HELIUM PRESSURES IN RHIC VACUUM CRYOSTATS AND RELIEF VALVE REQUIREMENTS FROM MAGNET COOLING LINE FAILURE

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Abstract

A catastrophic failure of the RHIC magnet cooling lines, similar to the LHC superconducting bus failure incident, would pressurize the insulating vacuum in the magnet and transfer line cryostats. Insufficient relief valves on the cryostats could cause a structural failure. A SINDA/FLUINT® model, which simulated the 4.5K/ 4 atm helium flowing through the magnet cooling system distribution lines, then through a line break into the vacuum cryostat and discharging via the reliefs into the RHIC tunnel, had been developed to calculate the helium pressure inside the cryostat. Arc flash energy deposition and heat load from the ambient temperature cryostat surfaces were included in the simulations. Three typical areas: the sextant arc, the Triplet/DX/D0 magnets, and the injection area, had been analyzed. Existing relief valve sizes were reviewed to make sure that the maximum stresses, caused by the calculated maximum pressures inside the cryostats, did not exceed the allowable stresses, based on the ASME Code B31.3 and ANSYS results.

INTRODUCTION

RHIC consists of twelve sextants in two rings [1]. Around 1740 magnets are populated along the two 3.8km circumference RHIC rings. Most of them are superconducting magnets, which are cooled by the 4.5K/4 atm liquid helium inside the cryostat (the insulating vacuum tank). There are two relief systems: one is for the pressure vessel containing cold helium and the other is for the insulation vacuum tank. Relief valves for the high pressure vessel are installed on the Valve Boxes in the service building. Low pressure vacuum tank reliefs, normally set at 0.2 to 0.3 atm, are installed along the magnet cryostats inside the tunnel to protect the tanks from over pressure. A catastrophic failure of the RHIC magnet cooling lines, similar to the LHC superconducting bus failure incident, would pressurize the insulating vacuum in the magnet and transfer line cryostats. Insufficient relief valves on the cryostats could cause a structural failure [2].

The maximum pressure inside the vacuum cryostat, due to the magnet cooling line failure, would depend on the size of the insulating vacuum tank and the total relief cross section areas on it. In the previous system and structural analysis, a pressure, equal to the relief setting, had been assumed [3] as the maximum pressure inside the vacuum tank, which could be under estimated.

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In this paper, a complete thermal/fluid model, developed in SINDA/FLUINT® (S/F) [4], was used to simulate the 4.5K/ 4 atm liquid helium flowing through a line break into the insulating vacuum volumes and the magnet cooling system distribution lines, then through discharging via the reliefs into the RHIC tunnel. The model included as many details as practical. Energy deposition due to the arc flash was applied to the helium flow. Heat load from the ambient was calculated based on the forced convection of the helium gas inside the tank and the heat conduction through the wall thickness. The calculated pressures inside and temperatures on the cryostats were used to review the safety of the existing cryostat tanks and the sufficiency of the relief valves, based on the ASME Code B31.3 [5] and ANSYS® [6] analyses. Three typical areas (see Fig. 1), including the sextant arc, the Triplet/DX/D0 magnet, and the injection area, with seven different sized insulating volumes had been analyzed and shown below.

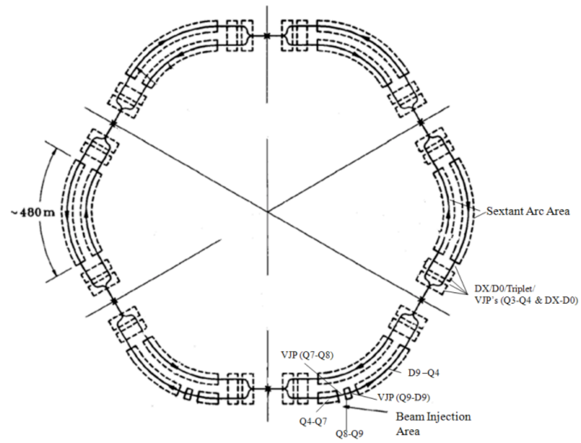

Figure 1: RHIC ring and seven typical areas.

RHIC CRYOSTAT CONFIGURATIONS

The sizes of the cryostat tanks and the relief valves are shown in Table 1. The magnet and the transfer line cryostats are made of carbon steel (SA-53 E/B) and stainless steel 304L (SA-358) respectively. Magnets inside the cryostat are alternating with dipole and quadrupole magnet along the length, with two supporting stands underneath every magnet's cryostat. One of the two dipole magnet supporting stands is a sliding support. There are two 24" (610 mm) ID bellows on both ends of every dipole magnet cryostat, a 20" (508 mm) ID bellow is installed on every 20" (508 mm) OD VJP, and a 14" (356 mm) ID bellow is on the VJP from DX to D0. Supports on the 20" (508 mm) OD VJP's are sliding supports, which allow thermal deformations on the pipes.

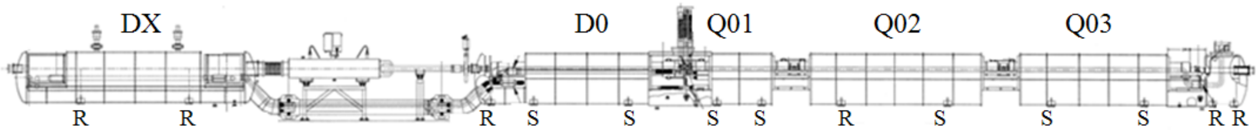

Figure 3: DX/D0/Triplet magnet connections.

Table 1: Sizes of Cryostat Tanks and Relief Valves

	RV Area[m^2]	CT wall thk.[mm]	CT Len [m]	CT OD [m]
Sext. Arc	.106	6	485	.61
Triplet/D0	.0157	4.8	18.6	1.22
DX		4.8	5.7	1.22
+VJP Q3-Q4		5.5	102	.51
+VJP DX-D0		7	6.2	.35
INJ Q8-Q9	.0035	6	18	.61
INJ Q4-Q7	.0142	6	39	.61
INJ D9-Q4	.0885	6	443	.61
VJP Q7-Q8	.0026	5.5	22	.51
VJP Q9-D9	.0026	5.5	11	.51

Note: CT= Cryostat Tank; RV= Relief Valve

A typical connection between the magnet and the transfer line cryostat (VJP) in the sextant region is shown Fig. 2. There is a vacuum break in-between the two vacuum chambers. Supporting stand on the magnet cryostat side is anchored to the floor.

Figure 2: A typical connection between magnet cryostat and transfer line VJP

Fig. 3 shows the DX/D0/Triplet magnet cryostats. Letter 'R' and 'S', on the side the supporting stands, represent the rigid and the sliding support, based on the present setup.

THE MODELS

Due to the large system size, the following five simplified models were used to calculate the maximum stresses on the cryostats at the three typical areas in the tunnel:

(1) Model for the dipole magnet cryostat: A 11.9 m long steel pipe with a rigid and a sliding support underneath (about 6 m apart).

(2) Model for the CQS magnet cryostat: A 2.64 m long steel pipe with two rigid supports underneath (about 2.54 m apart).

(3) Model for the pipe connection between the magnet and transfer line cryostat, including a vacuum break (see Fig. 2).

(4) Model for the DX cryostat assembly, including the DX-D0 VJP (see Fig. 3).

(5) Model for the Triplet/D0 cryostat assembly, including the DX-D0 VJP (see Fig. 3).

LOADS ON THE CRYOSTATS

The sustained loads on the cryostats are weight and one atm external pressure. Loads, due to the cooling line failure, are the pressures and the temperatures on the cryostat chambers, which were calculated by simulating the complete thermal/fluid system, using the developed SINDA/FLUINT® model in Ref. [4]. Results, including the minimum temperatures on the cryostats, the maximum temperature differences between the cryostats' outer and inner walls, and the maximum pressures inside the cryostats, are summarized in Table 2.

Table 2: Loads on the Cryostats due to Cooling Line Failure

	Max. Internal Pressure, Psig*	Min. Wall Temp., K	Max. Inner & Outer Wall Temp. Diff., K
Sextant Arc	4.4	291	1.07
Triplet/D0/DX +Q3-Q4 VJP +DX-D0 VJP	5.8	278	1.48
INJ Q8-Q9	37	228	5.41
INJ D9-Q4	3.4	292	0.3
INJ Q4-Q7	7.8	274	1.74
VJP Q7-Q8	33	223	5.69
VJP Q9-D9	35	180	6.94

*: 1 psi = 6895 Pa

ANALYSIS WITH EXISTING CRYOSTAT SETUPS

The stresses, S_L, in Eq. (1) were calculated based on the sustained loads. Loads, shown in Table 2, were used to calculate the displacement range stress, S_E, in Eq. (2). For Model (3), due to the presence of the vacuum break, loads in Table 2 were applied to one of the two chambers for every analysis case and a temperature transition, from cold to warm, was assumed to occur along the vertical pipe (see Fig. 2). Bellow spring forces, based on the minimum wall temperatures and the cryostat's length, and pressure forces, based on the maximum internal pressures, were applied on the pipe ends for all the analyses. Spring rates for the bellows are about 243, 603, and 350 lbs/in (or 4.34, 10.77 and 6.3 kg/mm) for the 24", 20" and 14" ID bellow respectively. Densities for the carbon steel and SST 304L material are 0.3 lbs/in^3 (8.30 g/cm^3) and 0.28 lb/in^3 (7.75 g/cm^3) respectively. Young's modulus for the steel and the SST304L material are 30 x 10^6 psi (2.07 x 10^{11} Pa) and 28 x 10^6 psi (1.93 x 10^{11} Pa) respectively. Thermal conductivity coefficient is 16 W/m-K for the SST304L and is 43 W/m-K for the steel material.

THE ALLOWABLE STRESSES

According to the ASME Piping code [5], Table A-1 and Fig. 323.2.2A, the stresses, due to the sustained loads and the displacement strains, must not exceed the allowable stresses, as shown in Eq. (1) and Eq. (2):

$$S_L <= S_h \tag{1}$$

$$S_E <= 1.25*(S_c+S_h)-S_L \tag{2}$$

where S_L = Stress due to sustained loads, including weight and 1 atm external pressure load; S_E = Stress due to displacement strains, including loads as shown in Table 2 and bellow spring forces due to the temperature changes on the cryostats; S_h = Allowable stress at room temperature: 20 ksi (8.27 x 10^7 Pa) for carbon steel pipe and 16.7 ksi (1.15 x 10^8 Pa) for SST 304L pipe; S_c = Allowable stress at temperature between the minimum material design temperature and 38 ^{o}C: 20 ksi (8.27 x 10^7 Pa) for carbon steel pipe and 16.7 ksi (1.15 x 10^8 Pa) for SST 304L pipe. The minimum design temperatures for the carbon steel and for SST304L material are about 244 K and 19K respectively.

ANALYSIS RESULTS

Finite element analyses, using ANSYS, had been performed to calculate S_L and S_E in Eq. (1) and Eq. (2) on the different cryostat chambers with the existing setups. The maximum stresses are summarized and compared with the allowable stresses in Table 3.

Table 3: Maximum Stresses vs. Allowable Stresses with Existing Cryostat Setups

	S_L psi*	Pass Eq. (1)?	S_E psi*	Pass Eq. (2)?
Sextant Arc	8329	Pass	17754	Pass
Triplet/D0/DX +Q3-Q4 VJP +DX-D0 VJP	9822 7046 2675	Pass Pass Pass	34516 22714 10800	Pass Pass Pass
INJ Q8-Q9	8329	Pass	24960	Pass
INJ D9-Q4	8329	Pass	17885	Pass
INJ Q4-Q7	8329	Pass	20272	Pass
VJP Q7-Q8	7046	Pass	26082	Pass
VJP Q9-D9	7046	Pass	26678	Pass

*: 1 psi = 6895 Pa

The highest stress, S_E, occurred on the Triplet/D0 cryostat chamber is due to an over constraint to the chamber's thermal deformation. By changing the support pattern from RRRSSSSRSSSRR (see Fig. 3) to RRRRSSSSSSSSS, the number could be reduced to 10859 psi (7.49 x 10^7 Pa), based on ANSYS analysis.

RESULT VERIFICATIONS

Simple analytical calculations had been performed to check the results obtained from the S/F simulations and the stresses obtained by ANSYS. Good agreement had been achieved [7].

CONCLUSIONS

The conclusions are as follows:

- The S/F simulation results show that the highest internal pressure in the cryostats, due to the magnet line failure, is ~37 psig (255115 Pa).
- Based on the simulation, the temperature on the cryostat chamber, INJ Q8-Q9, could drop to 228 K, which is lower than the minimum design temperature of 244 K for the carbon steel material, allowed by the Code [5].
- Based on the ASME Code and ANSYS results, the reliefs on all the cryostats inside the RHIC tunnel are adequate to protect the vacuum chambers when the magnet cooling lines fail.
- In addition to the pressure loading, the thermal deformations, due to the temperature decrease on the cryostat chambers, could also cause a high stress on the chamber, if not properly supported.

ACKNOWLEGEMENT

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REFERENCES

[1] H. Hahn, Editor, "RHIC Design Manual Brookhaven National Laboratory", October 1994.

[2] http://www.universetoday.com/2008/12/05/first-images-emerge-of-damage-to-the-lhc/

[3] K.C. Wu, "Safety Relief for RHIC Vacuum Tank", AD/RHIC/RD-71, RHIC, BNL, NY, August 1994.

[4] C.J. Liaw, Y. Than, and J. Tuozzolo, "Helium Release Rates and ODH Calculations from RHIC Magnet Cooling Line Failure", PAC 2011, New York, NY, March 28 – April 1, 2011.

[5] "ASME B31.3-2006, Process Piping", The American Society of Mechanical Engineers, 3 Park Avenue, New York, NY.

[6] "ANSYS", ANSYS Inc., Canonsburg, PA, USA. http://ansys.com

[7] C.J. Liaw, "Advanced Calculations on RHIC Helium Release Due To Magnet Line Catastrophic Failure", Review Presentation, 3A995154 CAL -01, C-AD, BNL, January 14, 2011.