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Cold compressor performance and energy consumption improvements at Jefferson Lab's Central Helium Liquefiers

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Abstract. Jefferson Lab operates two central helium liquefiers (CHLs) which both utilize full cold compression from the saturation pressure at operating temperature (approximately 2.1 K) to just over atmospheric pressure. The original plant, CHL1, was recently outfitted with a replacement subatmospheric cold box (SC1R) containing state-of-the-art cold compressor technology, while the newer plant, CHL2, uses an older cold compressor system. In both cases, the heat of compression is absorbed at low temperature at the expense of electrical power consumed by the warm compressors. Due to the superior efficiency and turndown capabilities of SC1R, a new operating mode has been identified for CHL1 in which the required number of operating warm compressors is reduced by one. A cost-based method for optimizing cold compressor stability and efficiency has been developed and applied to CHL2, improving its turndown and lowering the warm compressor discharge pressure. As a result of these efforts, power consumption of the combined CHLs during normal operations has been reduced by nearly 10%, or a total of 650 kW. The observed performance of CHL1 with SC1R, as well as the cold compressor optimization method, leading to this improved energy consumption rate will be discussed in detail.

1. Introduction

Thomas Jefferson National Accelerator Facility (Jefferson Lab) is home to the Continuous Electron Beam Accelerator Facility (CEBAF), which is comprised of two anti-parallel linear accelerators (LINACs) joined by semi-circular magnetic steering arcs. It was originally designed as a 4 GeV machine, but the beam energy was increased to 6 GeV and then doubled to 12 GeV [1]. Each LINAC is formed from a string of cryomodules containing niobium superconducting radio frequency cavities cooled to just below 2.1 K by saturated liquid helium. Heat shields inside the cryomodules are actively cooled to about 35 K. Four experimental halls located off the ends of the LINACs contain beam targets and particle detectors which also require cryogenic cooling to a variety of temperatures in the range of 4.5-80 K.

During the 6 GeV era (and earlier), cryogenic needs of both LINACs were met by the original Central Helium Liquefier (CHL1), which was designed and constructed alongside CEBAF [2]. Doubling the beam energy to 12 GeV roughly doubled the cryogenic load, resulting in the construction of a second Central Helium Liquefier (CHL2) [3]. Each CHL supports one LINAC during 12 GeV beam operations. Both are arranged in generally the same manner, coupling a conventional helium refrigerator with a separate subatmospheric or 2 K cold box, as illustrated in

Figure 1. The conventional portion includes several oil-flooded screw compressors and a main cold box, and produces 35 K helium for the cryomodule shields as well as a 4.5 K subcooled liquid helium primary supply stream. A small portion of this stream can be transported to the End Station Refrigerator (ESR) complex via vacuum jacketed transfer line and used to supplement the capacity of the stand-alone refrigerators that supply cryogens to the experimental halls.

Most of the primary supply is delivered to and returned from the LINAC via the subatmospheric cold box, which contains five cryogenic centrifugal (cold) compressors in series that generate and maintain the subatmospheric vapor pressure (0.0385 atm) required for nominally 2 K saturation conditions in the cryomodules. Each cold compressor is powered by its own motor. The rotational speed of one compressor is adjusted to keep the load return mass flow rate constant, and the speeds of the others are adjusted proportionally through fixed gear ratios.

CHL1 and CHL2 each have the equivalent of about 18 kW of refrigeration capacity at the normal boiling point. This capacity is designed to be distributed among three loads: up to 4.8 kW at 2.0 K for the primary load, up to 12 kW at 35 K for the shield load, and up to 10 g/s (CHL1) or 20 g/s (CHL2) for liquefaction at 4.5 K. If the primary load is taken as a combination of the LINAC and 2 K cold box, as laid out in Figure 1, then its exergy usage rate is about ten times higher than that of the other loads, and it is advantageous in terms of overall power consumption to reduce it to the greatest extent possible. This can be achieved by lowering the cold compressor mass flow rate and improving the cold compressor train efficiency. The former is easier to achieve in practice, and has a more significant effect.

Neither CHL typically operates at full load. Shield loads are essentially fixed by the geometry of the thermal shields, and liquefaction capacity is used only when required to fill cryomodules or support the ESR complex. Cold compressor mass flow is reduced as much as possible without introducing excessive risk of surge, a catastrophic flow reversal which requires the cold compressors to immediately shut down. Minimum flow rates are still higher than required based on cavity heat loads alone, and supplementary electric heat is added to the cryomodules to maintain enough flow for stability. However, recent work has led to reductions in the minimum attainable flow rates for both CHL1 and CHL2, and cold compressor efficiency has improved as well. Impact on the cryogenic system energy consumption rate is significant. At CHL1, an entirely new subatmospheric cold box featuring highly efficient modern cold compressor technology with substantial turndown has been installed. It affords a reduction in power consumption of 540 kW as will be discussed in Section 2. At CHL2, which is still using older cold compressor technology, an additional power reduction of 110 kW has been achieved using a cost-based optimization

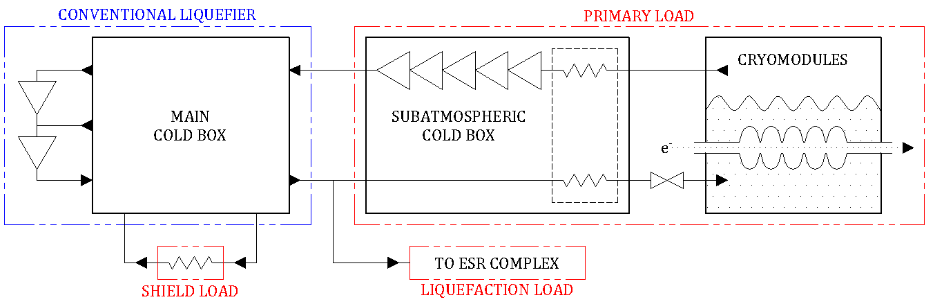

Figure 1. Basic overview of CHL1 and CHL2 depicting the warm compressor system and main cold box of the conventional refrigerator and the three loads that they are designed to carry.

routine to select gear ratios which represent the best combination of low-flow stability and polytropic efficiency. This method will be discussed in Section 3. In Section 4 the accomplishments will be summarized.

2. CHL1

Early in the 12 GeV era, CHL1 was coupled to an older 2 K cold box referred to as SCM, which contained obsolete cold compressor technology that resided completely inside of the insulating vacuum shell and had a minimum stable flow rate of about 185 g/s. A replacement called SC1R, which utilizes modern cold compressors mounted outside of the cold box where they can be serviced without breaking insulating vacuum, was recently designed, fabricated, installed, and commissioned by Jefferson Lab [4]. While serviceability was the primary motivation for SC1R, observations during the initial period of operations show that it substantially reduces the primary load on CHL1. Its design minimum flow rate is 170 g/s, which is well matched to the 12 GeV LINAC heat load and has been selected as the normal operating point. Furthermore, commissioning data, shown in the left panel of Figure 2, reveals that the primary load exergy usage rate is about 8% lower for all flow rates due to improved efficiency of the new cold compressors.

One effect—among many—of the newly decreased load is a corresponding decrease in thermodynamic availability required at the main cold box high pressure supply stream. Supply pressure is fixed by the turboexpander static gas bearing requirements, and temperature is fixed by the main compressor cooling water system controls. By extension, supply stream specific exergy is fixed, and it is the main compressor system mass flow rate that responds to the reduced input exergy demand. This effect is readily apparent from the right panel of Figure 2, which shows the measured mass flow rate at the main cold box high pressure inlet for the one-week period beginning on July 26, 2020, while SCM was still in use, as well as for the one-week period beginning on December 3, 2023, while SC1R was in use. Average values for these time periods are

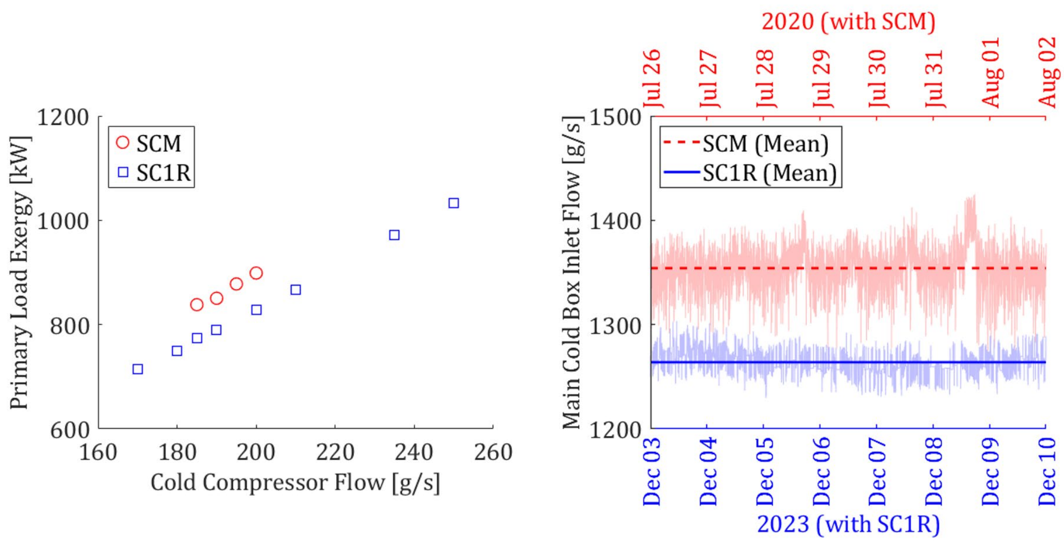

Figure 2. Left: Measured primary load exergy usage rate for the old (SCM) and new (SC1R) 2 K cold boxes at a number of cold compressor mass flow rates. Right: Main cold box high pressure inlet mass flow rate with SCM (dashed line indicates mean) and SC1R (solid line indicates mean).

shown with bold lines over top of the raw data streams; the reduction was 90 g/s, from 1,354 g/s with SCM to 1,264 g/s with SC1R.

Historically, the main compressor lineup at CHL1 included two first stage machines and two and a half second stage machines (which might be achieved in practice by a primary compressor at 50% load or the standby compressor, whose throughput is about half that of a primary machine). Interstage pressure is allowed to float in response to changing flow demand, but was typically kept around 2.5 atm. An interstage pressure of 2.6 atm satisfied the main cold box high pressure flow demand with SCM in use. Allowing interstage pressure to float slightly above 3 atm increases the second stage suction density so that two primary compressors can satisfy the reduced SC1R flow demand without the extra standby (or partially loaded primary) compressor.

Operation of CHL1 with only two second stage main compressors was briefly tested in November 2023, and adopted as the normal configuration shortly thereafter. Interstage pressure settled at an average value of 3.04 atm, as shown in the left panel of Figure 3. More importantly, as seen in the right panel, shutting down the extra second stage compressor reduced the electrical power drawn by CHL1 during normal operations from 4,025 kW to 3,483 kW.

The benefit of SC1R's turndown capability and superior efficiency is a net CHL1 power reduction of about 540 kW, more than 13% of its previous steady-state draw. However, elevated interstage pressure limits the ability to keep up with periods of increased loading. Testing has shown that short term events (on the order of hours), such as filling a cryomodule, are handled without issue. Longer term load increases (on the order of days), such as supporting the ESR complex by reintroducing the liquefaction load, still require an additional second stage compressor. Such events are not anticipated on a regular basis, so future operations of CHL1 are expected to take full advantage of the energy saving benefits delivered by SC1R.

3. CHL2

CHL2 was designed around the Ganni (Floating Pressure) Cycle, a constant pressure ratio cycle which allows the system to automatically turn down at constant Carnot efficiency when its load

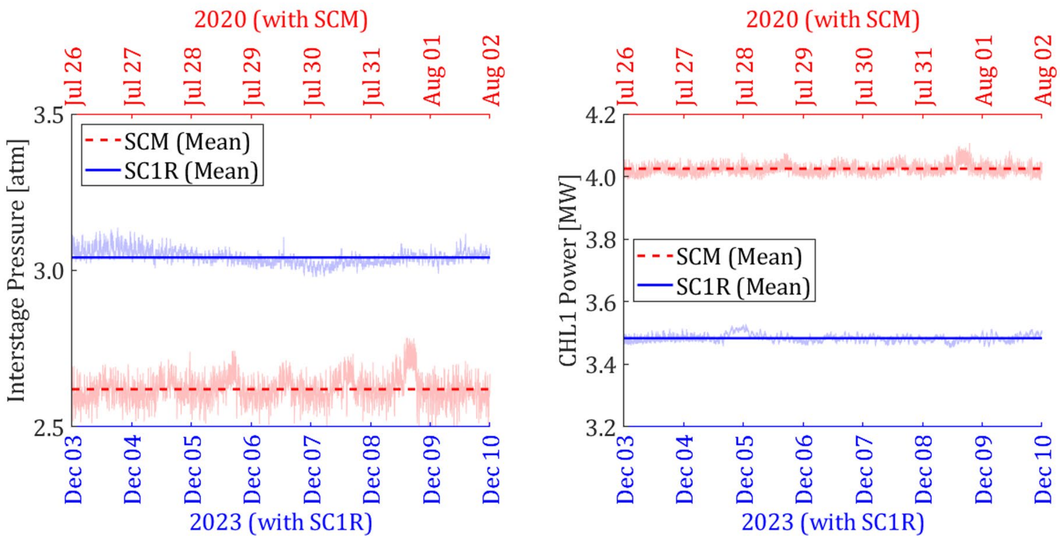

Figure 3. Main compressor system interstage pressure (left) and total CHL1 power (right) for SCM (dashed line indicates mean) and SC1R (solid line indicates mean).

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is decreased [5]. This technology would naturally reduce electrical power consumption in response to a high turndown, high efficiency cold box like SC1R, but CHL2 is currently using an older 2 K cold box, referred to as SCN, which is identical to SCM and was originally constructed to provide redundancy for CHL1 [6]. To better capitalize on the capabilities of CHL2, the SCN cold compressor speed ratios have been optimized for low flow stability, allowing the system to operate below 185 g/s with suitable surge margin.

If cold compressor performance laws and characteristics of the cryogenic distribution system are known, then the thermodynamic state of the entire cold compressor train may be determined from only the mass flow rate $\dot{m}$ and the rotational speed N of each compressor. The empirical relationships $P_1, T_1 = f(P_{sat}, \dot{m})$ were generated from data obtained during SCN commissioning to relate the first stage inlet pressure P_1 and temperature T_1 to the LINAC helium saturation pressure P_{sat} and the return mass flow rate [7]. Since P_{sat} is fixed at 0.0385 atm for 12 GeV operations, only the mass flow rate needs to be specified. Upon selecting a rotational speed N_1, the outlet pressure P_2 may be readily obtained from performance law data provided by the manufacturer in the form $P_{i+1}/P_i = f\left(X_i, N_{u,i}\right)$, since the reduced flow $X_i = f(\dot{m}, P_i, T_i)$ and reduced speed $N_{u,i} = f(T_i, N_i)$ are functions of known or specified parameters. The outlet temperature $T_2 = f\left(P_1, T_1, P_2, \eta_{p,1}\right)$ requires additional knowledge of the polytropic efficiency $\eta_{p,1}$, which is determined from the specific speed $N_{s,1} = f(\dot{m}, P_1, T_1, N_1, P_2)$ using empirical performance laws of the form $\eta_{p,i} = f\left(N_{s,i}\right)$ that were crafted for each SCN cold compressor from the commissioning data [7]. Once the outlet conditions P_2 and T_2 for the first stage cold compressor are obtained, they may be used as inlet conditions for the next stage, and so on, until the thermodynamic conditions P_6 and T_6 at the outlet of the fifth compressor are known.

Optimizing compressor speeds for a given mass flow rate begins by specifying an acceptable range of speeds for each stage, and then determining the thermodynamic state of the system for all possible combinations. In the optimization exercises carried out for SCN, this typically results in 0.5-2 million operating points from which to choose. A number of them may be immediately discarded because they result in surge or choking of one or more stages. Each of the remaining options is assigned a cost, and the one with the lowest cost is selected as optimal.

Costs for each operating point are determined by a main cost function Ξ, whose independent variables may be any calculable quantity pertaining to the performance of the cold compressors, and whose form is selected such that its minimum value occurs at the most desirable combination of those variables. For the optimization of SCN, a main cost function of the form

$$\Xi = \sum_{i=1}^{5} W_{N,i}\xi_{N,i} + \sum_{i=1}^{5} W_{M,i}\xi_{M,i} + W_T\xi_T + W_\sigma\xi_\sigma \tag{1}$$

was found to be suitable. This function simultaneously optimizes for 12 parameters: rotational speed of each compressor (contained within the first summation), surge margin for each compressor (contained within the second summation), final discharge temperature, and standard deviation of the surge margins. Importance of the compressor speeds against the other parameters may be adjusted by the weighting factor W_N; similarly, W_M weights surge margins, W_T the outlet temperature, and W_σ the surge margin standard deviation.

Turning the compressors at higher speed increases stress on the analog active magnetic bearing system, and drives motor torques toward their upper limits. Therefore a parameter cost function $\xi_{N,i}$ related to the speed of compressor i has been defined as

$$\xi_{N,i} = \frac{N_i - N_{i,min}}{N_{i,max} - N_{i,min}}, \tag{2}$$

where the subscripts max and min denote the fastest and slowest speed in the range under consideration. This creates a normalized linear relationship between the compressor speed and the cost associated with operating at that speed, as depicted in the left panel of Figure 4. Normalization of each parameter cost function ξ on a scale of 0 to 1 ensures that scaling is consistent across the main cost function, so that the weighting factors W have a meaningful effect.

Maximizing surge margins is particularly important because risk of surge at low flow rates is typically what limits turndown. Margin M_i for compressor i has been non-traditionally defined as

$$M_i = \begin{cases} \dfrac{X_i - X_{i,P}}{X_{i,P} - X_{i,S}} & \text{if } X_i \leq X_{i,P} \\ \dfrac{X_i - X_{i,P}}{X_{i,C} - X_{i,P}} & \text{if } X_i > X_{i,P} \end{cases}, \tag{3}$$

where the subscripts P, S, and C refer to the reduced flow on the peak efficiency line, surge line, and choke line, respectively, at the pressure ratio of the operating point under consideration. This creates a linear relationship in the range of -1 to 0 at constant pressure ratio between the surge and peak efficiency lines, and between 0 and 1 at constant pressure ratio between the peak efficiency and choke lines. A cost function $\xi_{M,i}$ based on surge margin has been selected as

$$\xi_{M,i} = \begin{cases} 1 - \mathrm{e}^{-\left(\frac{M_i}{0.3}\right)^2} & \text{if } X_i \leq X_{i,P} \\ 1 - \mathrm{e}^{-\left(\frac{M_i}{0.4}\right)^2} & \text{if } X_i > X_{i,P} \end{cases} \tag{4}$$

and is shown graphically in the center panel of Figure 4. The lowest cost is achieved by operating on the peak efficiency line. Surge is one of the leading causes of cold compressor downtime, while choking rarely, if ever, occurs in the normal operating flow range. Therefore, deviations toward surge conditions are penalized more aggressively than deviations toward choked conditions.

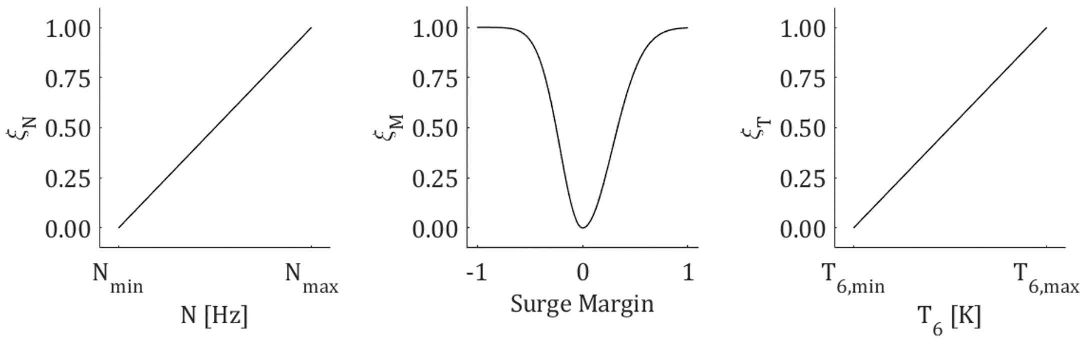

Figure 4. Parameter cost functions used to optimize the cold compressor operating point for rotational speed (left), surge margin (center), and discharge temperature (right), which represents efficiency.

Discharge temperature from the final compressor, T_6, is used to represent the efficiency of the operating point under consideration. Its cost function ξ_T is simply

$$\xi_T = \frac{T_6 - T_{6,min}}{T_{6,max} - T_{6,min}}, \tag{5}$$

which is a linear map in the range of $T_{6,min}$, the discharge temperature of the most efficient option, to $T_{6,max}$, the discharge temperature of the least efficient option, normalized as shown graphically in the right panel of Figure 4.

The final parameter cost function ξ_σ is simply the surge margin standard deviation σ_M computed for the five compressors at the operating point under consideration. Scaling with respect to the other parameter cost functions is preserved by the chosen definition for surge margin, since $0 \leq \sigma \leq 1$ for a set with range of -1 to 1. In some solutions, pushing one compressor onto its surge line allows the others to run very close to their peak efficiency lines, resulting in a low overall cost; by favoring small values of σ_M, such solutions are discredited.

This optimization method was implemented with Matlab and used to obtain speed ratios for SCN at a number of flow rates lower than its typical setting of 186 g/s. The lowest flow rate with all calculated surge margins equal to or greater than -30% was 178 g/s, so that was selected as the target. Gear ratio and flow adjustments were applied on November 29, 2023; Figure 5 shows that impact on the performance of CHL2 was significant. Speeds were adjusted to the new ratios prior to noon, with flow remaining at 186 g/s. Cold compressor efficiency improved during this process and discharge temperature fell from 25.8 K to 24.7 K, as seen in the top panel. A small corresponding decrease in the warm compressor power consumption is visible in the bottom

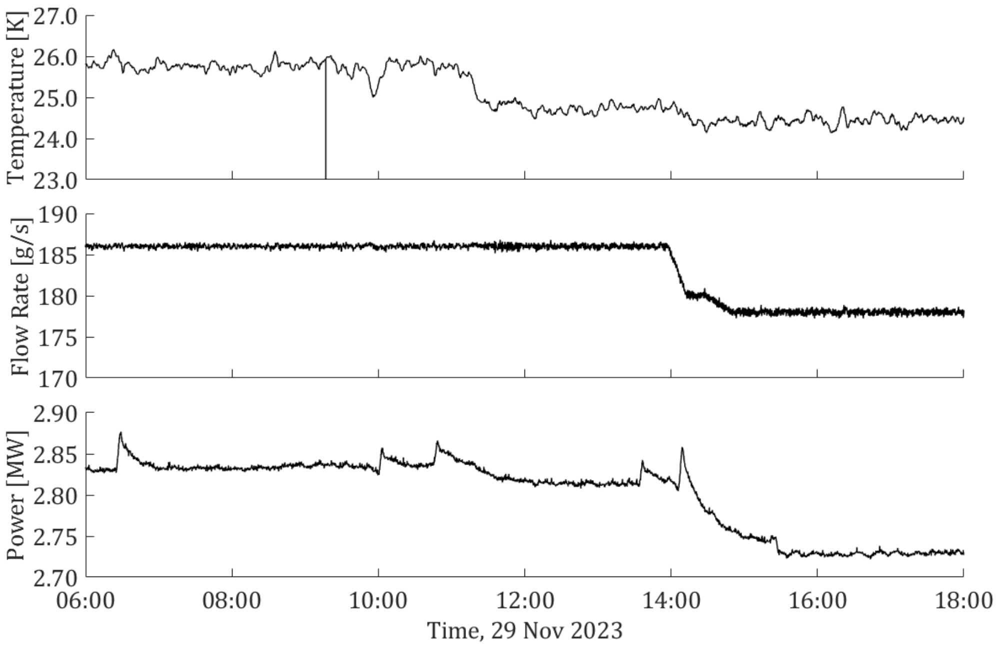

Figure 5. Effect of optimized SCN cold compressor speed ratios on the final discharge temperature (top), minimum stable mass flow rate (center), and CHL2 main compressor power consumption (bottom).

panel. Flow was decreased to 178 g/s after noon, as shown in the center panel. Discharge temperature was further reduced to 24.4 K, which was to be expected upon arriving at the operating point around which the system was optimized. Power consumption was significantly lowered in response to the flow rate reduction, as can be seen in the bottom panel, reinforcing the notion that lowering cold compressor flow rate is more impactful than raising efficiency.

At CHL2, the warm compressor lineup includes three load compressors, which process flow returning from the primary load (as well as exhaust flow from one stage of main cold box turboexpanders). Reducing the cold compressor flow rate creates room to unload a load compressor, the results of which are seen in the final step reduction in power consumption in the bottom panel of Figure 5. After this final adjustment, measured steady-state power consumption is 2,725 kW, down from the typical 12 GeV value of 2,835 kW. Optimizing the cold compressors for low-flow stability earned a reduction of 110 kW in the energy consumption rate for CHL2.

4. Conclusions

Improving cold compressor performance at Jefferson Lab's CHLs has reduced their energy consumption by a combined 650 kW, nearly 10% of the former 12 GeV era consumption. At CHL1, the inherent ability of the new SC1R cold compressor system to turn down to 170 g/s, combined with an 8% primary load exergy usage rate reduction compared to the old system, lowers the main cold box high pressure inlet flow requirement to a point where one less second stage compressor is needed to sustain 2 K operations. Shutting down the extra compressor reduces CHL1 power consumption by 540 kW. Operation of the old style cold compressors still in use at CHL2 has been optimized using a cost-based method that prioritizes high margin to surge at low flow rates, while simultaneously taking efficiency and rotational speed into account. A new minimum flow rate of 178 g/s has been attained, and floating pressure technology has lowered CHL2 power consumption by 110 kW in response.

Acknowledgments

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References

[1] Adderley P A *et al.* 2024 *Phys. Rev. Accel. Beams* **27** 084802
[2] Chronis W C, Arenius D M, Bevins B S, Ganni V, Kashy D H, Keesee M M, Reid T R and Wilson J D 1996 *Adv. Cryo Eng. A* **41** 641
[3] Ganni V, Knudsen P, Dixon K, Norton R, Creel J and Arenius D 2014 *AIP Conf. Proc.* **1573** 944
[4] Mastracci B, Yang S, Creel J, Dixon K, Norton R, Thompson S, Wieliczko J and Wijeratne T 2022 *IOP Conf. Ser.: Mater. Sci. Eng.* **1240** 012069
[5] Ganni V and Knudsen P 2010 *AIP Conf. Proc.* **613** 1057
[6] Ganni V, Arenius D M, Bevins B S, Chronis W C, Creel J D and Wilson J D 2002 *AIP Conf. Proc.* **613** 288
[7] Wilson J D 2003 *MSc Thesis* Christopher Newport University