PAPER • OPEN ACCESS

Modifications to JLab 12 GeV Refrigerator and Wide Range Mix Mode Performance Testing Results

To cite this article: P Knudsen *et al* 2017 *IOP Conf. Ser.: Mater. Sci. Eng.* **171** 012015

IOP Conf. Series: Materials Science and Engineering **171** (2017) 012015 doi:10.1088/1757-899X/171/1/012015

Modifications to JLab 12 GeV Refrigerator and Wide Range Mix Mode Performance Testing Results

P Knudsen, V Ganni, N Hasan, K Dixon, R Norton, J Creel

Cryogenics Group, Engineering Division, Thomas Jefferson National Accelerator Facility, 12000 Jefferson Ave., Newport News, VA 23606, USA

E-mail: knudsen@jlab.org

Abstract. Analysis of data obtained during the spring 2013 commissioning of the new 4.5 K refrigeration system at Jefferson Lab (JLab) for the 12 GeV upgrade indicated a wide capacity range with good efficiency and minimal operator interaction. Testing also showed that the refrigerator required higher liquid nitrogen (LN) consumption for its pre-cooler than anticipated by the design. This does not affect the capacity of the refrigerator, but it does result in an increased LN utility cost. During the summer of 2015 the modifications were implemented by the cold box manufacturer, according to a design similar to the JLab 12 GeV cold box specification. Subsequently, JLab recommissioned the cold box and performed extensive performance testing, ranging from 20% to 100% of the design maximum capacity, and in various modes of operation, ranging from pure refrigeration, pure liquefaction, half-and-half mix mode and at selected design modes using the Floating Pressure – Ganni Cycle. The testing demonstrated that the refrigerator system has a good and fairly constant performance over a wide capacity range and different modes of operation. It also demonstrated the modifications resulted in a LN consumption that met the design for the pure refrigeration mode (which is the most demanding) and was lower than the design for the nominal and maximum capacity modes. In addition, a pulsed-load test, similar to what is expected for cryogenic systems supporting fusion experiments, was conducted to observe the response using the Floating Pressure – Ganni Cycle, which was stable and robust. This paper will discuss the results and analysis of this testing pertaining to the LN consumption, the system efficiency over a wide range of capacity and different modes and the behaviour of the system to a pulsed load.

1. Background

The 12 GeV 4.5 K helium refrigerator system at Jefferson Lab (JLab) was commissioned in the spring of 2013. This system essentially doubled the refrigeration capacity to the LINAC [1]. The refrigerator proper is physically divided into two cold boxes (CBX's); an 'upper' CBX spanning 300 to 60 K, which incorporates the liquid nitrogen (LN) pre-cooler and 80 K adsorber beds, and a 'lower' CBX spanning 60 to 4.5 K, which incorporates the turbines, 20 K adsorber and 3000 liter helium sub-cooler. The 'upper' CBX is oriented vertically and located outside and the 'lower' CBX is oriented horizontally and located inside. It should be noted that all heat exchangers (HX's) are oriented vertically (warm-end on top), including in the 'lower' horizontal CBX. The CBX has three streams (high pressure supply, recycle return and load return) and four expansion stages comprising seven turbo-expanders that use a variable brake and have a dynamic gas bearing. The coldest expansion stage uses a Joule-Thompson (JT) turbo-expander discharging at approximately 3 bar. The other three expansion stages use two turbines in series (i.e., no HX in between), sometimes referred to as a 'turbine string'. The compressor system has three low pressure (LP) load return stages, one medium pressure (MP) recycle return stage and one high pressure (HP) stage that handles the flow from the low and medium stages. The only large helium systems (> 300 W) that use cold-cryogenic compressors to bring the entire sub-atmospheric (1.8 or 2 K) flow load to positive pressure are JLab (both the original CEBAF refrigerator and the new 12 GeV refrigerator), the Spallation Neutron Source (SNS) at Oak Ridge, Tennessee and the refrigerator at DESY for the XFEL. However, the FRIB refrigerator at MSU, anticipated to be commissioned in 2018, is designed for full cold compression as well. The initial system process design, specification and integration of all of these plants, except at DESY, were done by JLab. The primary load supported by the 4.5 K CBX is a liquefaction flow between 4.5 K and approximately 30 K. The JLab refrigerator also support a (nominal) 35-55 K helium shield and a modest 4.5 K (to 300 K) liquefaction load (for filling LINAC cryo-module SRF Niobium cavity vessels).

2. Liquid Nitrogen Pre-Cooler Performance Issue

As discussed in previous publications [2, 3], during the commissioning of the 12 GeV refrigerator, it was found that the system achieved the required capacity at all design modes while maintaining good efficiency. However, the LN consumption was roughly three to four times the design at maximum capacity and in a maximum 4.5 K pure refrigeration (supply to return 'balanced' flow) mode. Design and tested process conditions were close. The manufacturer thermal design margin, including longitudinal conduction, was ~30%. The JLab specification required all (brazed aluminum) HX's to be oriented vertically with the warm-end on top, a total thermal design margin of at least 10%, the NTU's per meter of core length to not exceed 10 and key pressure ratios to be at least 3; namely, the core to distributor pressure drop and the sum of core and distributor to the sum of the header and nozzle pressure drop. The manufacture did not meet the JLab specified NTU per meter of effective length (13.9). However, the pressure drop ratios were acceptable. These requirements have been developed over the past 30+ years of observation and analysis of various refrigerators in order to achieve good performance down to at least a 30% turn-down in capacity if the Ganni – Floating Pressure Process is implemented [4, 5] (ideally, using this process, the ratio of the pressure drop to pressure is roughly constant as the capacity is turned-down).

During the summer of 2015, the manufacturer installed a re-design of the LN pre-cooler. The original three stream helium HX was split into two HX's in a separate CBX; with the original helium HX abandoned. The original helium-nitrogen HX was kept, but this has a very small influence on the nitrogen consumption [6]. The two new helium-helium HX's paired the HP supply to the recycle return and to the load return flow in separate cores, are designed with greater than 50 NTU's and have less than 10 NTU's per meter of core length (9.3 for maximum capacity mode). This configuration is similar to that suggested in the 12 GeV cold box specification. The re-design also incorporated re-mixing headers at approximately mid-length. The re-commissioning in the fall of 2015 showed that even in the most demanding case of 4.5 K (pure) refrigeration, the re-designed heat exchanger performed as designed, achieving the design LN usage. For modes with some degree of flow imbalance, the LN usage was either the same as the design or somewhat lower. Table 1 shows some of the key results. The additional cost of using these heat exchangers in the original design phase would have been insignificant compared to the additional LN consumption cost.

NTU's can be thought of as describing the length of a heat exchanger and the net thermal rating, or (UA), can be thought of as describing the volume of the heat exchanger. This is intuitively plausible, and can be quantitatively justified as follows.

Table 1. Ratios of Test to TS Design

	2013 Test to TS Design		2015 Test to TS Design	
	Max. Capacity	Max. Refrig.	Max. Capacity	Max. Refrig.
Load exergy	0.95	0.99	1.01	1.02
Equivalent 4.5 K refrigeration	0.95	0.99	1.01	1.02
LN consumption rate	3.28	3.46	0.76	1.16
LN consumption per 1 kW of equivalent 4.5 K refrigeration	3.44	3.49	0.75	1.13

Recalling that the ratio of the Colburn 'j' factor (j_H) to the Fanning friction factor (f) is relatively invariant over a wide Reynold's number range, the heat transfer coefficient is, $h = \left(C_p \cdot G/Pr^{2/3}\right) \cdot (j_H/f) \cdot f$. Where, C_p is the specific heat (at constant pressure), G is the mass flux (i.e., mass flow per free flow area), and Pr is the Prandtl number. Recalling that for streams 'h' and 'l', the (UA), neglecting wall resistance, is, $(UA)^{-1} = \left\{(C \cdot N)_h{}^{-1} + (C \cdot N)_l{}^{-1}\right\}^{-1}$. Where, N_h and N_l are the stream specific NTU's, $(C \cdot N) = \eta_0 \cdot h \cdot A$, and, η_0 is the overall extended surface (i.e., fin) efficiency and A is the heat transfer area. Recalling the definition of the hydraulic radius, $r_h = A_c/(A/L)$, where A_c is the free flow (cross-sectional) area and L is the length; it can be seen that, N_h and N_l are proportional to $(f \cdot L)$. Taking the simplest case of, $C_h = C_l$, we see that, $NTU{\sim}L$. So, the stream temperature difference (which is the same for a balanced heat exchanger) is, $\Delta T_{hl} = \Delta T_{max}/(1 + NTU) \sim 1/L$. Where, ΔT_{max} is the difference between the stream inlet temperatures. For the net thermal rating, we have, $(UA) = \widetilde{U} \cdot (A_c \cdot L/r_h)$; where, $\widetilde{U} = (UA)/A$. So, $(UA){\sim}A_c \cdot L$, that is, the volume. However, the NTU's and (UA) are *not* geometric factors! So, using the empirical parameter of NTU's per core length is justified as an indicator of whether the heat exchanger is long enough.

Analysis of empirical data on brazed aluminum HX's by JLab, prior to deciding on a re-design solution, indicated that an important criteria was missing in the HX specification; that is, the aspect ratio [3]. This is defined as the ratio of effective length to the square root of the total free flow area. That is, even if the NTU's per core length requirement is acceptable, the HX may not be 'skinny' enough. It was found in examining the data that 300-80 K HX's which had an aspect ratio greater than five performed as designed. It should be kept in mind that the design of these HX's (presumably) included the additional (UA) required to compensate for axial conduction.

Brazed aluminum HX's are commonly employed in large plants, since they are cost effective, although they are vulnerable to improper flow distribution. This compounded with the challenging task of balancing the number of passes in multi-stream HX's for actual multi-mode operating (as opposed to design) and turn-down conditions, often results in these not performing as designed or anticipated. Although, this may not be commonly acknowledged, it is evident in the cold piping exiting CBX's (and, if a LN pre-cooler is used, higher than design LN consumption). It must be kept in mind by users specifying equipment and equipment designers/manufacturers that the 300 to 80 K helium-helium HX for the LN pre-cooler has a very demanding requirement, requiring greater than 40 NTU's for a reasonable LN consumption, and is very vulnerable to improper flow distribution [3, 7]. *It should be re-iterated that even with a net thermal rating margin of ~30%, less than 10 NTU's per meter of core length, adequate pressure drop ratios, and a simple two-stream HX using mid-length re-mix headers, the re-design just met the TS design for LN consumption in the maximum 4.5 K (pure) refrigeration mode.*

3. Wide Range Cold Box Testing

After the manufacturer installed the re-design, the 12 GeV refrigerator was re-commissioned. During this period, all design modes were tested and a wide capacity range of liquefaction, refrigeration and mixed liquefaction plus refrigeration (~50/50%) were tested. *In these tests the helium supply pressure* *to the CBX varied from 19.5 to 6.5 bar, without requiring the turbine inlet valves to be throttled for the entire pressure range,* except for maximum refrigeration without shield (1 out of 24 tests). Although, obviously there was operator interfacing (due to the nature of testing), the transition between different capacities and modes was very stable and required a minimum of operator interaction. In fact, even after re-commissioning, while supporting the LINAC at 2-K (i.e., cold compressor operation), the cold compressors did not trip when occasionally a turbine string would trip (this happened due to instrumentation issues and in one case due to an under sizing of T2's nozzle which caused a failure in T2, which was corrected). Nor did the cold compressors trip when one of the low pressure load return compressors tripped off. These are tangible proof of the robustness of the Floating – Pressure process working with a well matched compressor, 4.5 K CBX, and cold compressor system.

A particularly useful component implemented in the 12 GeV compressor system design was a bypass control valve between the recycle return (or MP) stream and the load return (or LP) stream; shown as "BYPML" in figure 1. This valve allows any excess LP stage capacity, which would otherwise be recirculated by the "BYPL" bypass (see figure 1), to be used to handle recycle load return flow. The exergy loss across this valve is small, since the MP and LP streams are usually close in pressure, and is outweighed by the increased pressure ratio across the upper two turbine strings and the inefficiency in implementing the compressor slide valve.

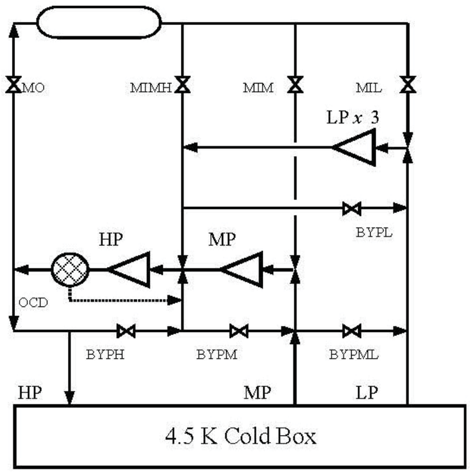

Figure 1. Simplified Process Flow Diagram of Compressor System

Preliminary data was presented previously [3]. Table 2 shows the tests performed during re-commissioning in the fall of 2015. As done during the commissioning in 2013 [2], test heaters were used to impose a shield load and the load from the cold compressors and allowed a precise load measurement. Testing of the cold compressor load did not require sub-atmospheric helium as used by others, and was quite similar to that used for commissioning at SNS [8]. Figure 2 shows the inverse coefficient of performance (COP; for the 4.5 K cold box system) and system exergetic efficiency vs. the equivalent 4.5 K refrigeration load. Figure 3 shows the total load exergy and the HP supply pressure to the CBX vs. the equivalent 4.5 K refrigeration load. In regards to the Floating-Pressure Process, note the direct correspondence in figure 3 between capacity and HP supply pressure. It should be clear from these figures that the 12 GeV refrigerator has a high performance over a very wide range of operation. Notwithstanding having good component efficiencies at design conditions, the primary factors allowing this is the implementation of the Floating Pressure Process, a compressor system/skid design allowing a wide range of operation [9], an equal 'Carnot-step' (expansion stage) CBX design [10], a variable turbine brake, proper HX specification/design [3] and proper sizing of internal piping/components.

Table 2. Tests Performed During Re-Commissioning

Mode	HP [bar]	R [kW]	L [g/s]	SH [kW]	CC [g/s]	E_L [kW]	q_L [kW]
Maximum Capacity	19.4	0.0	0	12.6	305.0	1282	17.8
Nominal Capacity	14.2	0.0	0	8.0	217.7	827	11.3
Liquefaction	17.8	0.0	189	8.0	0.0	1226	16.4
	16.3	0.0	177	0.0	0.0	1109	14.9
	12.8	0.0	122	0.0	0.0	733	10.0
	10.2	0.0	93	0.0	0.0	584	8.0
	8.7	0.0	75	0.0	0.0	466	6.3
	8.1	0.0	64	0.0	0.0	401	5.4
	7.6	0.0	51	0.0	0.0	317	4.3
	6.6	0.0	40	0.0	0.0	228	3.1
Refrigeration	10.7	11.4	0	12.5	0.0	860	12.3
	11.4	11.0	0	0.0	0.0	779	11.2 (‡)
	9.5	9.7	0	6.5	0.0	743	10.4
	8.4	8.0	0	6.5	0.0	620	8.6
	7.7	6.2	0	6.5	0.0	493	6.8
	6.6	4.3	0	6.3	0.0	355	4.9
	6.3	3.8	0	6.5	0.0	329	4.5
	7.2	3.3	0	12.5	0.0	304	4.2
	7.0	2.9	0	6.5	0.0	235	3.2
50% L + 50% R	12.6	7.4	78	0.0	0.0	1046	14.7
	11.6	6.4	67	0.0	0.0	909	12.6
	11.3	5.4	60	0.0	0.0	793	11.0
	9.6	3.5	49	0.0	0.0	586	8.0
	6.3	2.7	22	0.0	0.0	347	4.7

Notes:	(‡)	Max. refrigeration without shield; T5-T6 turbine string inlet valve throttled due to cold outlet temperature
Abbreviations:	HP	HP - High pressure supply pressure to cold box
	R	4.5 K refrigeration load
	L	4.5 K (to 300 K) liquefaction load (make-up)
	SH	Nominal 35-55 K shield load supplied by T1-T2 turbine string
	CC	Cold compressor flow (injected into cold box at < 30 K)
	E_L	Total load exergy
	q_L	Equivalent 4.5 K refrigeration (3 bar 4.6 K supply, 1.25 bar return)

For this data, the LP compressor slide valve was adjusted so that there was no bypass. Of course, in actual operation, some modest amount of bypass is allowed (say ~5%), so that if a MP or LP stage compressor trips, the cold compressors can remain operating as operation crews respond and restart the compressor or a back-up. This 'adjustment' was made based upon test data taken from a test of the LP stage over a range of slide valve positions and discharge pressures. Figure 4 shows the three dimensional characterization of this test data for the volumetric and isothermal efficiencies. The vertical axes in these plots is the ratio of the measured (part load) efficiency to the predicted efficiency at a fully loaded condition. Using a variable frequency drive (VFD) is an alternative to the slide valve, as the cost of these even for larger machines is becoming more practical. However, the 12 GeV compressor system was not designed with VFD's (for the compressor motors) and given that the efficiency of rotary screw compressors is highly dependent on their tip speed [11], it is uncertain whether the VFD or slide valve would be more efficient and/or reliable.

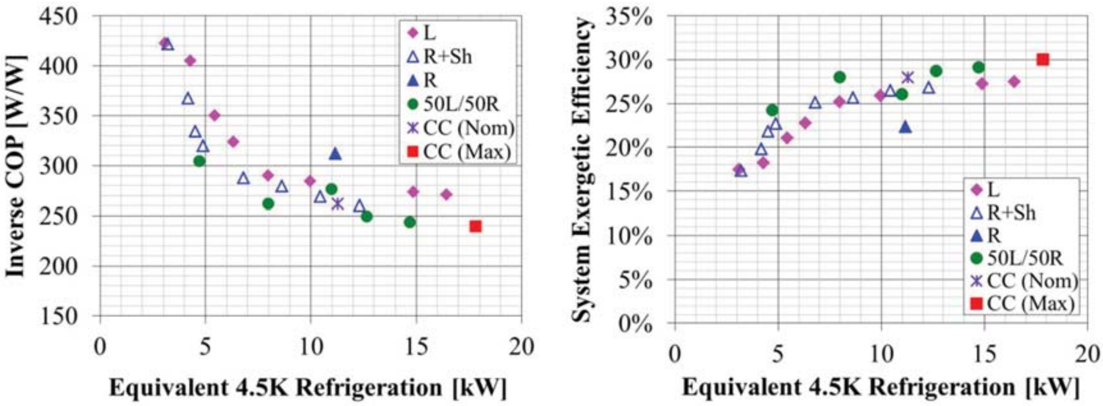

Figure 2. Inverse COP and System Exergetic Efficiency vs. Equiv. 4.5 K Refrigeration Load

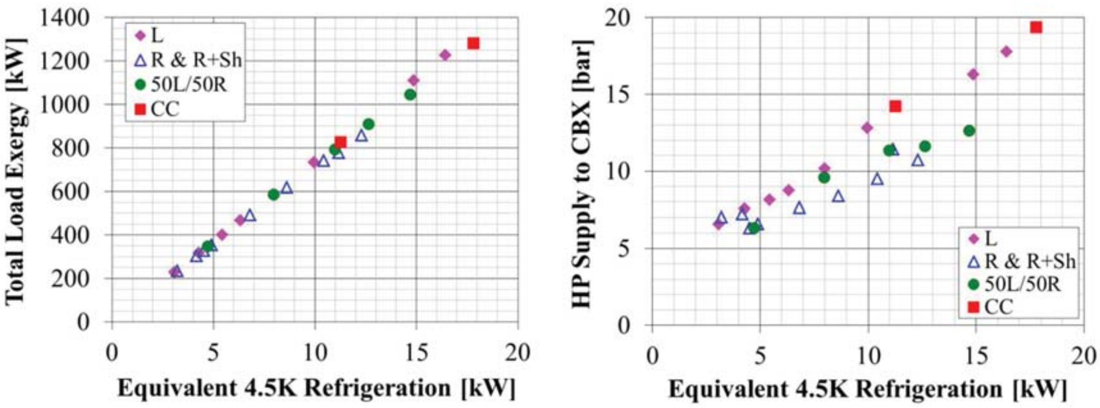

Figure 3. Total Load Exergy and HP Supply Pressure to CBX vs. Equiv. 4.5 K Refrigeration Load

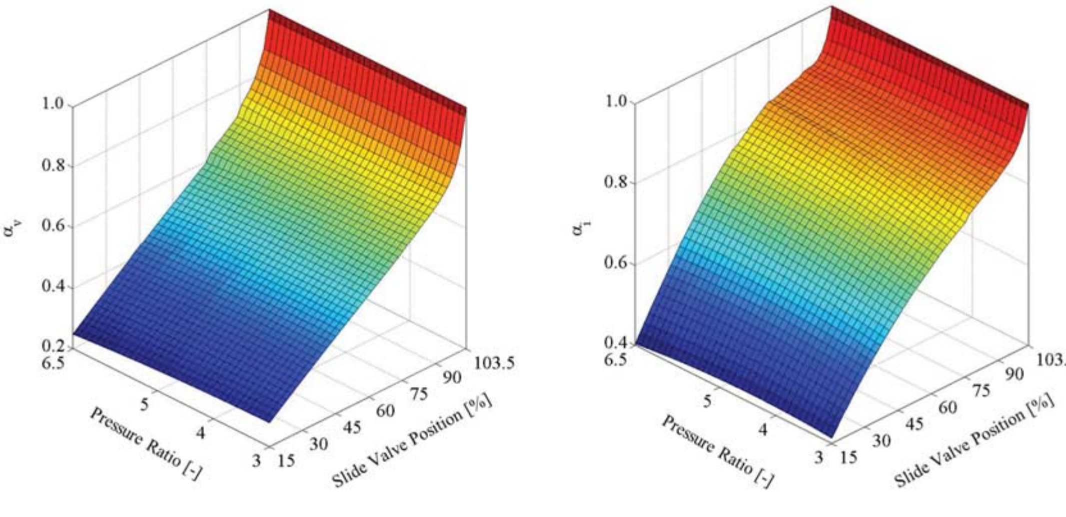

Figure 4. LP Stage Volumetric and Isothermal Efficiency vs. Pressure Ratio
and Slide Valve Position

In general, inverse process modeling of the system indicated the venturi flow meters at the interface between the compressor and cold box system, and the CBX venturi meters at the inlet to each turbine string had reasonable corroboration. Many cross-checks were performed with an uncertainty analysis for the flow meters. The inverse process modeling also indicated that the turbine adiabatic efficiencies are equal to the design or higher than expected. Over the entire range of conditions tested, the turbine strings (i.e., T1-T2, T3-T4, T5-T6, and T7) are operating at an average adiabatic efficiency of 83.5 to 86.7% with a standard deviation of about 1.4%.

4. Pulsed Load Test

Near the completion of the re-commissioning tests, a pulsed load test was conducted using heaters in the 3 m^3 sub-cooler within the 'lower' CBX and the external 10 m^3 dewar. Three tests were conducted as summarized in Table 3. The intent was to impose a pulsed load similarly scaled to that expected to be imposed on the ITER cryo-plants [12]. That is, a ratio of the peak-to-peak to mean heat load of ~30%, a period of 1800 seconds and a ramp rate of 30 W/s. *It is important to note that no special or additional instrumentation, controls, algorithms or equipment was implemented for this pulsed load* [13, 14]; that is, the same Floating – Pressure process and system was used throughout all testing (and normal operation). During this testing, the compressor bypass valves did not open and there was no mass in or mass out (to gas storage). Figure 5 shows the heat load pulse profile, the HP supply pressure response to the CBX, the LP load return pressure from the CBX and the supply mass flow to the CBX for test #2. The behavior is quite similar for the other tests. The top two curves belong with the left hand vertical axis and the bottom two curves belong with the right hand vertical axis.

Table 3. Pulsed Load Test Summary

	Test	*1*	*2*	*3*
4.5 K Heat Load (§)	[kW]	9.64	9.73	9.63
4.5 K Liquefaction Load	[g/s]	0	0	13.3
Shield Load	[kW]	0	7.83	0
HP Supply Pressure to CBX	[bar]	9.06	9.92	9.67
Ratio of Peak-to-Peak Heat Load to Average (§)	[-]	32.3%	32.8%	32.4%
Ratio of Peak-to-Peak HP Supply Pressure to Average	[-]	5.4%	7.6%	6.5%
Ratio of Peak-to-Peak LP Return Pressure to Average	[-]	4.1%	4.2%	3.4%
Ratio of Peak-to-Peak HP Supply Mass Flow to Average	[-]	12.3%	12.3%	11.1%

Notes: (§) Integrated average

It should be kept in mind that although one of the design modes for the 12 GeV refrigerator was a maximum 4.5 K pure refrigeration load, it was not primarily designed as a refrigerator; rather this mode was imposed to ensure 'well-rounded' HX's. It was primarily designed for the cold compressor load with a modest amount of 4.5 K (to 300 K) liquefaction. Consequently, if it were designed for a pure refrigeration load, the nozzle coefficients and the low pressure load return compressor displacement would have been selected differently. Also, if the objective is to prevent any variation in the load return pressure, the displacement of the low pressure load return compressors could be sized so that the compressor bypass to the load return can keep this pressure from varying. This pulsed load refrigeration test not only demonstrated the capability of a well-balanced refrigerator system design (compressors and cold box), but also the effectiveness of the Floating – Pressure Process.

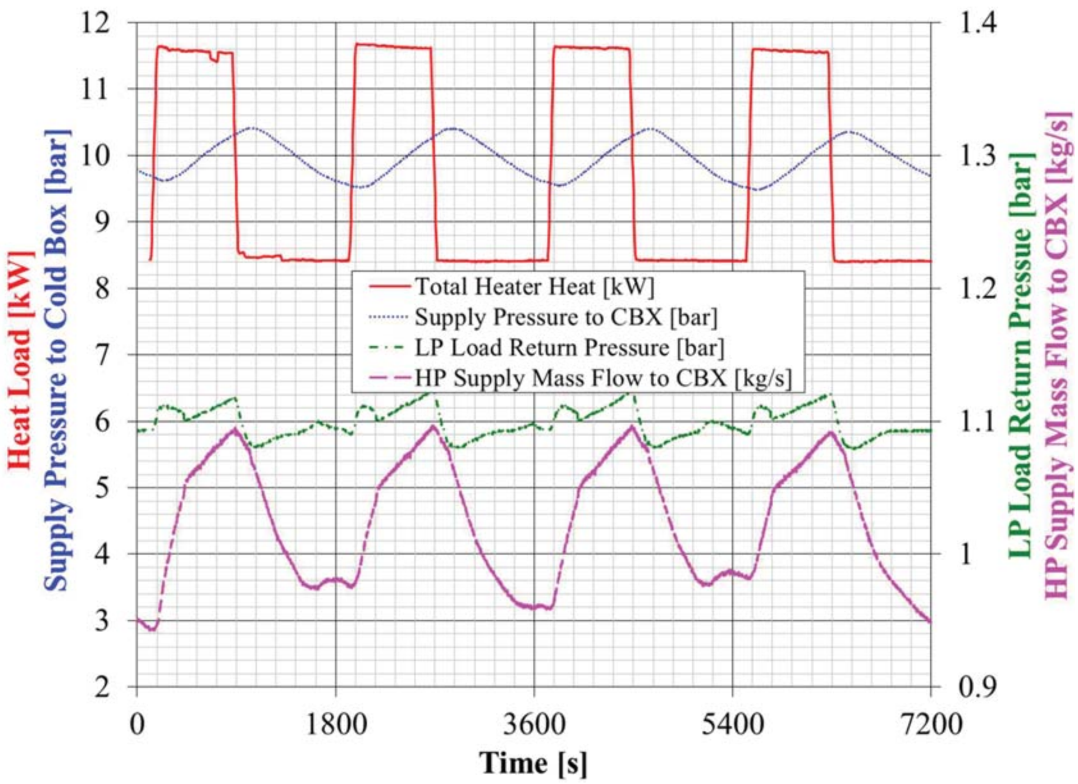

Figure 5. Pulsed Heat Load and System Response

5. Conclusion

These test results and other accomplished elsewhere [15] have demonstrated the robustness and effectiveness of the Floating – Pressure process. The advances gained at JLab over the past 15 years in the design of cryogenic systems have culminated in the 12 GeV cryogenic system. There is still great opportunity for advancement as there are many areas needing further work and development. The FRIB project at MSU and the LCLS-II project at SLAC are directly benefiting from the advances accomplished previously [15] and from the 12 GeV cryogenic system. It is extremely fortunate for highly schedule driven projects like LCLS-II that the time, money and effort was expended to develop the 12 GeV cryogenic system. Unfortunately, projects like this are incapable of advancing the state of the art of cryogenic systems given their schedule, and as it is too often regarded as 'established technology'. It is hoped that this paradigm can be abandoned and laboratories take the initiative to further advance cryogenic systems in partnership with industry, being encouraged and afforded this investment by their government sponsors.

6. Acknowledgements

The authors would like to express their appreciation and thanks to the TJNAF management and to their colleagues for their support. This work was supported by the U.S. Department of Energy under contract no. DE-AC05-06OR23177.

References

[1] Arenius D, Creel J, Dixon K, Ganni V, Knudsen P, Sidi-Yekhlef A, Wright M 2008 An overview of the planned Jefferson Lab 12 GeV helium refrigerator upgrade *Adv. Cryo. Eng.* **53** (New York: American Institute of Physics) pp 588-593

[2] Ganni V, Knudsen P, Dixon K, Norton R, Creel J, Arenius D 2014 Commissioning of helium refrigeration system at JLab for the 12 GeV upgrade *Adv. Cryo. Eng.* **59** (New York: American Institute of Physics) pp 944-948

[3] Knudsen P, Ganni V, Dixon K, Norton R, Creel J 2015 Commissioning of helium refrigeration system at JLab for the 12 GeV upgrade *IOP Conf. Series: Materials Science and Engineering* **101**

[4] Ganni V, Knudsen P 2010 Optimal design and operation of helium refrigeration systems using the Ganni cycle *Adv. Cryo. Eng.* **55** (New York: American Institute of Physics) pp 1057-1071

[5] Ganni V *Design of Optimal Helium Refrigeration and Liquefaction Systems* (CSA Short Course Symposium, Tucson, Arizona, June 28 – July 2, 2009)

[6] Knudsen P, Ganni V 2010 Helium refrigerator liquid nitrogen pre-cooler component parameter sensitivity analysis *Adv. Cryo. Eng.* **55** (New York: American Institute of Physics) pp 215-222

[7] Fleming R 1967 The effect of flow distribution in parallel channels of counterflow heat exchangers *Adv. Cryo. Eng.* **12** (New York: Plenum Press) pp 352-362

[8] Arenius D, Chronis W, Creel J, Dixon K, Ganni V, Knudsen P 2004 Cryogenic system or the Spallation Neutron Source *Adv. Cryo. Eng.* **49** (New York: American Institute of Physics) pp 200-207

[9] Knudsen P, Ganni V, Dixon K, Norton R, Creel J 2015 Commissioning and operational results of the 12 GeV helium compression system at JLab *IOP Conf. Series: Materials Science and Engineering* **101**

[10] Knudsen P, Ganni V 2006 Simplified helium refrigeration cycle analysis using the 'Carnot Step' *Adv. Cryo. Eng.* **51** (New York: American Institute of Physics) pp 1977-1986

[11] Persson J 1990 Performace mapping vs design parameters for screw compressors and other displacement compressor types *VDI-Berichte* **859** pp 15-31

[12] Sanmarti M 2008 ITER Cryogenic System *ITER CODAC Colloquium (Barcelona, Spain, 27-28 October 2008)*

[13] Clavel F, Alamir M, Bonnay P, Barraud A, Bornard G, Deschildre C 2011 Multivariable control architecture for a cryogenic test facility under high pulsed loads: model derivation, control design and experimental validation *J. Process Control* **21(7)** pp 1030-1039

[14] Booth W, Bradu B, Blanco E, Gayet P, Maekawa R, Serio L, Chang H, Chalifour M 2012 Dynamic simulation of the ITER helium cryogenic system under pulsed heat loads *Proc. 24th International Cryogenic Engineering Conference (Fukuoka, Japan, 14-18 May 2012)* pp 595-598

[15] Homan J, Redman R, Ganni V, Sidi-Yekhlef A, Knudsen P, Norton R, Lauterbach J, Linza R, Vargas G 2014 Commissioning of a 20 K helium refrigeration system for NASA-JSC chamber-A *Adv. Cryo. Eng.* **59** (New York: American Institute of Physics) pp 307-314