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**A Full-Scale Thermal Model of a Prototype Dipole Cryomagnet
for the CERN LHC Project**

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A FULL-SCALE THERMAL MODEL OF A PROTOTYPE DIPOLE CRYOMAGNET FOR THE CERN LHC PROJECT.

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The design, construction and thermal performance of an LHC Cryostat Thermal Model are described. In place of an LHC prototype dipole cold mass a standard cryostat houses a cold mass with the same overall dimensions, cold surface area, thermal insulation and support systems. The Cryostat Thermal Model is connected to suitably dimensioned end boxes fixing the required temperature levels, by means of saturated helium baths at 1.9 K and 4.2 K, and controlled flows of gaseous helium at 50 to 75 K. Comprehensive instrumentation and fully automatic process control and data acquisition allow precise calorimetric measurements to be performed at all temperature levels.

INTRODUCTION

The Large Hadron Collider (LHC) project at CERN [1] is based on the large-scale use of high-field superconducting magnets. To be installed in an existing 26.7 km circumference tunnel it will operate below 1.9 K in pressurized helium II. The current LHC prototype dipole magnets are about 10 m long and 0.56 m in external diameter [2]. The high cost of low-temperature refrigeration imposes tight budgeting of the cryogenic heat loads, and a precise assessment of cryostat thermal performance. Such precision measurements are difficult to perform during single-magnet bench tests due to parasitic heat loads coming from the cryogenic plant, large-current feedthroughs and equipment for "warm" magnetic measurements in the apertures.

A full-scale thermal model of the LHC dipole cryostat and cold mass has therefore been designed, built and is being commissioned. It simulates the real cryomagnet thermally and mechanically using a light structure and a small helium inventory thus minimizing cooldown and warmup times. This coupled with simplicity and operational flexibility will permit a rapid investigation of the most critical components (superinsulation systems, thermal screen, radiative insulation and support posts) in order to find a cost-optimized thermo-mechanical design. Furthermore, a precise study of full-scale cryomagnet heat transfer in particular through superinsulation systems is programmed to accurately quantify the contributions of solid conduction, residual gas conduction and thermal radiation.

CONSTRUCTION

A cross-section of the Cryostat Thermal Model is shown in Figure 1. Two concentric stainless steel cylinders 1.5 m long are spot welded together at regular intervals to form a double walled structure promoting uniform hydraulic resistance. Seven such elements are axially welded together in series and stiffened with internal rings to form a hollow cylinder 0.56 m in external diameter and 10.3 m long offering a uniform temperature distribution under steady state conditions. The maximum longitudinal thermal gradient during cooldown is fixed at 150 K/m and the design pressure at 4 bar.

The **collector**, made of stainless steel and containing a copper heat exchanger and a stainless steel tube to simulate a cold bore, is placed inside this cylindrical structure and hydraulically connected in series with it. The hollow cylinder is closed with thin aluminum covers fixed to the end flanges to form the **cold mass**. The 1.8 K **lines A** and **B** are simulated with preformed copper sheets fixed and

thermalized to the cold mass. The 20 K line **D** is of identical outer diameter to that of the LHC prototype dipole cryostat, but double walled to minimize the helium inventory.

The **thermal shield, radiative insulation, support posts, vacuum vessel, line C** at 4.5 K and the 50 to 75 K lines **E** and **F** are all identical to those of the LHC prototype dipole magnet cryostats currently under manufacture in European industry [3].

The Cryostat Thermal Model is closed at one end by the supply end box and at the opposite end by the return end box.

CRYOGENICS

The nominal steady state temperature levels envisaged for the LHC cryogenic system have been maintained and are as follows:

- 1) A thermal shield operating at **50 K to 75 K** and intercepting the heat load from ambient temperatures.
- 2) The **4.5 to 20 K** level allowing initial filling of the cold mass and thermalizing the radiative insulation.
- 3) Quasi-isothermal saturated superfluid helium cooling the cold mass at **1.9 K**.

The hydraulic circuit is shown in Figure 2. Helium at 4.2 K and 1.25 bar from the phase separator is distributed to line C at 4.5 K, to line E at 50 K, to the supply end box low temperature screens and to the cold mass for initial filling. Liquid helium subcooled in the Joule-Thomson heat exchanger expands to saturation in the valve V1 and is directed through a small bore tube to the blind end of the heat exchanger. Conduction heat exchange causes helium vapourization during its return through the heat exchanger and cools the cold mass from 4.2 K to 1.9 K.

The steady state heat load at 1.9 K is given by:

$$Q = \dot{m}_v \cdot L$$

where $\dot{m}_v$ is the vapour mass flow-rate and L the latent heat of vapourisation at 1.9 K.

The heat losses at 1.9 K can be compensated and the level in the Joule-Thomson heat exchanger kept constant by controlling valves V1 and V3.

At 4.5 to 20 K and 50 to 75 K steady state heat loads are calculated from measured mass flow-rates and inlet-to-outlet temperature differences as follows:

$$Q = \dot{m} \cdot (H(T_2) - H(T_1))$$

where $\dot{m}$ is mass flow-rate and $H(T)$ the enthalpy of the flow at temperature T .

Table 1 lists calculated heat loads at steady state nominal conditions [4]. With an overall nominal cooling rate of 0.2 g/s it has been estimated to cool the cold mass from ambient temperature to 1.8 K in 37 h where the total helium inventory of the system is 1.3 kg.

From these nominal conditions the effect of changing the residual gas pressure, thermal gradient, thermal contact quality and superinsulation system efficiency and the direct consequences on the thermal budget will be investigated. The range of measurements is limited by the maximum heater power (50 W for EY1, EY2 and EY4, and 100 W for EY3) and by the maximum and minimum flow-rates at the different temperature levels (0.005-0.25 g/s in the 1.8 K circuit, 0.03-1.5 g/s in the 5-20 K circuit, 0.018-1 g/s in the 50-75 K circuit and 0.3-1.4 g/s at the helium pumping unit at 1.8 K).

Component	Heat load					
	50-75 K		4.5-20 K		1.9 K	
	(W)	(W/m)	(W)	(W/m)	(W)	(W/m)
End boxes	22.1	9.84	0.78	0.35	0.09	0.04
Cryostat Thermal Model	58.1	5.64	1.31	0.13	1.56	0.15
Total	80.2	6.40	2.09	0.17	1.65	0.13

Table 1: Calculated heat loads at nominal steady state conditions

INSTRUMENTATION CONTROL AND DATA ACQUISITION

Temperatures are measured with Allen-Bradley carbon sensors used from 1.5-30 K and PT100 platinum sensors from 30 K to ambient temperature. Thermally resistive cryogenic wire is used and appropriate intermediate thermalization implemented on all instrumentation. Current sources are restricted to 10 μ A for carbon sensors and 100 μ A for the platinum sensors to minimize resistive heating. Platinum and carbon resistances on the cold mass will allow measurement of both transverse and longitudinal temperature distribution during cool-down and warm-up.

The control and data acquisition system is based on the LabVIEW [5] application. During all operational phases data are saved on files as spreadsheets readable by most data analysis applications. Temperatures TTC1 and TTC2 on lines C and D and TTP1 and TTP2 on lines E and F as shown in Figure 2 are controlled by varying heater power on EY1, EY2, EY3 and EY4. The temperature difference between the heat exchanger and TTC3 is maintained at less than 20 mK by controlling the flow of liquid helium through valve V1.

From an overall estimate of instrumentation errors we expect to record heat loads with a precision of ± 0.03 W at 1.8 K, ± 0.1 W at 5-20 K and ± 3 W at 50-75 K.

PROJECT STATUS

To establish their contribution to the global thermal budget the supply and return end boxes have been tested as one unit and under a range of insulation vacuum conditions from 10^{-6} mbar to 10^{-2} mbar. The heat loads to the nominal temperature levels as a function of the degraded vacuum increased substantially only for residual gas pressures higher than 10^{-3} mbar. The Joule-Thomson heat exchanger showed, as expected, a linear relation between heat load to 1.9 K and electrical power input from Q0.

After assembling the two end boxes with the Cryostat Thermal Model, first tests will establish the performance of the assembly at nominal conditions. Following this, various thermal gradients on cryogenic lines will be imposed in order to simulate the range of conditions expected in the LHC half-cell. In combination heat loads will be investigated as a function of degraded vacuum.

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Figure 1: Cross-section of the Cryostat Thermal Model

Figure 2: Hydraulic and instrumentation scheme