Commissioning and operational results of the 12 GeV helium compression system at JLab

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Abstract. The new compressor system at Jefferson Lab (JLab) for the 12 GeV upgrade was commissioned in the spring of 2013 and incorporates many design changes, discussed in previous publications, to improve the operational range, efficiency, reliability and maintainability as compared to previous compressor skids used for this application. The 12 GeV helium compression system has five compressors configured with four pressure levels supporting three pressure levels in the new cold box. During compressor commissioning the compressors were operated independent of the cold box over a wide range of process conditions to verify proper performance including adequate cooling and oil removal. Isothermal and volumetric efficiencies over these process conditions for several built-in-volume ratios were obtained. This paper will discuss the operational envelope results and the modifications/improvements incorporated into the skids.

1. Introduction

The successful commissioning of the 12 GeV Jefferson Lab (JLab) compressor system in the spring of 2013 and subsequent operation for over two years has accumulated ~15,000 of run time. During this time the compressor system has been operated over a very wide range of operation with very little special attention. A successful development of a wide range compressor system was a crucial element in allowing a full implementation of the patented Ganni – Floating Pressure Process Cycle [1-3]. Even if the cold box design has been properly considered [4, 5], the compressor system forms the cornerstone of a helium refrigeration system that must be capable of a wide range of operation while maintaining high efficiency [3, 6]. This compressor skid design is being used for the Michigan State University (MSU) Facility for Rare Isotope Beam (FRIB) project [7], planned for the Linac Coherent Light Source (LCLS-II) project and is sought internationally.

2. Background

Even with a well match compressor and cold box system [4], there is typically considerable margin allowed in the load estimation. This can be due to performance uncertainty one-of-a-kind specialized equipment, commissioning requirements (the support of partial loads or large cool-down loads) and transient process requirements. The actual load being supported can vary considerably from the design goals as well. This can be due to under-performing and/or superior-performing sub-system components, evolution and/or changes in the experimental physics program (that the refrigerator is supporting) and operating budget restrictions. So, notwithstanding differences in the actual helium
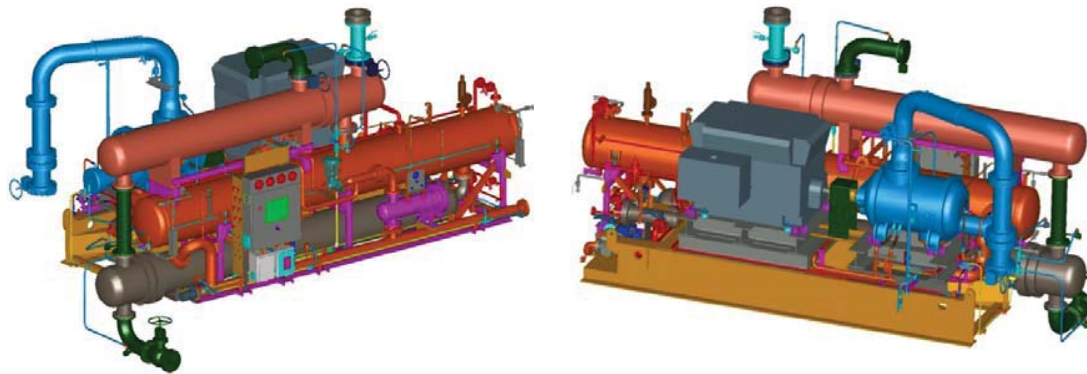
Figure 1. 12 GeV high pressure (HP) compressor skid

refrigeration system vs. the design, it is quite common for the actual load to be significantly less than the expected maximum. Even the nominal and turn-down (or stand-by) loads can be significantly different. Simply put, the system design of the anticipated loads is an estimate. It is therefore of great importance to design the cryogenic system with considerable flexibility, balanced with cost and good efficiency for the anticipated primary operating mode(s).

Additionally, as is well known in the field, even large helium refrigeration systems (which are typically more efficient [8, 9]), the availability lost in the compression system typically accounts for two-thirds of the input power losses [6, 10, 11]. This is often not well appreciated, as these systems are extremely energy intensive processes [12, 13], especially 2-K helium refrigeration systems [9, 14].

Table 1. 12 GeV compressor skid specifications (selected)

Stage		Units	HP	MP	LP	*Notes*
No. Units			1	1	3	
Compressor			Howden	Howden	Howden	
			WLViH 321/193 (Variable Volume)	WLVi 321/165 (Variable Volume)	WLVi 321/193 (Variable Volume)	
Displacement		[m^3/s]	1.774	1.577	1.774	*(a)*
Motor Rating		[kW]	1864	671	671	*(b)*
Motor Service Factor			1.15	1.15	1.15	
Weight (Est.)		[t]	30	22	20	*(c)*
Oil Charge (Est.)		[ℓ]	1400	1100	1000	
Water Flow (Est.)		[ℓ/s]	61	23	21	*(d)*
Oil Cooler	Type		AEU: Shell (Oil, 1 pass) & Tube (Water, 2 pass)			
	Duty	[kW]	1671	617	617	
	(UA)	[kW/K]	77.6	29.4	31.2	*(e)*
Helium After-Cooler	Type		AEU: Shell (Water, 1 pass) & Tube (Helium, 2 pass)			
	Duty	[kW]	448	184	105	
	(UA)	[kW/K]	20.4	8.44	4.66	*(e)*
Oil Pump Motor		[kW]	5.59	3.73	3.73	
Rotor Oil Injection Filter		[μm abs.]	8	8	8	*(f)*
Bearing Oil Injection Filter		[μm abs.]	8	8	8	*(f)*
Helium Pressure Rating		[barg]	22.4	12.1	12.1	
Oil Pressure Rating		[barg]	26.5	15.5	15.5	

Notes: *(a)* at 59.17 Hz; *(b)* Westinghouse 4160 V; *(c)* Metric tonnes; *(d)* At 8.3 K temperature difference; *(e)* (UA) is the net thermal rating; *(f)* 98% efficiency

The development of the 12 GeV compressor system was brought about by an internal development formed by a key motivating factor, observations of existing compressor systems and a number of fortuitous opportunities. The internal development of the Ganni Cycle – Floating Pressure Process [1-3], which was motivated in part by having to deal with the mentioned inevitable load variances and an awareness of the inherent energy intensiveness of these systems, formed the key motivating factor. Although the idea of this was birthed at Superconducting Super-Collider Laboratory (SSCL), several refinements and applications (to the existing systems) occurred at JLab. The benefit of operating screw compressors used in helium refrigerators encompassing a wide range of sizes (100 to ~2000 kW), compressor manufacturers and skid packagers for the past 35 years provided a broad range of experience. However, it was the fortuitous opportunity provided by NASA Johnson Space Center's (JSC) request to the JLab Cryogenics Group to design and specify a 12 kW 20-K helium refrigeration system [15] for the James Webb program which was pivotal. The development of the 12 GeV compressor system would not have been possible without the courageous support of the NASA-JSC project team. This refrigeration system provided a demonstration of new wide range compressor design concept which afforded an acceptable level of risk to the JLab 12 GeV project.

The design aspects of the 12 GeV compressor skids have been previously discussed [16]. Table 1 re-iterates the skid specifications. Figure 1 shows the high pressure (HP) compressor skid design. For the 12 GeV and MSU-FRIB project (and anticipated for the LCLS-II project) they are completely design by JLab and built-to-print by industry.

3. Commissioning and operational envelope

Two of the key compressor skid features described in [16] that allow a wide range of operation, of both suction and discharge pressures at a fully loaded compressor condition, are the bulk oil removal design and oil management. Table 2 and figure 2 show the range of conditions tested during commissioning and table 3 and figure 3 indicate the range of conditions that these skids have been operated at since being commissioned. Both the bulk oil removal design and automatic cooling oil injection were found to perform as planned for all test and operational conditions listed. As anticipated in the design, except for the medium pressure (MP) stage at high suction pressures (~3 bar), the external coalescer collected a minimal amount of oil.

Table 2. Range of conditions tested during commissioning of 12 GeV compressors

Compressor Stage	Suction Pressure [bar]	Discharge Pressure [bar]	Pressure Ratio	BVR [a]
LP Stage	1.06	3.55 to 6.08	3.33 to 5.71	2.2 to 3.2
MP Stage	1.06 to 3.04	3.55 to 6.08	2.00 to 3.33	2.2 to 3.2
HP Stage [c]	1.07 to 5.47	13.37 to 18.75	3.00 to 16.98	2.2 to 5.0 [b]

[a] BVR increments were 0.2

[b] Not every BVR setting was used for each test point due to manufacturer maximum suction pressure limitations (for a given BVR)

[c] HP stage can be operated as a (swing) LP or MP stage; under these conditions the suction pressure can be as low as ~1.04 bar, the discharge pressure as high as ~18.7 bar, and the pressure ratio ~18

Table 3. Range of conditions that 12 GeV compressors have been operated at since being commissioned

Compressor Stage	Suction Pressure [bar]	Discharge Pressure [bar]	Pressure Ratio
LP Stage	1.04 to 1.22	3.40 to 7.03	3.17 to 6.55
MP Stage	1.12 to 2.26	3.37 to 7.07	2.48 to 5.58
HP Stage [c]	3.19 to 5.78	7.25 to 20.04	2.27 to 3.70

[c] Same note as in table 3

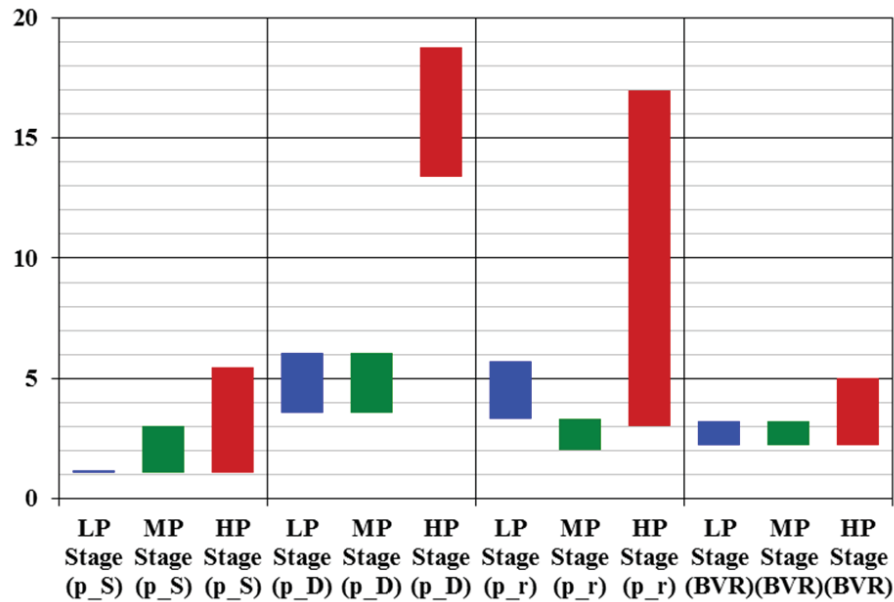LP BVR set to 2.4 for operation; MP and HP BVR set to 2.2 for operation

Figure 2. Range of conditions tested during commissioning of 12 GeV compressors
Note: p_S – suction pressure [bar], p_D – discharge pressure [bar], p_r – pressure ratio,
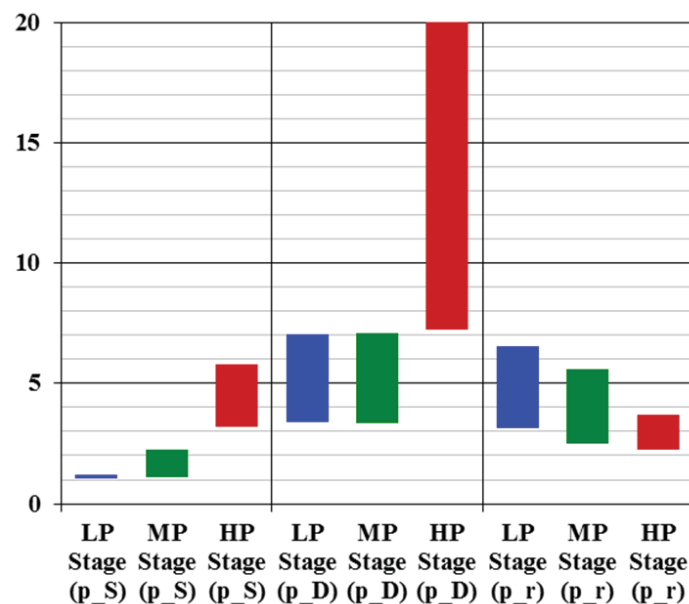BVR – built-in volume ratio (setting)

Figure 3. Range of conditions that 12 GeV compressors have been operated at since being
commissioned
Note: p_S – suction pressure [bar], p_D – discharge pressure [bar], p_r – pressure ratio

4. Minor component improvements

During commissioning several minor improvements were incorporated to further improve reliability and efficiency.

a. Installation of a helium after-cooler (A/C) liquid level gauge. This modification utilized existing ports on the compressor A/C's and allows a local visual check to adjust the oil drain in order to minimize the helium bypass.

b. Modification of A/C oil drain line. This increased the size of the line (including check valve and needle valve) to prevent blockage from fabrication debris and incorporates o-ring seal unions to allow easy removal if necessary.

c. Elimination of HP stage super-feed injection flex-hose. Original flex-hose was a convoluted metal braided hose and developed a crack at the weld between the cuff and bellows. This line was replaced with hard-tubing to prevent fatigue of thin metal but allow adequate flexibility.

d. Oil pump VFD modification. This allowed the oil pump to ride through small power cycle interruptions.

e. Elimination of cooling oil control valve positioner. This valve supplies cooling oil to the super-feed port. The original positioner would stop functioning. Although less critical on the LP and MP stages, it will cause a shut-down of the HP stage very quickly. The upgraded positioner developed a similar problem. These were replaced with an I/P.

f. Ability to de-pressurize LP and MP stage suction lines. The helium discharge check valve will tend to leak at a greater rate than the helium suction check valve. This can result in a high initial gas charge in the LP and MP stages which can cause difficulties during a re-start. Connections were made to existing ports to de-pressurize the trapped volume of helium gas to reduce the starting torque.

g. Heat shield between compressor PLC cabinet and the coolers and BOS. Although no known issues resulted from not having the shield, one was installed to protect local control system.

All of these have been incorporated into drawings for future compressor skids, such as the MSU-FRIB project.

5. Conclusion

This present development is a culmination of many years of experience, observation, questioning and opportunities presented and taken. It has allowed a very wide range of operation and a full implementation of the Floating Pressure Process [1-3]. With ~15,000 hours of operation (by the beginning of the 2015 summer) this design has proven to be extremely flexible to meet the floating pressure process demands in a very efficient and reliable manner. They have shown to be easily maintainable as well.

Further work is planned to develop alternate BOS designs and investigate methods and processes to improve efficiency [14]; specifically in regards to (1) oil choice, oil injection temperature and oil injection method on the isothermal efficiency and (2) the behavior of the helium-oil mix in the compression process and the effect of dissolved helium in the bulk oil flow.

The demand of ~250 W of input power per 1 W of cooling at 4.5-K and at least ~750 W of input power per 1 W of cooling at 2-K (~30 mbar), and the fact that the compression system losses in helium refrigeration systems comprise the majority of input power loss, warrants continued research and development.

6. Acknowledgements

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