EUROPEAN ORGANIZATION FOR NUCLEAR RESEARCH
European Laboratory for Particle Physics

Large Hadron Collider Project

LHC Project Report 857

PERFORMANCE ASSESSMENT OF 35 COLD HYDRODYNAMIC COMPRESSORS FOR THE 1.8 K REFRIGERATION UNITS OF THE LHC

F. Millet[1,] S. Claudet[2], G. Ferlin[2]

Abstract

The cooling capacity below 2 K for the superconducting magnets in the Large Hadron Collider (LHC), at CERN, will be provided by eight refrigeration units of 2400 W at 1.8 K, each of them coupled to one 18 kW at 4.5 K refrigerator. The supply of the series units was linked to successful testing and acceptance of the pre-series units delivered by the two selected vendors. The two pre-series units were temporarily installed in a dedicated test station to validate the overall capacity and to properly assess the performance of specific components such as cold compressors. Then the cold compressor cartridges to be installed in the six series and associated spare cartridges have been intensively and systematically tested in the test station. After a brief description of the test bench and the main achieved features of the pre-series units, we will present the results of the tests of 35 cold compressor cartridges. These tests show isentropic efficiency in the 75 % range, excellent reproducibility and interchange ability. Some process and system specificities observed will be reported and finally perspectives will be proposed.

1 Département de Recherche Fondamentale sur la Matière Condensée/SBT, CEA Grenoble, Grenoble, France
2 CERN, Accelerator Technology Department, Geneva, Switzerland

Presented at the CEC-ICMC'05 Conference
29 August-2 September 2005, Keystone, Colorado, USA

CERN
CH - 1211 Geneva 23
Switzerland

Geneva, 11/01/2006

PERFORMANCE ASSESSMENT OF 35 COLD HYDRODYNAMIC COMPRESSORS FOR THE 1.8 K REFRIGERATION UNITS OF THE LHC

F. Millet*, S. Claudet, G. Ferlin

Accelerator Technology Department, CERN, 1211 Geneve 23, Switzerland

* Département de Recherche Fondamentale sur la Matière Condensée/SBT, CEA Grenoble, 17 rue des Martyrs, 38054 Grenoble cedex 9, France

ABSTRACT

The cooling capacity below 2 K for the superconducting magnets in the Large Hadron Collider (LHC), at CERN, will be provided by eight refrigeration units of 2400 W at 1.8 K, each of them coupled to one 18 kW at 4.5 K refrigerator. The supply of the series units was linked to successful testing and acceptance of the pre-series units delivered by the two selected vendors. The two pre-series units were temporarily installed in a dedicated test station to validate the overall capacity and to properly assess the performance of specific components such as cold compressors. Then the cold compressor cartridges to be installed in the six series and associated spare cartridges have been intensively and systematically tested in the test station. After a brief description of the test bench and the main achieved features of the pre-series units, we will present the results of the tests of 35 cold compressor cartridges. These tests show isentropic efficiency in the 75 % range, excellent reproducibility and interchange ability. Some process and system specificities observed will be reported and finally perspectives will be proposed.

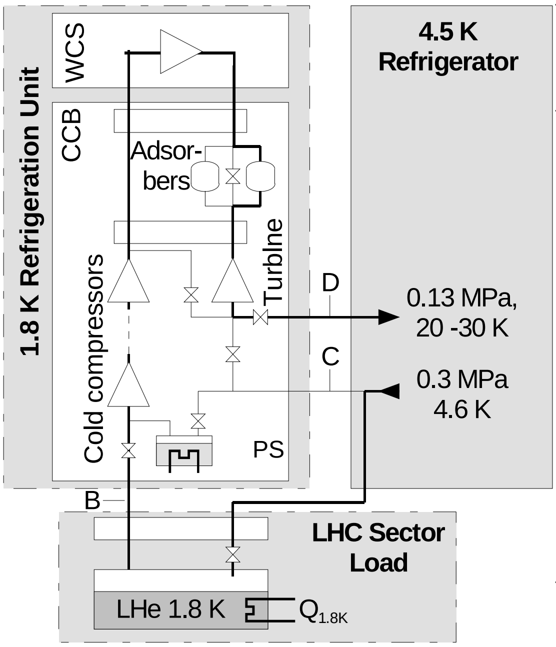
KEYWORDS: Superfluid helium, cold compressors, refrigeration.
PACS: 07.20, 05.70, 89.20.-Bb

INTRODUCTION

The 1.8 K refrigeration units for the LHC have been specified in 1998 [1] and ordered in 1999 to IHI-Linde (four units) and Air Liquide (four units) [2,3]. As described in FIGURE 1, both selected process cycles are based on a combination of cold hydrodynamic compressors in series with warm sub-atmospheric screw compressors with volumetric characteristics. One of the challenging issues of the 1.8 K refrigeration units is the cold compressors and their ability to operate over a large range of steady-state and transient modes. According to CERN technical specification and procurement scenarios, the first unit per supplier called “pre-series unit” had to be validated by extensive testing at CERN before launching production of the series units. In addition, all of the 35 cold compressor cartridges of the eight 1.8 K refrigeration units as well as associated spare cartridges (five sets of four compressors from IHI and five sets of three compressors from AIR LIQUIDE) had to be tested in the two preseries units to properly assess the performances of these specific components.

TEST FACILITY DESCRIPTION AND TEST PROCEDURE

The two pre-series units were temporarily installed at ground level in a dedicated cryogenic test station [4] connected to one 18 kW at 4.5 K refrigerator and equipped with a specific test cryostat. The test station aimed at simulating the steady-state and transient operating modes foreseen for the LHC. The steady-state conditions are defined in FIGURE 1 which also outlines the specified turn-down capability for the pumped mass flow rate (42 to 125 g/s). The transient modes can be summarized as (i) cool-down, (ii) warm-up, (iii) pump-down to 1.5 kPa and (iv) pumping flow variation up to ± 6 g/s per minute between the steady-state modes. The test station is able to process up to 130 g/s at the required conditions for the LHC and to generate flow variation up to ± 10 g/s per minute. As defined on FIGURE 1, the test cryostat has to simulate mass flows at defined temperatures with an accuracy guaranteed better than ± 1 % for the mass flow rate at header B, ± 0.1 K for the temperatures at headers B and C and ± 3 kPa for the pressures at headers C and D. For all operating modes, flow through cold compressors could be achieved with a stability of ± 0.5 g/s (for values from 40 to 130 g/s) and ± 0.1 K (for values between 4 and 5 K) over several hours. After commissioning and acceptance of the two pre-series units in 2002 and 2003 [5], systematic tests of the 35 cold compressor cartridges required for all 1.8 K refrigeration units plus one spare per supplier were performed in 2003 and 2004 to assess the performances of the cold compressor cartridges.



Modes	Interface conditions m_B [g/s]	T_B [K]	P_B [kPa]	m_C [g/s]	T_D [K]
Installed	124	4.0	≤ 1.5	≤ 2	≤ 20
Normal	83	4.0	≤ 1.5	≤ 2	≤ 25
Low beam intensity	62	4.2	≤ 1.5	≤ 2	≤ 28
Injection standby	42	4.8	≤ 1.5	≤ 2	≤ 30
Cold standby	37	5.6	≤ 24	≤ 2	≤ 30
Capacity check	0	\	≤ 1.5	126	≤ 20

$m_D = m_C + m_B$
m_C limited to 2 g/s for process optimisation

FIGURE 1. Generic scheme of a 1.8 K refrigeration unit (WCS = Warm Compression Station / CCB = Cold Compressor Box) and interface conditions for steady state modes

After warm-up of the previous train of cold compressors, the tests started with the installation of the new cartridges, and continued with piping conditioning and cool-down. The pump-down phase was then started down to 1.5 kPa and the specified process configuration (mass flow, temperature) and stability were checked. Finally, measurements were carried out once the overall process stability has been achieved.

COLD COMPRESSOR DESCRIPTION

Cold centrifugal compressors fitted with cold (80 K) active magnetic bearings and equipped with individual analog frequency drive were already successfully used in the Tore Supra [6] and CEBAF [7] installations. For LHC, as detailed in FIGURE 2, the cold compressor designs of both suppliers are similar and based on technically proven components such as room temperature active magnetic bearings, digital frequency drive and electrical motor (3-phase induction) cooled with water. All cold compressors are of the 3D (axial-radial) impeller type with fixed diffuser. Special attention was paid to prevent any air inleaks to sub-atmospheric helium circuits. Therefore, one helium guard system (associated with high-vacuum feedthroughs) isolating the process from air, is installed for IHI-Linde cartridges, while Air Liquide cold compressors are mounted in individual vacuum-pumped housing. To reduce the heat inleaks to the process flow, IHI-Linde has installed a cold heat intercept (50 to 85 K) on the short impeller shaft, supplied with cold gas subtracted from the main HP flow, whereas Air Liquide has preferred a longer shaft and a housing design with dissociated mechanical and sealing functions. Each cold compressor has its individual operating field indicating the combination of rotating speed, mass flow, pressure ratio and inlet conditions.

FIGURE 3 shows a typical operating field of these hydrodynamic compressors, which displays the pressure ratio as a function of the reduced mass-flow Mr and the reduced speed Nr, defined as follows:

$$\mathrm{Mr} = \frac{\mathrm{m}}{\mathrm{m}_0} \cdot \sqrt{\frac{\mathrm{Tin}}{\mathrm{Tin}_0}} \cdot \frac{\mathrm{Pin}_0}{\mathrm{Pin}} \qquad \mathrm{Nr} = \frac{\mathrm{N}}{\mathrm{N}_0} \cdot \sqrt{\frac{Tin_0}{Tin}} \tag{1}$$

where m is the mass-flow rate, N the rotational speed, Tin and Pin the inlet temperature and pressure of the cold compressor. The subscript 0 corresponds to the design conditions.

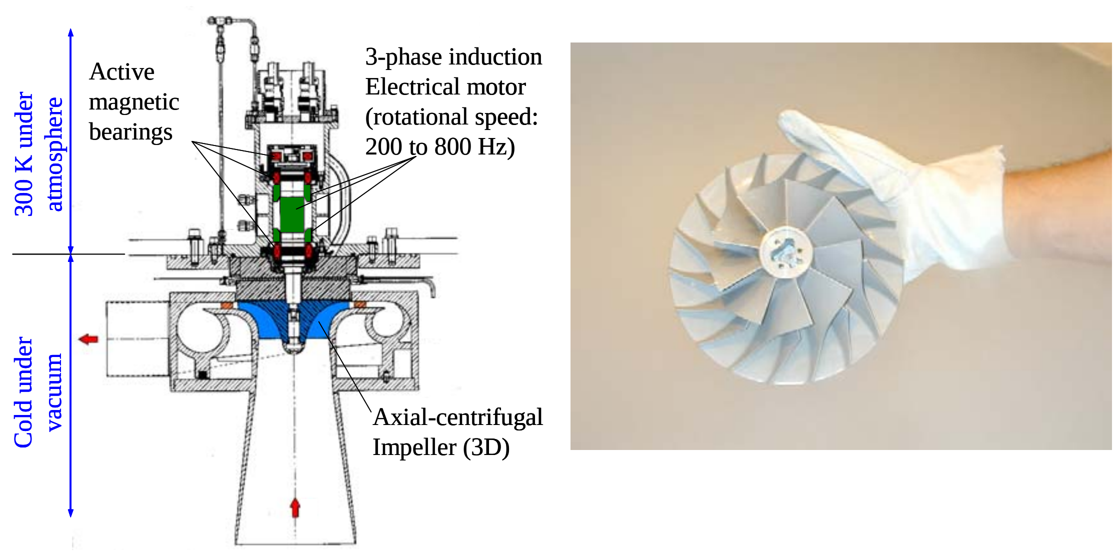


FIGURE 2. Typical cross section and 3D axial-centrifugal impeller of a cold compressor

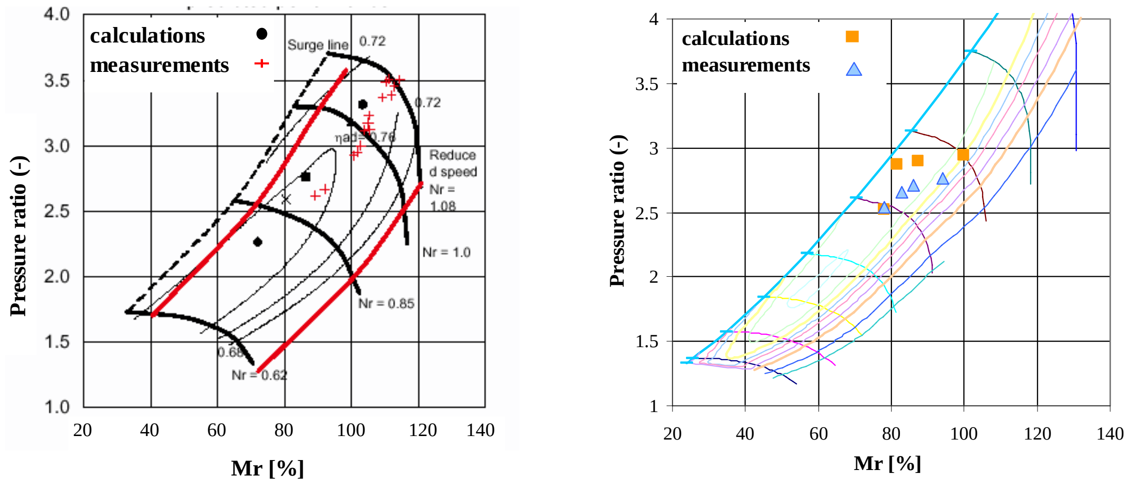


FIGURE 3. Typical operating field for a cold cartridge from IHI-Linde (left) and Air Liquide (right)

The working area is limited on the left side by the surge line, on the right by the choke line and on the top by the maximum rotational speed of the drive. A margin of 20 % for the rotational speed of each cold compressor is specified with respect to the maximum value encountered in the different operation modes.

In multistage configuration, the different cold compressor stages are interacting and have to enable safe operation of each compressor within their respective operating field. Special automatic control strategies [3,8] based on the volumetric characteristics of the warm compressors, the variable frequency drive and (for transients) the suction temperature adjustment, were developed by each supplier to keep the inlet pressure of the first stage below 1.5 kPa ± 50 Pa in steady-state operation or below 1.7 kPa during transient modes as specified.

ACHIEVED PERFORMANCE OF COLD COMPRESSOR CHAINS

The fully automatic pump-down 1.5 kPa is started when the cold compressor box is cold, with the screw compressor initially close to the required pressure at the inlet of the first stage (typically 15 to 25 kPa for 40 to 90 g/s) allowing the final pump-down with the cold compressors to start. FIGURE 4 illustrates such automatic pump-down from 20 to 1.5 kPa achieved in less than 30 min at fastest for the present pumped volume. This pump-down duration will adapt automatically to any volume to be pumped down.

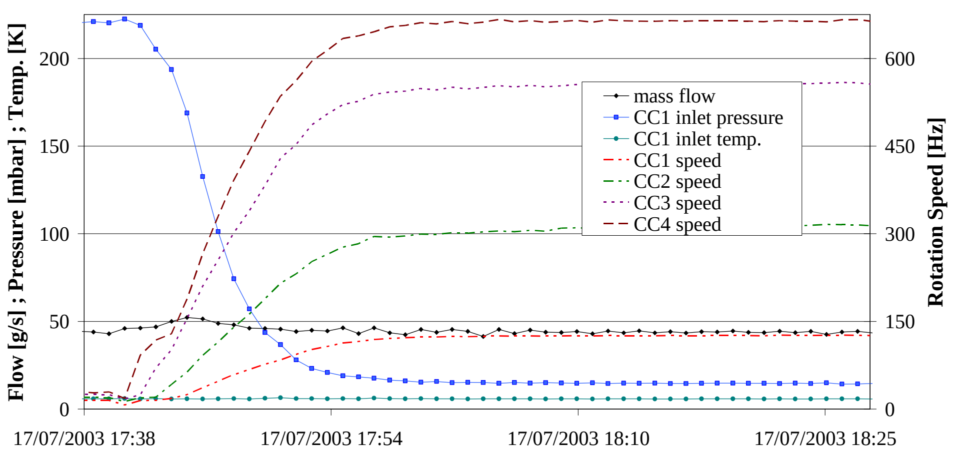


FIGURE 4. Typical automatic pump-down to 1.5 kPa (15 mbar) for IHI-Linde

TABLE 1. Averaged performances of cold compressors in the four steady state modes

Operation Mode		IHI-Linde					Air Liquide			
		CC1*	CC2	CC3	CC4	CC chain	CC1*	CC2*	CC3*	CC chain
Installed	Pressure ratio [-]	3.82	2.73	2.18	1.73	39.09	3.16	2.90	2.77	25.38
	Isentropic efficiency [%]	81	83	76	64	65	75	73	75	62
Normal	Pressure ratio [-]	2.31	2.94	2.17	1.77	26.03	2.24	2.90	2.77	18.05
	Isentropic efficiency [%]	84	84	76	58	64	74	72	73	62
Low	Pressure ratio [-]	1.84	2.74	2.20	1.8	20.01	1.93	2.72	2.70	14.23
Intensity	Isentropic efficiency [%]	78	83	76	55	61	70	71	73	62
Injection	Pressure ratio [-]	1.50	2.25	2.23	1.87	14.09	1.68	2.38	2.60	10.37
Stand by	Isentropic efficiency [%]	68	83	75	47	54	71	70	71	64

** temperature sensor correction done for CC1 inlet of IHI-Linde, CC1-CC3 inlets and CC2 outlet of Air Liquide*

The intensive tests of the cartridges have also permitted to validate the operating fields (see FIGURE 3) of each cold compressor. The measurements show a good agreement between predicted and measured values of the surge line and of the choke line, as well as the maximum speed limitation. TABLE 1 summarizes the averaged results in the four steady-state modes for the cold compressor trains. The values were measured several times for each set of cold compressor cartridges with good reproducibility (average 2 %). Air Liquide had made important effort to achieve the highest reproducibility (± 1 %) whereas IHI-Linde has tuned the compressors around ± 5 %. FIGURE 5 illustrates the test results at installed mode for the cold compressors from each supplier. To calculate isentropic efficiencies, systematic and constant corrections on temperature were applied to compensate for inaccurate measurements. The temperature discrepancies come on one hand from partial mixing of flows at the inlet of first stage (see below Process and System Specificities). On the other hand some immersed temperature sensors are not correctly cooled, due to an inadequate positioning in the pumped flow. After correction (0.1 to 0.5 K), in most of the operation modes and for most of the cold compressors, the isentropic efficiency is above 75 % (> 80 % for IHI-Linde). One can note that larger pressure ratios have been achieved (around 4 in off-design modes), without any disturbance of the cold compressors and preserving high efficiency above 70 % for any pressure ratio > 1.5.

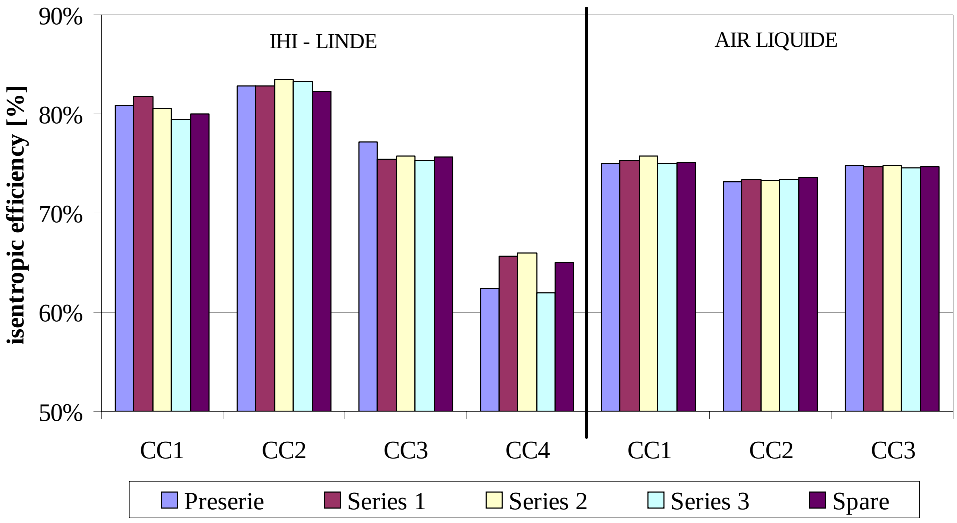


FIGURE 5. Meas. isentropic efficiency in Installed mode (2400 W at 1.8 K) for IHI-Linde and Air Liquide

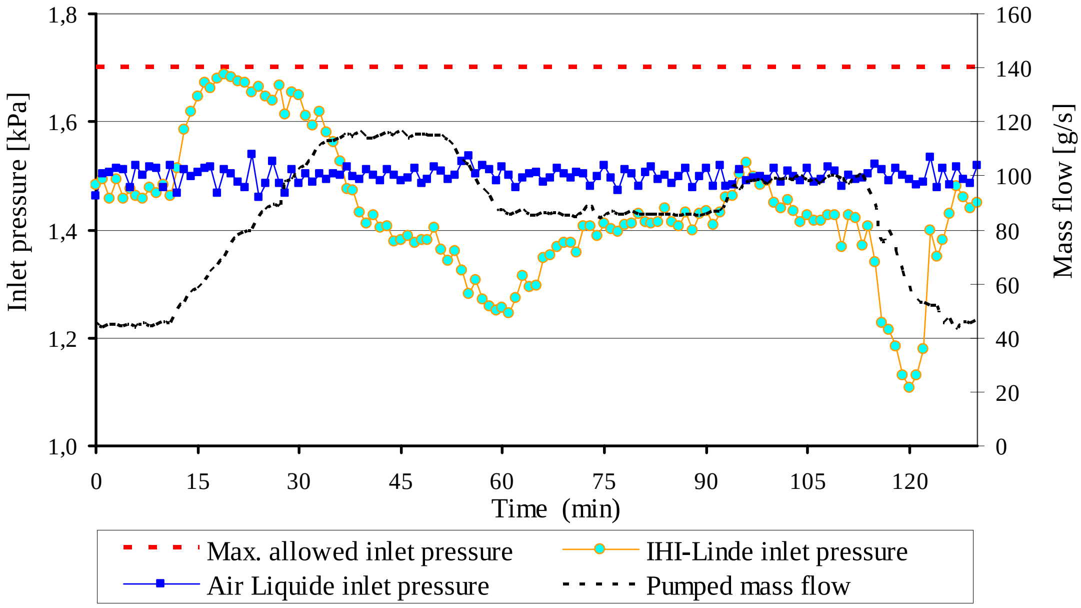


FIGURE 6. Typical specified transients and corresponding measured values for IHI-Linde and Air Liquide

The test cryostat also allowed to perform transient tests called "LHC daily" which simulate the expected change of heat loads during a nominal working day for LHC physics runs. FIGURE 6 shows the typical specified transients and measured values. This test was systematically performed in addition to the measurements in steady state modes, to assess dynamic behaviour of cold compressor chains. Both Air Liquide and IHI-Linde control solutions have provided efficient control of the mass flow variation without any additional electrical heating. The suction pressure stays all the time below the specified value of 1.7 kPa. Mass flow variations above ± 8 g/s per minute were successfully tested with maximum 500 Pa increase at suction pressure and have shown promising capability of the 1.8 K refrigeration unit to achieve the required turn-down of three (from 2400 W at 1.8 K to 800 W) in less than 15 min. In addition, one can note that the specified rotation speed margin (20 %) for each cold compressor is very useful to smoothly achieve the pressure control along mass flow variation without excessive overshoot.

Finally, unexpected stops of utilities or tuning problems of the installations during commissioning have generated landing of magnetic bearings at reduced capacity and sometimes at full capacity without any damage or performance degradation of the cold compressors. Only the Mean Time Between Maintenance (MTBM) guaranteed at 8000 hours has not yet been validated and will be assessed in the coming years.

PROCESS AND SYSTEM SPECIFICITIES

With "mixed" compression cycles (ie cold and warm) to achieve operation below 2 K, the design of the heat exchanger is facilitated by operating at higher pressure. Moreover the turn-down capability of the installation can be more easily achieved. Indeed with sub-atmospheric warm compressors having volumetric characteristics, the pressure at the outlet of the cold compressors decreases linearly with respect to the pumped flow-rate. Such a mixed cycle can therefore accommodate a large dynamic range without any additional electrical heating.

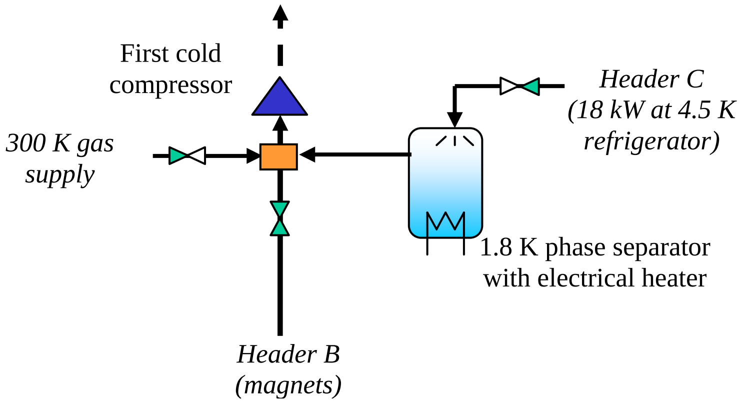


FIGURE 7. Simplified scheme for the mixing function at inlet of first cold compressor

In addition, the warm sub-atmospheric compressors are very useful during transient operation modes like cool-down and pump-down, in which the cold compressors are far from their design conditions and difficult to tune. Consequently, at constant suction pressure and for a given load or mass-flow, the only free parameter to control the cold compressor is the inlet temperature (see FIGURE 3 and Equation (1)). As shown in FIGURE 7, this inlet temperature is adjusted by flow mixing: injection of warm gas allows raising the temperature and a built-in phase separator equipped with electrical heater can generate mass-flow at lower temperature. Therefore, a temperature mixer at the inlet of the first cold compressor is implemented. This mixing chamber has to be thermally efficient (good flow mixing) and hydraulically transparent (negligible pressure drop).

To be consistent with the common practice for expander turbines, the exchange of cold compressor cartridge without complete warm-up of the installation was specified in less than five hours. It turned out to be the case with additional features for magnetic bearings and impeller positioning and balancing. The replacement of a cold compressor cartridge without need of re-tuning of magnetic bearings is performed thanks to reduced fabrication and mounting tolerances combined with thorough factory tuning of all cartridges. In addition, isolation and by-pass valves will allow keeping cold the magnets during isolation and exchange of cold compressor cartridges. The exchange of one cold compressor cartridge is therefore driven by the cool-down and warm-up phases which required specific design (valves and electrical heaters) to ensure complete warm-up of the cartridge and all associated components before opening to air in order to prevent water condensation on cold surfaces. These exchanges were performed more than fifty times without any trouble as illustrated in FIGURE 8.

FIGURE 8. Cartridge handling (left), mechanical installation (middle) and utility plugging (right)

CONCLUSION

All 35 cold hydrodynamic compressors to cool the LHC superconducting magnets below 2 K have been intensively and successfully tested prior their final underground installation at CERN.

The tests have demonstrated the mature technology of cold compressors and the interchange ability of any cold compressor cartridge. The exchange is carried out without any vacuum break of the cold box or overall warm-up of the 1.8 K refrigeration unit in less than five hours as commonly performing for expander turbine (“plug-and-play” practice). Each supplier as well as the CERN team has developed a fully automatic control of the cold compressor chain for transient and steady state modes to achieve the required stable pumping at 1.8 K.

The achieved performance of the multistage cold compressors is reproducible in terms of positioning in compression fields (isentropic efficiency, pressure ratio, speed) as well as of turn-down capabilities (mass flow variation up to ± 8 g/s per minute) and rangeability (factor 3) without additional electrical heating. Isentropic efficiencies of 75 % are now currently reached with achieved pressure ratio above 3 per stage.

The final installation of the eight refrigeration units of 2400 W at 1.8 K is now under completion and the operation with LHC magnets is expected in the coming months for one of the eight LHC sectors.

ACKNOWLEDGMENTS

The authors would like to thank all colleagues of the Accelerator Cryogenics Group at CERN and of SBT at CEA Grenoble involved in defining, building-up and operating the test station and also the pre-series unit teams from IHI-Linde, Air Liquide and their industrial partners.

REFERENCES

1. Claudet S., Gayet P., Jager B., Millet F., Roussel P., Tavian L., Wagner U., “Specification of eight 2400W@1.8K refrigeration units for the LHC”, *Proceedings of ICEC18*, edited by K.G. Narayankhedkar, Narosa, Mumbai, India, 2000, pp. 207-210.
2. Asakura H., Honda T., Mori M., Yoshinaga S., Bosel J., Kündig A., Kurtcuoglu K., Meier A., Senn A.E., “Four 2400 W@1.8 K refrigeration units for CERN-LHC: the IHI/Linde system”, *Proceedings of ICEC18*, edited by K.G. Narayankhedkar, Narosa, Mumbai, India, 2000, pp. 215-218.
3. Hilbert B., Delcayre F., Maccagnan M., Monneret E., Walter E., “Experimental results obtained with Air Liquide cold compression system for pre-series unit for CERN LHC project”, *Proceedings of ICEC20,* Beijing, China, 2004, to be published.
4. Claudet S., Ferlin G., Gully P., Jager B., Millet F., Roussel P., Tavian L., “A cryogenic test station for the Pre-series 2400 W@1.8 K Refrigeration units for the LHC”, *Proceedings of ICEC19,* edited by Guy Gistau Baguer and Peter Seyfert, Narusa, Grenoble, France, 2002, pp. 83-86.
5. Millet F., Claudet S., Ferlin G., Tavian L., “1.8 K refrigeration units for the LHC : performance assessment of Pre-series units”, *Proceedings of ICEC20*, Beijing, China, 2004, to be published
6. Claudet G. and Aymar R., “Tore Supra and helium II cooling of large high-field magnets”, *Advances in cryogenic engineering,* 35A, edited by R.W.Fast, Plenum, New York, 1990, pp. 55-67.
7. Rode C.H., Arenius D., Chronis W.C., Kashy D., Kesee M., “2.0 K Cebaf cryogenics”, *Advances in cryogenic engineering*, 35A, edited by R.W.Fast, Plenum, New York, 1990, pp. 275-286.
8. Kundig A. and Asakura H., “Control consideration of multi stage cold compression system in large helium refrigeration plants”, *Proceedings of ICEC19,* edited by Guy Gistau Baguer and Peter Seyfert, Narusa, Grenoble, France, 2002, pp. 681-686.