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of the ATLAS Magnet Cryogenic System**

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Abstract

The ATLAS magnet system, comprising a superconducting central solenoid and three superconducting toroids, has been successfully ramped up for the first time to the nominal operational current of 20.4 kA on 4th August 2008. Since then, new cryogenic operational challenges have been raised, like the smoothing of steady-state parameters, the enhancing of transient procedures to minimize thermal shocks on the magnet cold masses, the optimization of the complex cryogenic system in order to reduce the compressors electric consumption and finally how to avoid regular clogging of the shield refrigerator by water contamination. This paper presents the heat load identification of the various cryogenic sub-systems done at 4.5 K and how one of these loads was reduced, what was gained - in term of electrical consumption - by tuning the turbines settings of the main refrigerator and finally the first consolidation of the cryogenic system implemented in order to minimize the detector downtime during LHC beam runs.

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FIRST OPERATIONAL EXPERIENCE AND PERFORMANCE OPTIMIZATION OF THE ATLAS MAGNET CRYOGENIC SYSTEM

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ABSTRACT

The ATLAS magnet system, comprising a superconducting central solenoid and three superconducting toroids, has been successfully ramped up for the first time to the nominal operational current of 20.4 kA on 4th August 2008. Since then, new cryogenic operational challenges have been raised, like the smoothing of steady-state parameters, the enhancing of transient procedures to minimize thermal shocks on the magnet cold masses, the optimization of the complex cryogenic system in order to reduce the compressors electric consumption and finally how to avoid regular clogging of the shield refrigerator by water contamination. This paper presents the heat load identification of the various cryogenic sub-systems done at 4.5 K and how one of these loads was reduced, what was gained – in term of electrical consumption – by tuning the turbines settings of the main refrigerator and finally the first consolidation of the cryogenic system implemented in order to minimize the detector downtime during LHC beam runs.

KEYWORDS: ATLAS detector, large helium refrigerator, LHC experiment, superconducting magnet, turbines optimization.

INTRODUCTION

ATLAS, the largest detector operated at the LHC, has a magnet system consisting of four superconducting magnets: one barrel toroid, two end-cap toroids and one central solenoid [1-5].

The complete cryogenic system to cool-down and maintain at 4.5 K these magnets has been based on two helium refrigerators used in common by all magnets and on two proximity cryogenic systems interfacing respectively the toroids and the central solenoid [2, 3].

The main refrigerator, manufactured by Air Liquide in 1991 with an equivalent cooling capacity of 6 kW at 4.5 K, ensures the steady-state operation of all magnets at 4.5 K and the current leads cooling. It has five compressors providing a maximum mass flow of 0.5 kg/s (500 g/s) at 1.94 MPa (19.4 bar), and its cold box is based on a two-pressure cycle with three turbines. The second helium plant, the so-called shield refrigerator manufactured by Linde Kryotechnik in 2003, has a cooling capacity of 20 kW provided by two turbines in series and maintains all thermal shields circuits at a temperature between 40 K and 80 K [3].

Although the magnets are indirectly cooled by two-phase helium flowing into pipes welded on the coil casings, different flow schemes have been adopted for the toroids and for the central solenoid, thus requiring two independent proximity cryogenic systems. One centrifugal pump providing a maximum flow of 1.2 kg/s (1200 g/s) of liquid helium with a pressure rise of 40 kPa (400 mbar) is installed to keep the toroids at 4.5 K whereas a thermo-siphon cooling is applied for the central solenoid having a much simpler cylindrical geometry and piping distribution. FIGURE 1 presents a simplified flow scheme of the complete ATLAS magnets cryogenic system.

FIGURE 1. Simplified flow-scheme of the ATLAS magnets cryogenic system.

TABLE 1. Isothermal heat load distribution at 4.5 K as measured in 2008.

Cryogenic sub-system	Heat load @ 4.5 K [W]	Percentage [%]
Barrel toroid	510	30
Two end-cap toroids	290	17
Centrifugal pump providing 1.2 kg/s at 40 kPa	670	39
Proximity cryo system + valve box + transfer lines	220	13
Solenoid	20	1
TOTAL	1710	100

The isothermal heat load at 4.5 K of the entire cryogenic system seen by the main refrigerator has been measured [5] at a value of 1710 W, which is distributed among the different sub-systems as shown in TABLE 1.

In addition to this load at 4.5 K, the main refrigerator has to provide a liquefaction rate of 11.4 g/s required for the current leads cooling, divided into 10.4 g/s for the three pairs of leads servicing the three toroids and 1.0 g/s for the solenoid.

Until the nominal currents of 20.4 kA on the toroids and 7.73 kA on the solenoid were achieved on 4th august 2008, the objective was obviously to push all the cryogenic components to their maximum capacity to make this achievement possible.

OPTIMIZATION OF THE CRYOGENIC SYSTEM

The cryogenics optimization has been carried out with two objectives. The first one was to obtain the highest liquefaction rate of the main refrigerator – for a predefined cycle high pressure – while the second objective was to reduce as much as possible the total heat load at 4.5 K seen by the main refrigerator. A simultaneous improvement in these two directions (towards these two goals) would then lead to a substantial reduction of the cycle flow required by the cold box, which in turn means the possibility to switch off one of the four booster compressors. Having one compressor stopped during steady-state operation of ATLAS magnets features a win-win solution: on the one hand, there is a substantial saving on the compressors electric cost and on the other hand, from that time onwards, one booster is available for redundancy.

Optimization of the turbines operating efficiency

In order to get the highest liquefaction rate of the cold box for a predefined cycle high pressure, the three turbines have to be operated close to their maximum efficiency. Therefore, on each turbine, the spring-loaded pressure reducing brake valve – installed since the delivery of the refrigerator in 1991 – were replaced by three motorized brake valves which now automatically adapt the speed of each turbine to approach the maximum isentropic efficiency during all refrigerator transient modes and also in steady-state operation. As an important result, due to the installation of these motorized brake-valves, the duration of a quench recovery of the three toroids – starting when the 600 tons of cold masses are warmed up to an average temperature of 58 K until sufficient liquid helium is present to restart the centrifugal pump – has been reduced from 89 hours to 72 hours.

Reduction of the heat load at 4.5 K

The second objective of the optimization was a reduction of the heat load at 4.5 K seen by the main refrigerator.

When considering TABLE 1, it appears that the two largest contributors to the total heat load at 4.5 K were the barrel toroid itself and the centrifugal pump when providing the helium design mass flow of 1.2 kg/s (1200 g/s). Knowing that the power dissipated by a pump in the helium bath is related to the pump flow by equation (1):

$$\text{input power [W]} = \frac{\text{hydraulic power [W]}}{\text{pump efficiency [-]}} = \frac{Q * \Delta p}{\eta} = \frac{Q * \rho * g * \Delta H}{\eta} \quad (1)$$

where Q is the volumetric flow rate; Δp is the pressure rise; η is the pump efficiency; ρ is the density; g is the gravity; and ΔH is the head rise across the pump, it becomes obvious that a reduction of the pump flow is the only way to reduce proportionally the power dissipated in the helium bath.

The main concern was thus to determine the minimum pump-flow attainable without disrupting the smooth cooling of the toroids. Although the design operating point of the pump was calculated to keep a vapor mass fraction at the outlet of each coil of less than 10% [2, 5], we determined empirically the minimum flow required by the eight barrel toroid coils and by the two end-cap toroid coils. FIGURE 2 shows a typical test where the flow in the barrel coil B1 is reduced step by step until a small temperature rise (order of 500 mK) is detected on one of the thermometers glued onto the coil casing.

FIGURE 2. Empirical determination of the minimum forced-flow in barrel B1 coil.

Once having determined the minimum helium flow necessary in each coil, an extra margin of 10 % is taken on top of the minimum flow. The sum of the 10 minimum flows with their contingency is about 0.75 kg/s (750 g/s) instead of the 1.2 kg/s (1200 g/s) design value used before.

If the pump were capable to keep its efficiency constant even when running off-design, the pump heat load would be reduced proportionally to the reduction of the pump flow. In reality, as the pump is running off-design, a few percentage points are lost on its efficiency resulting in a pump heat load equal to 493 W (instead of 670 W measured at design point) given by the equation (2), when assuming the same pressure rise Δp :

$$input\ power_{750} = \frac{Q_{750}}{\eta_{750}} * \frac{\eta_{1200}}{Q_{1200}} * input\ power_{1200} \quad (2)$$

where $\eta_{1200} = 60\%$ is the pump efficiency measured at design point; $\eta_{750} = 51\%$ is the pump efficiency measured off-design; $Q_{750} = 0.75$ kg/s is the actual pump flow; and $Q_{1200} = 1.2$ kg/s is the design pump flow.

This reduction of 177 W (i.e. 670 W – 493 W) of the load generated by the pump corresponds to a reduction of 10 % of the total heat load at 4.5 K seen by the main refrigerator.

This heat load reduction, combined with an improved operating efficiency of the turbines, give us the opportunity to reduce the cycle flow required by the refrigerator cold box to a value of 390 g/s. As a consequence, one booster compressor providing 100 g/s of helium and consuming 250 kW of electric power is now stopped.

AIR CONTAMINATION

Since the start of operation of both refrigerators in 2005, clogging of the cold boxes is certainly the most recurrent problem we had to face with the cryogenic system. For the main refrigerator, a few months after its re-commissioning in the ATLAS technical cavern, charcoal clogging (contamination) problems occurred in the turbine circuits [3]. As this refrigerator was used for ten years in a test station prior to its re-installation for ATLAS, this problem was solved by changing the 80 K and 20 K adsorber beds and their 30 μ m filters mounted downstream.

Afterwards, clogging of the shield refrigerator by air contamination appeared repeatedly after four or five months of continuous operation. This problem required every time a warm-up at 300 K of the refrigerator and a complete regeneration of the compressors and cold box adsorber beds. Until the LHC started its continuous operation in November 2009, the cryogenic downtime of five or six days – as result of these clogging – was annoying but not really affecting the detector operation because they were in the shade of other system tunings. It was expected that the few warm-up and regeneration cycles of the refrigerator would remove definitely all contamination trapped in the circuits. However, when in April 2010, another clogging of the cold box occurred which stopped the ATLAS detector, it became clear that all these long circuits are not perfectly tight and that some small contaminations – nitrogen and water – coming from the surrounding air are unavoidable.

An existing dryer equipped with molecular sieve was installed in July 2010 to cope with this problem. FIGURE 3 presents the dryer installation, at the outlet of shield refrigerator compressor station, to which a by-pass valve was added to be able to continue the refrigerator operation, even during the dryer's four-day regeneration time.

FIGURE 3. Integration of a dryer in the shield refrigerator compressor station.

Since then, even if the leak tightness of the circuits has not been fully solved, the cryogenic system has not caused any downtime of the ATLAS detector. In order to quantify this operational availability, a “cryogenic operational availability for magnet powering” is defined for each magnet as the ratio of the time during which the magnet is effectively powered to the time during which there is a request of magnet powering (see European Standard EN 15341:2007-06). And for the first five months of operation in 2011, the “cryogenic operational availability” for the toroids was 96.2 %, when it was 97.4 % for the central solenoid.

FUTURE CONSOLIDATION

Following the performance optimization of the cryogenic system, it resulted that a cycle flow of 390 g/s is sufficient for the main refrigerator cold box to ensure, with some margin, the steady-state operation at 4.5 K of all ATLAS magnets.

The 390 g/s are first compressed from 0.11 MPa (1.1 bar) up to 0.58 MPa (5.8 bar) by four booster compressors discharging in parallel into a fifth high-stage machine which then raise the pressure up to a maximum of 1.9 MPa (19.4 bar). FIGURE 4 shows that the four booster compressors, namely C1, C2, C3 and C4, are not identical machines. C1 and C2 can provide a maximum flow of 100 g/s each, whereas C3 and C4 can provide a maximum flow of 150 g/s each. Thanks to the optimization campaign, one of the small boosters C1 or C2 can already remain off. Therefore, if C1 or C2 fails, the three remaining boosters can provide a total mass flow of $100 + 150 + 150 = 400$ g/s, a value sufficient to ensure proper operation. On the other hand, if one of the big boosters C3 or C4 fails, the three remaining boosters can provide a total mass flow of $100 + 100 + 150 = 350$ g/s, a value which is well below the required cycle flow of 390 g/s.

FIGURE 4. Consolidation of the existing main refrigerator compressor station (left graph) with the installation of new booster C2 and new high-stage C6 compressors (right graph).

Therefore, it was decided to replace the small booster compressor C2 by a new booster C2 providing a mass flow of 150 g/s, like C3 and C4 already do. With this future configuration, the fourth booster in stand-by can replace any of the other three, which will significantly reduce the risk of cryogenics downtime.

For similar reason, another obvious consolidation was to purchase a new high-stage compressor C6 which will be installed in parallel to the existing high-stage C5.

A contract has been placed in January 2011 to deliver the new C2 and C6 compressors by the end of 2011. It is foreseen to start their installation during the 2011-2012 winter shut-down.

Further consolidations of the whole magnet cryogenic system are currently under investigation to decouple the operations of the central solenoid from the toroids, while liquid helium is provided by the main refrigerator. This modification will allow the central solenoid to remain energized at 4.5 K even if, for instance, the toroids are under a three-day recovery phase after a fast dump.

CONCLUSIONS

The ATLAS magnet cryogenic system has been optimized by tuning the three turbines operating conditions of the main refrigerator to achieve a higher liquefaction rate and by reducing the heat load at 4.5 K seen by this refrigerator, via a substantial reduction of the centrifugal pump flow. The gain of this optimization has allowed the stopping of a small booster compressor providing a flow of 100 g/s.

The first step of the consolidation has been the installation in July 2010 of a dryer at the outlet of the shield refrigerator compressor station which has avoided, since then, a periodic clogging by air of the cold box after some five months of continuous operation.

The second step of the consolidation will be initiated, during the winter shut-down 2011-2012, with the installation of new redundant booster and high-stage compressors for the main refrigerator.

Further improvements of the magnet cryogenic system are currently under investigation as to decouple the operations of the central solenoid from the toroids.

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