductor,
 51 with theoretical gaps at 2.7 meV and 6.7 meV [3] and a much higher transition
 52 temperature of 39 K. These two gaps give surface resistances of 0.12 and
 53 $1.1 \times 10^{-6} n\Omega$ (at 350 MHz and 4 K), and it is not clear as of yet which gap
 54 will dominate actual performance. While there is some theory about this
 55 issue [4, 5], this analysis used the conservative, lower gap energy with the
 56 BCS model.

57 3. Simulation Methodology

58 All of the presented simulation was done in ANSYS-APDL [6]. The RF
 59 volume and cavity solid volume are created as separate entities and imported
 60 into ANSYS. First, the RF volume is meshed and used to simulate the accel-
 61 erating eigenmode. The mode parameters are then calculated and normalized

62 to the desired accelerating voltage. The cavity solid volume is then meshed
63 and both volumes are given appropriate material properties. The RF heating
64 is applied to the inner surface of the cavity solid volume.

65 In addition to the RF heating distribution, a heat load is assigned to each
66 beam pipe to simulate a reasonable thermal leakage from the beamline warm
67 to cold transition, alignment fixturing, and tuning mechanism.

68 In this simulation, radiation heat loads are not considered because it is
69 assumed that a cold radiation shield would be employed, tied to the first stage
70 of any cryocoolers used. Given the experience with the Superconducting
71 Undulator (SCU) at the Advanced Photon Source [7] and the heat load maps
72 provided by the manufacturers of the cryocoolers used there [8], it is expected
73 that a single shield tied to the first stage would maintain a temperature
74 below 70 K. Assuming a reasonable multi-layer insulation blanket is used,
75 the radiative heat load should be negligible.

76 Finally, a fixed temperature boundary condition is applied on whatever
77 thermal flags are on the cavity. The cavity is assumed to be cooled through
78 high-quality thermal links connected to a cryocooler. Using the load maps
79 of the cryocooler, estimated properties of low ΔT thermal links designed
80 to operate at 20 K, and achieved thermal contact impedances in the SCU, a
81 correlation between applied thermal load and cavity thermal flag temperature
82 has been derived. While the temperature of the flag is calculated based on
83 heat load per flag, the initial conditions for both the flag temperature and
84 cavity RF surface resistance are calculated assuming a starting temperature
85 of 4 K.

86 With these boundary conditions set, the model is solved for the static
87 thermal distribution. The resulting temperature shift is generally enough
88 to violate the constant surface resistance assumption, even at low power,
89 and further convergence is required. This is achieved by using the cavity
90 temperature to recalculate the BCS surface resistance element by element
91 and reapply the new (higher) heat load. After updating the RF heating
92 distribution, the new total heat load is calculated and used to determine the
93 new thermal flag temperature. With these new boundary conditions, the
94 static thermal distribution is resolved. This process is repeated until the
95 thermal distribution converges (defined by less than a 0.5% shift in peak
96 temperature solve-to-solve). It should be noted that the mathematical form
97 of the BCS surface resistance means that this method *always* converges, but
98 any peak temperature over the quench limit of 39 K means that a real cavity
99 would undergo thermal run-away.

3.1. Material Properties

Because the losses are associated with the RF volume elements, the elements of the RF volume cannot be suppressed for the thermal simulation. This meant that the RF volume must be given some material properties such that it does not affect the simulation. Studies were done to demonstrate that setting the RF volume thermal conductivity to at least five orders of magnitude below the cavity solid volume conductivity ensures that this approximation will be $< 0.1\%$ perturbation on the resulting thermal distribution (see Figure 1). The vacuum thermal conductivity was set to five orders of magnitude lower than the poorest thermal conductivity to be used, that of niobium at 6 K. This ensured that the contribution from the vacuum materials could always be neglected.

Figure 1: Change in cavity thermal distribution (peak and minimum temperatures) for different ratios of cavity wall thermal conductivity to vacuum space thermal conductivity for a representative cavity.

The metallic cavity walls were simulated with either pure copper or niobium. While copper is clearly a superior option due to its much higher thermal conductivity, it may be necessary to use a different material with an inferior thermal conductivity to aid in MgB_2 film formation. Niobium is one potential option for this, and should be well representative of a low-thermal conductivity substrate. The temperature dependant material properties for both oxygen-free, high-conductivity copper and niobium ($RRR = 60$) were

119 taken from CryoComp [9], which relies on the NIST database. Low-RRR
 120 niobium was chosen to represent a “worst-case” thermal scenario.

121 3.2. Cavity Performance Limitation

122 While MgB_2 has a transition temperature of 39 K, the effects of the tem-
 123 perature dependence of the critical magnetic field are less well understood.
 124 While some literature reports $H_c = 4290 \text{ Oe}$ [10], achievable performance
 125 will depend heavily on the actual film quality. To avoid this complication
 126 and to provide a realistic engineering performance margin, it was decided to
 127 set the upper limit of performance at 30 K. This meant that any simulation
 128 that converges to a peak temperature of 30 K or over on the cavity cell is
 129 considered to have quenched.

130 4. Thermal Management Approximation

131 The cryocooler chosen was the Sumitomo Cryogenics RDK-408S2 (60 Hz).
 132 This cryocooler model is designed to operate in the appropriate temperature
 133 range for MgB_2 (6-30 K) and has been used with success for the SCU. The use
 134 of cryocoolers means that the cavity temperature has some dependence on
 135 total heat load. A simple model of this dependence was done by assuming a
 136 reasonable thermal management strategy of cryocoolers, thermal buses, and
 137 low ΔT thermal links (see Figure 2).

138 Given this scheme, there are three major components that need to be
 139 accounted for: cryocooler load map, thermal link temperature drop, and
 140 contact impedance between sections.

141 From the load mapping data given by the supplier, a first order approxi-
 142 mation gives a temperature dependant thermal capacity of the second stage
 143 of

$$P_c = 1.624 * T - 10.26 \quad (2)$$

144 with the first stage under no loading. The first stage of the cryocoolers
 145 was assumed to be used for all thermal intercepts such as thermal shield
 146 and beamline warm-to-cold transitions. Fortunately, the second stage per-
 147 formance is a relatively weak function of first stage loading at the levels
 148 expected (between 40-80 Watts at 50 K [11]).

149 Thermal bus bars were considered to be made of copper, with a cross-
 150 section large enough to ensure small thermal drops across their length. Tem-
 151 perature drop across components was calculated from either bus bar cross

Figure 2: Example of thermal management scheme chosen. Flexible thermal links connect directly to the cavity equator at two points and conduct the heat out to a thick copper bus bar. This bus bar is directly cooled by the cryocooler second stage. The bus bar is sized to minimize the temperature drop across the cavity.

152 section or specified components from the SCU design. For instance, the flex-
 153 ible race-track shaped thermal connections are modeled from an equivalent
 154 designed for the SCU to operate at 20 K, with a rated temperature drop of
 155 15.25 W/K .

156 Using thermal data from the SCU0 cryostat that was installed in the
 157 APS ring in January 2013, an achieved contact conductance of roughly 10
 158 $\text{kW/m}^2\text{K}$ was calculated for several joints. While there was special attention
 159 paid to these joints to ensure excellent thermal contact, the methods used
 160 were not extreme, and it is expected that this performance could be reliably
 161 duplicated. The joints used for these calculations were chosen because they
 162 operated at similar temperatures (20 K) and heat loads (10s of Watts) to the
 163 expected environment to be simulated.

164 Cavity thermal flags were designed with these careful thermal connection
 165 procedures in mind. Especially important would be the use of flat connections
 166 which do not traditionally exist on an elliptical cavity. Especially if niobium

167 sheet is the fabrication material for the cavity, it would be most straight-
 168 forward to mill flats into the cavity equator as shown in Figure 3.

Figure 3: Section view of the cavity wall with a thermal flag milled into the cavity equator (at top). The depth of this cut would be as large as possible while still ensuring adequate cavity thickness for processing and mechanical stiffness.

169 The surface area of the thermal flag is clearly limited by the thickness
 170 of the cavity wall, but is on the same order as the surface areas available in
 171 the rest of the thermal path such as thermal straps and the footprint of the
 172 cryocooler second stage head.

173 As a baseline design, the cavity simulated would have two thermal flags
 174 180 degrees apart on each cell's equator, and these thermal straps would
 175 connect to a bus bar. The copper bus bar would, in turn, be connected to
 176 a cryocooler through another thermal link. This design is relatively simple
 177 to implement without special design exercises. The thermal flag power
 178 dependence of this design is

$$T_{flag}(K) = 0.733 * Q_{tot,W} + 6.305. \quad (3)$$

179 It should be noted that this approximation is less accurate in the very low
 180 heat load limit because it was assumed that the cavity would be run close
 181 to the limit of performance, i.e. closer to 30 K. This assumption determined
 182 what part of the cryocooler load map was approximated, and the cryocooler

Table 1: Electromagnetic parameters of the cavity geometry used for these simulations. Parameters calculated assuming $\beta = 1$.

Cavity Parameter	Units	
Frequency	500	MHz
Geometry Factor	244	Ohms
R/Q	99.2	Ohms
$V_{acc}/\sqrt{U}$	0.558	MV/ $\sqrt{J}$
$E_{pk}/\sqrt{U}$	4.05	MV/m $\sqrt{J}$
$B_{pk}/\sqrt{U}$	9.94	mT/ $\sqrt{J}$
TTF	0.52	-

183 behavior changes slightly as you reach this model's ultimate temperature
184 of approximately 6 K. The resulting relationship also depends on the exact
185 thermal management scheme chosen. This model is not to be considered
186 an optimal design, but one that represents a reasonable approximation of a
187 final, realistic design.

188 4.1. Cavity Design

189 For this study, it was desired to have a cavity with a high frequency to
190 minimize surface area required for coating. Conversely, it was also desired
191 to have a low frequency cavity to minimize the BCS contribution to R_s . For
192 obvious practical reasons related to potential coating techniques, complex
193 geometries like half-wave cavities, quarter-wave cavities, and spoke cavities
194 were avoided. This lead to the selection of a 500 MHz elliptical cavity similar
195 to the one used for BEPC-II [12]. This design is quite close in parameter to
196 the traditional CESR and KEKB cavities, and seems to provide a balanced
197 solution to the needs of this study. Besides the removal of the end groups
198 for simplicity, no other changes were made to the cavity design. The cavity
199 parameters as simulated can be seen in Table 1.

200 5. Ideal Cooling Analytic Model

201 The non-uniformity of cooling presents the largest amount of complex-
202 ity, requiring simulation. If the cavity can be approximated as uniform in
203 temperature, either because of infinite conductivity in the wall material or

204 use of some non-existent cooling liquid, then the cavity performance becomes
 205 analytically solvable. This can also act as an excellent ideal case to judge
 206 the design of any cooling scheme.

207 Uniform cooling is modeled by assuming a known dissipated power and
 208 using that to calculate the resulting temperature of the thermal flags. This
 209 temperature is used to calculate R_s^{BCS} , which gives the cavity quality factor
 210 and stored energy. From here, the other field parameters can be calculated.
 211 As can be seen, this method presumes a cooling design and will be dependant
 212 on cavity design. Other realistic parameters like thermal leakage can be
 213 included easily.

214 The baseline performance of such a cavity can be seen in Figure 4. The
 215 plot of Q_0 vs. V_{acc} has many features that we will see repeated in further
 216 simulations: An initially high Q_0 with a sharp Q-drop at mid field gradually
 217 leading to a thermally induced quench. The mid field Q-drop is associated
 218 with a sudden increase in the rate of cavity heating. This change in slope of
 219 $T(V_{acc})$ remains relatively constant all the way out to cavity quench.

Figure 4: Cavity quality factor and temperature vs. accelerating voltage for a cavity with uniform cooling, no residual resistance, and no thermal leakage.

220 In the remainder of this section, we will use the uniform cooling model
 221 to investigate the effects of thermal leakage and residual resistance before
 222 moving to full thermal simulations.

223 5.1. Leakage Current

224 With any cryogenic system, the static heat leak is a major design con-
 225 cern and the subject of significant engineering effort. In a more traditional
 226 cryogenic system, this leakage is seen as an additional load on the liquid
 227 refrigerator, and can often be taken within the margin of operation there
 228 without diminished performance. In a system using cryocoolers, this leakage
 229 means a direct increase in temperature to accommodate the increased heat
 230 load. Furthermore, in a dry system such as the one being studied, the exact
 231 heat leak distribution has the potential to further reduce performance by
 232 changing the temperature distribution.

233 In this study, radiation heating is neglected because a high quality thermal
 234 shield with well designed multi-layer insulation is a reasonable assumption
 235 for a design focused around thermal management. This means that the heat
 236 leak though the beamline warm-to-cold transitions, tuner, and alignment
 237 features are the major concerns. Fortunately, these can all be approximated
 238 together with a heat flux on the cavity beampipes.

239 To study this with the model, the heat leak per beampipe (the sum of the
 240 contributions from tuner, alignment, and warm-to-cold transition) is doubled
 241 and added to the RF heat load. This load is then used to calculate cavity
 242 performance with the same procedure mentioned above.

243 Because the dissipated power is relatively large at high field (10s of
 244 Watts), the addition of a leakage current, even up to 1 Watt, has little effect
 245 at high fields. The drop in performance is most noticeable in the low-field Q
 246 where it is a larger proportional heat leak. As can be seen in Figure 5, the
 247 drop in performance at low-field is only significant in the most optimistic of
 248 cases: those with very low residual resistance.

249 At high field, it is simple to show that a fixed quench temperature gives
 250 a straightforward relationship between leakage current and ultimate voltage.
 251 Choosing a quench temperature of 30 K gives

$$V_{acc,30K} = \frac{V_{acc}}{\sqrt{U}} \sqrt{\frac{G}{R_s \omega}} \sqrt{32.326 - P_{leak}} \quad (4)$$

$$V_{acc,30K} \cong V_{acc,30K,P_{leak}=0} \left(1 - \frac{1}{2} \frac{P_{leak}}{32.326} \right) \quad (5)$$

$$\frac{dV_{acc}}{dP_{leak}} = -\frac{1}{64.326} V_{acc,30K,P_{leak}=0}. \quad (6)$$

Figure 5: Effect of increased thermal leakage on low-field quality factor for three different residual resistance cases. Calculated using an analytical model with ideal cavity cooling.

252 The small correlation between leakage current and ultimate voltage is fur-
 253 ther suppressed by less than ideal cavity performance which limits $V_{acc,30K,P_{leak}=0}$.
 254 Ultimately, this means that thermal leakage is less of an issue than it is with
 255 more traditional helium cryogenic systems operating at 2 K or 4.2 K. These
 256 systems are significantly less efficient because of their lower operating tem-
 257 perature, and have a smaller total capacity relative to the leakage currents
 258 achievable.

259 This analytical model does not take into account non-linear effects caused
 260 by changes to the cavity thermal distribution because of the thermal leakage,
 261 which will be studied in a later section.

262 5.2. Residual Resistance

263 Achievable residual resistance is a substantial unknown in the production
 264 of large scale MgB₂. Existing films have been measured with R_{res} on the
 265 order of $\mu\Omega$. Using these measurements to predict ultimate performance
 266 is extremely challenging because the temperature, frequency, and field level
 267 dependence of the residual resistance is essentially unknown. Given this
 268 uncertainty, the most simple model of R_{res} will be used; simply a constant
 269 addition to the total R_s with no temperature or field level dependence. It
 270 should be stated that this is not a strictly accurate approximation, even in

271 modern SRF bulk niobium cavities [13]; it is simply the best model available
 272 given the lack of experience with MgB₂.

273 The effects of adding a significant residual resistance are evident in Fig-
 274 ure 5. The three curves in this plot represent the addition of 100 nΩ and 500
 275 nΩ to the ideal cavity at low field. Here the drop in Q is magnified because
 276 of the low operating temperature, and thus low R_s^{BCS} contribution (~28 nΩ).

Figure 6: Cavity high field Q_0 and V_{acc} at 30K peak cavity temperature for a range of residual resistances up to 1 $\mu\Omega$. Calculated using an analytical model with ideal cavity cooling.

277 In Figure 6 the effects of the residual resistance on the cavity's high field
 278 performance is less severe, but still profound. The drop in both achievable
 279 V_{acc} and Q_0 can be seen, but at this higher field level, the R_s^{BCS} contribution
 280 is ~300 nΩ, fractionally reducing the effect of the residual resistance.

281 6. Non-linear Thermal Simulations

282 The analytical model used in the previous section assumed a uniform
 283 cooling method for the entire cavity body. This is a best-case scenario as
 284 any non-uniformities in the temperature distribution will increase the RF
 285 losses, diminishing cavity performance.

286 The simulated models will consist of two materials: OFHC copper and
 287 RRR = 60 niobium. Copper and niobium were chosen to represent a realistic
 288 best and worst case for the thermal conductivity. These materials will be

289 simulated under different cooling schemes called two-flags, four-flags, and
 290 uniform cooling. Two and four flags are realistic cooling methods that call for
 291 flats to be milled on the cavity equator 180 or 90 degrees apart and thermally
 292 sunk to a cryocooler. Bath cooling is an unrealistic but instructive method
 293 where the entire cavity surface is kept uniform by a moderating liquid. Bath
 294 cooling will also be the best comparison to the analytical model.

295 First, the cooling schemes and materials will be simulated and compared
 296 to the “ideal cavity” used in the analytical model (0.2 total heat leak, R_{res}
 297 $= 0$) to determine the effects of the thermal distribution on the cavity per-
 298 formance at all field levels. Second, the non-linear effects of the thermal
 299 distribution on the thermal leakage will be simulated and compared to ear-
 300 lier analytical results.

301 6.1. Two-Flag Cooling Scheme

302 Quality factor vs. accelerating field was simulated for both copper and
 303 niobium cavities using the cooling scheme modeled in Figure 2. The difference
 304 in cavity performance seen in Figure 7 is stark.

Figure 7: Cavity quality factor vs. accelerating voltage up to a peak temperature of 30 K cavity temperature for two models cooled by two thermal flags set 180 degrees apart on the cavity equator. This is compared to the analytical model which assumes ideal cooling. For all models shown, a realistic thermal heat leak of 0.1 Watts per beam pipe was included with no residual resistance.

305 Compared to theory, the copper cavity performs very well, having a
 306 slightly lower quality factor and an ultimate field of 1.59 MV versus 1.64
 307 MV for the ideal cavity. The reason for this is evident when looking at the
 308 thermal distribution from the simulation in Figure 8. The thermal conduc-
 309 tivity of the 3 mm copper cavity wall is large enough that the peak cavity
 310 temperature is less than a degree over the flag temperature.

Figure 8: Converged thermal simulation of a copper cavity cooled by two flags at ultimate field (note the 30 K peak temperature). One eighth of the cavity is simulated with only the cavity material volume shown (RF volume is suppressed). The cooling flag is located at the top of the cavity cell and is the location of the minimum temperature.

311 While the qualitative shape of the quality factor curve is the same, the
 312 niobium cavity has a much earlier onset of Q-drop and lower ultimate field.
 313 Contrasting Figure 8 with Figure 9, we can see that the lower thermal con-
 314 ductivity has a significant impact on cavity performance. This simulation,
 315 including the non-linear behavior of thermal conductivity and surface resis-
 316 tance, gives a peak cavity temperature almost 10 K over the thermal flag
 317 temperature. This temperature distribution also limits the cavity ultimate
 318 field quite seriously (1.23 MV versus 1.64 MV).

319 6.2. Improved Cooling for Niobium Cavities

320 Two simulations were done to quantify the relationship between the cool-
 321 ing management scheme and a niobium cavity's performance. Clearly the two

Figure 9: Converged thermal simulation of a niobium cavity cooled by two flags near ultimate field (note the 30 K peak temperature). One eighth of the cavity is simulated with only the cavity material volume shown (RF volume is suppressed). The cooling flag is located at the top of the cavity cell and is the location of the minimum temperature.

flag cooling scheme produced seriously degraded cavity performance. First, to justify the simulation procedure, an ideally cooled niobium cavity was simulated. This cavity is uniformly cooled on the cavity outer surface as if by liquid with the liquid temperature determined by the same cryocooler performance as the previous cavities. This should result in cavity performance very similar to the theoretical model, even if it is unrealistic.

The resulting thermal distribution can be seen in Figure 10. The cavity is now a uniform temperature with the peak temperature on the beam pipe. It should be noted that the ultimate field was defined as 30 K on the cavity cell, which is not the peak temperature in the simulation in this case.

Given the thermal distribution seen in the two flag simulation, a four flag cooling scheme was simulated. This has four cooling flags 90 degrees apart on the cavity equator (see Figure 11) to provide a more uniform cooling distribution while still being realistically achievable.

The use of four cooling flags on a cavity means that the cooling distribution is no longer symmetric with only one cryocooler. In order to keep this simulation fairly comparable to the other simulations, the relationship between cryocooler and cavity flag temperature was recalculated to accurately

Figure 10: Converged thermal simulation of a niobium cavity cooled uniformly over the entire cavity outer surface at ultimate field (note 30 K temperature on cavity cell). One eighth of the cavity is simulated with only the cavity material volume shown (RF volume is suppressed).

Figure 11: Example of a more aggressive thermal management scheme. Flexible thermal links connect directly to the cavity equator at four points and conduct the heat out to a thick copper bus bar. This bus bar is directly cooled by the cryocooler second stage. The bus bar is sized to minimize the temperature drop across the cavity.

340 reflect the thermal flux per flag. In addition to this, the copper bus bar has
 341 been resized to better approximate even cooling on each flag despite the dif-
 342 ferent thermal path lengths. While this makes a small change to the applied
 343 cooling ($T_{flag} = 0.7 * Q + 6.305$ instead of $T_{flag} = 0.733 * Q + 6.305$), this
 344 change is considered minor compared to the change of the cavity thermal
 345 distribution.

346 The resulting cavity thermal distribution can be seen in Figure 12. The
 347 cavity delta-T is now reduced to just over 5 K from the original 10 K, not
 348 quite half, as expected because of the non-linear material properties.

Figure 12: Converged thermal simulation of a niobium cavity cooled by two flags near ultimate field (note the 30 K peak temperature on cavity cell). One eighth of the cavity is simulated with only the cavity material volume shown (RF volume is suppressed). The cooling flags are located 90 degrees apart on the cavity cell equator and are the locations of the minimum temperature.

349 The performance of the different niobium cavities are shown together in
 350 Figure 13. From this it is clear that performance of a cavity made of a poor
 351 thermal conductor will suffer, but can be at least partially recovered with
 352 careful thermal management design.

Figure 13: Cavity quality factor vs. accelerating voltage up to a peak temperature of 30 K on the cavity cell for cavities cooled by three different cooling schemes and compared to an analytical model with ideal cooling. For these calculations, a realistic thermal heat leak of 0.1 Watts per beam pipe was included with no residual resistance.

353 6.3. Simulated Thermal Leakage

354 The theoretical calculation gave a relationship between ultimate field and
 355 thermal leakage of

$$\frac{dV_{acc}}{dP_{leak}} = -\frac{1}{64.326} V_{acc,30K,P_{leak}=0}. \quad (7)$$

356 Simulation of this parameter on a niobium cavity cooled by the most
 357 conservative method (two-flags) gives a measured coefficient of 1/36.47 as
 358 opposed to 1/64.326. This means that the achievable ultimate field of a cavity
 359 with a poor cooling scheme is almost twice as sensitive as a cavity with either
 360 higher thermal conductivity or more uniform cooling scheme. The simulation
 361 method used is very simplified, but this reinforces the message that careful
 362 attention to thermal management and interception of thermal leakage will
 363 be critical to the performance of a cryogen-less superconducting cavity.

Table 2: A comparison of cavity ultimate performance for different materials and cooling schemes.

Cavity Variation	Ultimate Voltage [MV]
Theoretical	1.624
Niobium Uniformly Cooled	1.624
Copper Two-Flag	1.594
Niobium Four-Flag	1.47
Niobium Two-Flag	1.24

364 7. Conclusions

365 The goal of this study was to better understand the effects of the rel-
 366 evant complexities (temperature dependant surface resistance, cryocooler
 367 heat loading, non-linear thermal conductivities, thermal leakage) of a dry
 368 cryogenic system for a superconducting cavity. The variation in ultimate
 369 performance can be seen in Table 2.

370 From the distribution of ultimate voltages, it is clear that the best per-
 371 formance will be achieved if the MgB_2 can be formed on a material with high
 372 thermal conductivity like copper. Additionally, a well-designed thermal at-
 373 tachment between the cavity and cryocooler improves ultimate performance
 374 and reduces sensitivity to outside thermal leakage.

375 The major assumption in all of this is the formation of a high-quality,
 376 uniform MgB_2 film with low residual resistance. Residual resistance was
 377 not considered for most of this study because there are no good models for
 378 its temperature or field dependence. If, as existing samples have shown,
 379 the cavity has a significant residual resistance, it will dampen cavity quality
 380 factor and ultimate field in relative proportion to the BCS surface resistance.
 381 Until more research has been done, the exact nature of the BCS behavior of
 382 MgB_2 and non-linearities of the residual resistance must be approximated as
 383 they have here.

384 Taken all together, this study gives evidence that operation of a MgB_2
 385 cavity in a dry, cryogen-less mode should be feasible. Even at high tem-
 386 perature, the load on the cryocooler seems to fit well within the existing
 387 design limitations. One potential issue is cooldown/recovery time. Standard
 388 cryomodules are designed with a less-efficient cooldown mode to rapidly re-
 389 move the stored enthalpy of the cold mass before switching to the standard,

efficient operational mode. No such mode obviously presents itself with a cryocooler-based system. While the cooldown of such a system would be relatively uneventful, it may take on the order of days to cool a full cryomodule cold mass. Additionally, this study was done with static thermal simulations. The system's reaction to transients such as a quench would have to be simulated to better understand effects like transient heating and recovery time.

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