DESIGN, FABRICATION, COMMISSIONING, AND TESTING OF A 250 G/S, 2-K HELIUM COLD COMPRESSOR SYSTEM

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ABSTRACT

In June 1999 the Thomas Jefferson National Accelerator Facility (TJNAF) Cryogenic Systems Group had completed the design, fabrication, and commissioning of a cold compressor system capable of pumping 250 g/s of 2-K helium vapor to a pressure above 1 bar. The 2-K cold box consists of five stages of centrifugal variable speed compressors with LN_2 cooled drive motors and magnetic bearings, a plate fin heat exchanger, and an LN_2 shield system. The new 2-K cold box (referred to as the SCN) was built as a redundant system to an existing four stage cold compressor SCM cold box that was commissioned in May 1994. The SCN has been in continuous service supporting the facility experiments since commissioning. This system has achieved a significant improvement in the total 2-K refrigeration system capacity and stability and has substantially increased the operating envelope both in cold compressor flow and operating pressure range. This paper describes the cold box configuration and the experiences in the design, fabrication, commissioning and performance evaluation. The capacity of the system for various operating pressures (0.040 to 0.025 bar at the load corresponding to a total compressor pressure ratio of 28 to 54) is presented. An effort is made to characterize the components and their operating data over the tested range. This includes the return side pressure drop in the distribution system, the heat exchanger, and the cold compressor characteristics. The system design parameters and their effects on performance are outlined.

INTRODUCTION

The TJNAF (also known as CEBAF or JLab) cryogenic system is described in [1]. In brief, the Central Helium Liquefier (CHL) consists of a relatively conventional 4-K helium refrigerator and a separate subatmospheric cold box for 2-K operation of a superconducting linear accelerator (linac). The subatmospheric part of the system (SCN) consists of five

centrifugal cold compressors (CC1-CC5) and the subcooler heat exchanger (HX-10N) as shown in FIG 1.

FIGURE 1. The Jefferson Lab 2-K cryogenic system

Until June of 1999, the original Subatmospheric Cold Box – Main (SCM) was in use. CVI Corporation built this cold box, with the CC's supplied by L'Air Liquide and the heat exchanger supplied by Altech. Operation of SCM began in May of 1994 and the original commissioning experiences are described in [2]. To reduce the impact of a single system failure on 2-K operations at TJNAF, it was decided to build a redundant 2-K cold box system (SCN). Building on the lessons learned from the SCM, the design goals for the new cold box were to purchase the same design cold compressors (reduce the number of expensive spare parts), to extend the useful operating envelope in both flow and load pressure, and to improve the operating efficiency. The SCN was designed, fabricated, commissioned, tested and continues to be operated by the TJNAF cryogenic staff.

DESIGN

The improvements made in the SCN over the SCM design include the following. There are five stages of cold compressors instead of four. Parallel thermo-siphons rather than forced flow circulates the LN₂ used to cool the cold compressor motors. The sub-cooler heat exchanger (HX-10N) is larger. The piping between the stages has increased volume and is circular in cross section with flow straighteners rather than the rectangular ducts used in the SCM. Smooth transitions are used from the CC square discharge cross-section nozzle to the conventional round pipe. Design improvements are made for thermal isolation of the cold compressors, valves and piping from the support structure. In addition, the valves and bayonets have LN₂ heat stations. The radiation shield between the vacuum shell and the cold components is improved to reduce the heat input to the process helium

and also to reduce the LN₂ use. We consider these changes to be the main contributors for the improved operating envelope and efficiency of the system. The improved instrumentation greatly helped to characterize the components and the over all system.

The main mechanical design characteristics for each cold compressor are given in TABLE 1. They are the wheel diameter and the flow areas. The inlet volumes at each cold compressor are also provided. The time constant is calculated as the ratio of inlet piping volume to the volume flow rate at the design condition. D_r represents the diameter ratio of the compressors with respect to stage 4, which was the final stage in the SCM. D_r^{-2} represents the ideal gear ratio (for constant tip speed squared) between the stages for equal efficiency and equal pressure ratio compressors.

TABLE 1. Cold Compressor Design Details

	Units	CC-1	CC-2	CC-3	CC-4	CC-5
CC Inlet Connection Dia	mm	273	129.5	86.0	60.3	60.3
Wheel Inlet Dia	mm	195.5	139.3	95.3	72.8	72.8
Wheel Outlet Dia	mm	347.4	259	182.8	142.6	142.6
Volute Outlet (square)	mm	118	79	57	57	57
CC Inlet Connection Area	mm ²	55990	11979	5281	2489	2489
Wheel Inlet Area	mm ²	14691	7663	3560	1880	1880
Wheel Outlet Area	mm ²	10390	5020	2527	1779	1779
Volute Outlet Area	mm ²	12996	5776	2916	2916	2916
Inlet Pipe Dia	mm	266	163	110	110	110
Inlet Pipe length	mm	Linac+HX	4915	2794	2375	2134
Volume at Inlet	liters	25000	102.2	26.6	22.6	20.3
Time Constant	sec	43.9	0.27	0.11	0.14	0.13
Ideal Gear ratio ($1/D_r^{-2}$)		0.168	0.303	0.609	1.000	1.000
Actual Gear ratio		0.170	0.365	0.690	1.000	varies

The design characteristics of the heat exchanger are given in the following TABLE 2.

TABLE 2. Heat Exchanger (HX-10N) Design Details

	Units	High Pr.	Low Pr.
No of layers		15	30
Fin Surface	Type	1/8" serr	2" serr
Fin Height	mm	5.11	9.65
Fin Thickness	mm	0.20	0.41
Fin Spacing	fins/mm	0.89	0.32
Effective Pass width	mm	391	391
Free Flow area	m ²	0.0235	0.0939
Effective Pass Length	mm	2054	2280
Ht. Transfer Area	m ²	124.2	207.7
Dp at design	bar	0.0017	0.0022
Inlet Nozzle Size	mm	82.8	265
Outlet Nozzle Size	mm	82.8	265

PERFORMANCE

Two aspects of system performance are considered. The first is the transient operation of the system during pump down (most challenging) where the load pressure changes from

1.2 bar to the final operating pressure of 0.039 bar and 2.08K. Some of the parameters affecting this transient operation, which were recognized during the commissioning of SCM, were described in [3]. The second is the steady state operation at the final operating pressure.

The operating characteristics for all the main components are presented in the following two sets of figures. The transient characteristics during pump down are given in FIGS W.1.Yz and the steady state conditions are given in FIGS W.2.Yz. The FIG numbering system is given in the following TABLE 3.

TABLE 3. Figure numbering system (FIG W.X.Yz)

W=	2	Cold Compressors	X=	1	Transient	Y=	0	System	Y=	4	CC4
	3	HX NTU's & UA		2	Steady State		1	CC1		5	CC5
	4	LP side Pressure drop					2	CC2		6	CC Train
	5	Capacity					3	CC3		7	Pump down

Although the dimensional characteristics provide ready reference, to evaluate or to compare the performance of the centrifugal compressors or turbo-machines in general, non-dimensional parameters are used. The normal choices used for these comparisons are given in [4]. The equations used in the analysis were described in [4,5] and are given in appendix-A. Specific speed and the flow coefficient are chosen as the independent parameters (x-axis) and the torque coefficient, head coefficient and efficiency are used as the dependent parameters (y-axis). Three different efficiencies are shown in the plots. Each one has its own limitations for use in comparisons. They are:

- 1) Isentropic efficiency (equation 9): This efficiency is traditionally used in the specification and in comparison of the systems at the design point. This efficiency reduces as the pressure ratio increases for a given aerodynamic quality of the wheel.
- 2) The polytropic efficiency (equation 8): This efficiency removes the pressure ratio effect. Compressors of equal aerodynamic quality will have the same polytropic efficiency for different pressure ratios.
- 3) Isothermal efficiency (equation 11): This efficiency is traditionally used in the specifications and in comparison of room temperature compressors. It is also useful in the system Carnot efficiency evaluations.

TRANSIENT OPERATION (PUMP DOWN)

The operating characteristics for the transient operation (pump down) for each cold compressor are given in FIGS 2.1.1z-2.1.5z. FIG 2.1.6z shows cold compressor train (all five CC's) performance with respect to CC4. The relative performance of the CC's during pump down are presented in FIGS 2.1.7z. The thermal performance of the heat exchanger is given in FIGS 3.1.0z. The low-pressure side pressure drop of the return header, the sub cooler HX-10N in the SCN and the total pressure drop are given in FIGS 4.1.0z. Similarly for the steady state conditions, the operating characteristics are given in FIGS W.2.Yz

The major issue in transient operation is ensuring that each cold compressor stage remains within its allowed operational envelope during pump down from 1.2 bar to the final nominal operating pressure of 0.039 bar at the load. The challenge here is to find a satisfactory thermodynamic path (pump down path) to reach the final condition from the initial condition without violating any operational limits and causing an emergency shutdown (a "trip") of the cold compressor system. In addition, the desired path must satisfy the requirements of the operational objectives, e.g. fast pump down to improve the system availability or slow pump down to reduce the load on the 4-K plant. To get the linac

to the final operating pressure, the preset pump down mass flow rate is maintained by adjusting the CC4 speed with respect to the load (linac) pressure. CC1, CC2 and CC3 maintain a preset gear ratio with respect to CC4. CC5 is a new addition to SCN (SCM has only four CC's), it is set to maintain ~0.3 bar pressure differential across the compressor. Known major contributing factors for the trips are:

Stability of the 4-K plant during pump down

The flow in the two streams of HX-10N changes considerably depending on the liquid demand in the cryomodules (high pressure side flow) and the set mass flow through the cold compressor train (low pressure side flow). This flow imbalance and the change in discharge temperature of the CC's cause considerable load change on the 4-K cold box. The 4-K cold box turbines have fixed brake design and have a limited range of operating envelope due to their critical speed limits. This flow imbalance also causes large variations in CC1 inlet temperature, which in turn causes flow variations through the CC train. These transients can lead to CC trip.

Surge and stall characteristics of the wheels

The cold compressor impeller blades are radial in design. They are inherently capable of developing high head (high-pressure ratio) for low volume flow through the wheel. To keep away from the surge line, the mass flow rate (and thus the volume flow) through the compressor train is set high during pump down. In general the higher mass flow rate during pump down, the higher the inlet volume flow rate for each cold compressor and the process follows a more stable pump down path. A high pump down rate results in fast recovery from a trip and improved availability. The disadvantages are the increased transients in the 4-K cold box and increased load on the CC motors during pump down. In addition, the high flow rate can cause choking (stall) in the wheels. These tend to cause CC trips.

Volume available at the suction

The volume at the suction of each compressor contributes to the stability of the CC's during pump down. In SCN, the volume between stages was increased considerably (by a factor of two or more compared to SCM at some critical locations) to have a minimum time constant of 0.1 sec. The time constant is the ratio of suction (pipe) volume to the design volume flow rate. These increased volumes reduce the transfer of rapid transients from stage to stage during the pump down. In addition, flow straighteners are installed in the piping between stages to reduce the influence of one wheel's transients on the following wheel. The return header volume and the cryomodule vapor space (~25 m³) help to promote the stability of CC1 during the pump down.

Starting conditions of the helium in the linac

This depends on the length of time after the CC trip before the start of the pump down, the temperature of the helium gas used for backfill, the liquid level in the cryomodules etc. It is very difficult to come up with one thermodynamic path for pump down since the

process conditions of helium in the load vary for each pump down depending on the above conditions.

Lambda transition of the helium in the linac

When the pressure in the linac reaches 0.05 bar (lambda transition) the pump down stalls. The cryomodule JT valves open due to dropping liquid levels and the load on the 4-K plant is at a maximum. At this time HX-10N has the highest imbalance in the two stream flows. This increases the CC1 inlet temperature, causing the CC's to speed up to the maximum speed limit. This increased load on the CC's resulted in trips.

Torque (motor current limit)

The torque required by each wheel during pump down with respect to speed is shown in FIGS 2.1.1a-2.1.5a. The linear relationship between the motor voltage and speed (imposed by the variable frequency motor controllers) leads to high current draw during pump down. This conflicts with the tight current safeties programmed by the manufacturer and caused numerous CC trips. This mismatch between the wheel load and the motor current limit was greatest in CC1 due to its large wheel size, low speed and low maximum current limit. The speed versus voltage curve and the high current limit for CC1 has been modified by TJNAF to eliminate some of these problems. Some of the torque issues can be explained as follows:

1. The torque demand by the CC wheel on the motor depends on:
 - a) The density of helium at the start of the pump down, which can be ~30 times greater than the normal operating condition (Initial 1.2 bar to the final 0.039 bar). It also depends on the fifth power of the wheel diameter and on the square of the rotational speed of the wheel ($\tau = C \cdot \rho \cdot D^5 \omega^2$).
 - b) The wheel efficiency (due to the non-ideal velocity triangles at inlet and outlet of the wheel, the developing boundary layer skin friction losses etc.) will be substantially lower for the wheel during pump down. This in turn increases the torque requirement by the wheel.
 - c) The acceleration load of the CC wheel mass (significant for CC1 and CC2) as a function of the speed.
2. The CC wheel torque demand that can be supported by the motor (torque capacity of the motor) at the transient pump down operating condition (low to final speed) depends on:
 - a) Percent full load on the motor.
 - b) Power factor at the operating condition.
 - c) Efficiency of the motor.
 - d) The current and voltage limits on the motor and the equipment (the frequency drives, transformers etc.) supporting the motor.
3. The available torque for the pump down:

The available motor torque (2) for acceleration of the wheel for the pump down at any operating point on the pump down path has to be greater than the wheel demand (1). There may be some additional factors, which further reduce the available motor torque for the acceleration of the wheel for the pump down.

The pump down path chosen at JLab with the above objectives is shown in FIG 2.1.7a with respect to linac pressure and with respect to time in FIG 2.1.7b. The CC torque characteristics for this path are shown in FIGS 2.1.7c-d. The pressure ratios for each CC are shown in FIGS 2.1.7e-f. The ~30K helium gas used to pressurize the linac after a CC trip is not in equilibrium with the liquid in the cryo-modules so it has a transient effect on the measurements for the first 20 min. Since the duration of pump down is very short (on the order of one hour), no attempt has been made to improve this path any further.

An attempt is made to find the minimum flow required for the pump down. The final test was done with CC flow not to exceed 170 g/s. With 170 g/s pump down flow, it took ~3 hours to complete the pump down and 1 hour to stabilize the system, compared to less than 1.5 hours with 300 g/s pump down flow. Although it is possible to do pump down with flow less than 170 g/s, the time required is too long for practical use.

STEADY STATE OPERATION

Cold Compressors

The characteristics for the steady state operation of each cold compressor are given in FIGS 2.2.1z-2.2.5z. FIG 2.2.6z shows the performance of CC train with respect to CC4.

The cryomodule heaters control the pressure in the linac. CC4 (and the remaining CC's with fixed gear ratios) maintain the set flow from the linac. Switching the two control loops proved to be less stable on pressure at the load and for other transient operating conditions. CC set flow is adjusted to meet the linac RF load.

HX-10N and Return Header

The steady state thermal performance of the heat exchanger is given in FIGS 3.2.0a-b. This heat exchanger has approximately 3 NTU's (~2 NTU's in the SCM cold box HX) and the UA variation with flow is also shown in the figures. The increased length (NTU's) of the sub-cooler HX improved the heat transfer between the streams and thus reduced the cold end differential temperature to less than 0.2-K, which improved the system capacity. The increased cross section of the HX reduced the pressure drop. The fin selection is changed from SCM to improve the HX performance for the given HX volume.

The low-pressure side pressure drop performance of the return header, the subcooler HX-10N in the SCN, and the total pressure drop are given in FIG 4.2.0a-b.

LN₂ System

The liquid nitrogen (LN₂) thermo-siphon allowed the motors to operate near 80K; compared to 150K in the forced flow cooling in the SCM cold box. This in turn reduced the heat leak to the helium flowing in the cold compressors and also has a positive effect on the motor current characteristics. Extreme care is taken in the design of the LN₂ cooled shield and also in installing the components with good thermal isolations.

The cold box LN₂ use as a stand-by unit at 80K temperature is measured to be 3.5g/s. The radiation shields have a surface area of ~55m². It is estimated that half the LN₂ is used by the shields and the other half by the heat stations on the valves, bayonets, and internal support structure. During normal operation, the inefficiency factor of the drive motors input

power also uses LN_2 . The pre-cooling circuit in the 4-K cold box utilizes the 80K-vapor nitrogen from the shield and the CC motors.

SYSTEM CAPACITY

The system refrigeration capacity as a function of flow is given in FIG 5.2.0a. The three curves represent the primary load (Q_p , energy input at constant temperature and pressure) in the cryomodules, the heat leak into the return header distribution system (Q_s) and the total ($Q_t=Q_p+Q_s$) of the above two. The load flow capacity of the system with respect to the load pressure is given in the FIG 5.2.0b. In this graph up to 0.036 bar, the system capacity is limited by the maximum capacity available in the 4-K cold box and up to 0.032 bar both 2-K and the 4-K reach the maximum capacity and below 0.032 bar, the CC's reached the max speed limit. Therefore, the system that reaches its limit depends on the operating conditions. Using FIGS 5.2.0a and 5.2.0b, the maximum available refrigeration capacity at the linac (for the JLab 2-K system) can be determined for operating pressures from 0.025 bar to 0.04 bar.

DISCUSSION

The improvements made in the design increased the system capacity by more than 10% over the original 2-K cold box SCM ($> 400\text{W}$ at 2-K) at the given 4-K cold box load and also provided an increased operating envelope (actual tested range) given in FIG 5.2.0b. It is estimated that approximately half of the capacity improvement came from the improved heat exchanger design. The other half came from the reduced heat leak to the helium flow from the return header, cold compressor motors, CC supports, improved heat stationing of the valves and bayonets and the reduced pressure drop between the stages. The capacity improvement was substantiated by the operation of the SCN system at a reduced CC flow by more than 20 g/s at the same energy level in the linac. It was difficult to compare the results of the two 2-K cold boxes (SCN and SCM) due to the instrumentation and the limited operating range for the SCM.

JLab developed many satisfactory pump down paths that keep all the system components within the design limits and satisfactorily reach the final required load (linac) pressure with the goals of fast pump down or minimum load on the 4-K cold box. It was found to be very important to design all the components over the entire range of operation including pump down rather than on the steady state operating conditions. The data obtained from the testing and presented here can be used for the future improvement of the existing wheels or to predict the performance of similar compressor wheels of different capacity and/or operating at different conditions.

Due to the limited time allocated by the lab for testing the cryogenic systems, an effort was made to characterize the operating envelope as close to the limiting conditions as possible while minimizing the risk of CC trips to maximize the testing and minimize the wasted time for pump down. Therefore it may be possible to improve the performance over the presented range, but improvements are expected to be marginal.

Figure-2.1.1a

Figure-2.2.1a

Figure-2.1.1b

Figure-2.2.1b

Figure-2.1.1c

Figure-2.2.1c

Figure-2.1.1d

Figure-2.2.1d

Figure 2.1.1z: CC1 During Pumpdown

Figure 2.2.1z: CC1 In Steady State

Figure 2.1.2z: CC2 During Pumpdown

Figure 2.2.2z: CC2 In Steady State

Figure 2.1.3z: CC3 During Pumpdown

Figure 2.2.3z: CC3 In Steady State

Figure 2.1.4z: CC4 During Pumpdown

Figure 2.2.4z: CC4 In Steady State

Figure-2.1.5a

Figure-2.2.5a

Figure-2.1.5b

Figure-2.2.5b

Figure-2.1.5c

Figure-2.2.5c

Figure-2.1.5d

Figure-2.2.5d

Figure 2.1.5z: CC5 During Pumpdown

Figure 2.2.5z: CC5 in Steady State

CONCLUSIONS

The new 2-K cold box was commissioned in three days and placed into operation in June 1999. It has been continuously supporting experiments at TJNAF with improved efficiency, capacity, and operational stability since that time. The SCN is more forgiving than the SCM of load disturbances and failures of instrumentation and controls. When the system trips it is easy to bring it back on line. The TJNAF management's decision to rebuild the SCM like the SCN is a testimony to its success. The rebuild is presently under way.

ACKNOWLEDGEMENTS

The authors wish to express their appreciation and thanks to the TJNAF management for their support and to the design and fabrication team members for their dedicated efforts to make this project a great success for the cryo group and JLab. This work was supported by the U.S. Department of Energy under contract no. DE-AC05-84ER40150.

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APPENDIX-A

Power	$P_{actual} = \dot{m}(h_o - h_i)$	(1)	Where
Torque	$\tau = P_{actual} / \omega$	(2)	D = Wheel Diameter (m)
Energy Transfer	$E_n = g(P_o - P_i) / \rho_i$	(3)	E_n = Head Rise (m^2/s^2)
Flow Coefficient	$\phi = V_i / \omega D_o^3$	(4)	h = Enthalpy (kJ/kg)
Specific Speed	$N_s = \omega V_i^{1/2} / E_n^{3/4}$	(5)	$\dot{m}$ = Flow (kg/s)
Torque Coefficient	$C_\tau = \tau / \rho_i D_o^5 \omega^2$	(6)	N = Rotational Speed (Hz)
Head Coefficient	$C_h = E_n / \omega^2 D_o^2$	(7)	p = Pressure (bar)
Polytropic Efficiency	$\eta_p = \Psi \ln(P_r) / \ln(T_r)$	(8)	P = Power (kW)
Isentropic Efficiency	$\eta_{s=c} = P_{s=c} / P_{actual}$	(9)	Q = Energy input (W)
	$\eta_{s=c} = (P_r^\Psi - 1) / (P_r^{\Psi/\eta_p} - 1)$	(10)	R = Gas Constant
Isothermal Efficiency	$\eta_{t=c} = \dot{m} R T \ln(P_r) / P_{actual}$	(11)	T = Temperature (K)
			V = Volume flow (m^3/s)
			ρ = Density (kg/m^3)
			γ = Sp heat ratio ≈ 1.667 for He
			$\Psi = (\gamma - 1) / \gamma = 0.4$ for He
			τ = Torque (Nm)
			ω = Rotational Speed (rad/s)
			Subscripts
			i = inlet
			o = outlet
			r = ratio