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Commissioning Results of the ESR2 Compressors at JLab

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Abstract. The future operation of the 4 kW 15 Kelvin MOLLER experiment at Jefferson lab necessitates an increase of cryogenic capacity at the End Station Refrigerator. The current plant is the former 1.5 kW (4.5 K) ESCAR plant that has been operating at Jefferson Lab since 1995. The existing 1.5 kW plant is not able to support the load for MOLLER and will be replaced with a refurbished plant comprised of the cold box and compressors of the 4 kW ASST-A plant from the Superconducting Super Collider in Texas. This paper reports the commissioning results of the warm compressor system. We compare the compressors' current and past performance, specifically examining their Isothermal and Volumetric efficiency. The results indicate that the refurbished equipment's efficiency is comparable to its original performance, proving that the reuse and refurbishment of old systems is a worthwhile endeavor.

1. Introduction

The End Station Refrigerator 2 (ESR2) is the eventual replacement for the ESR1 at Jefferson Lab [1]. This paper describes commissioning the warm gas management and compressors for the ESR2 plant. The ESR2 compressor system consists of 4 refurbished Sulair compressor skids previously operated at the Superconducting Super Collider Lab (SSCL): 2 First stage compressors (displacement rate: 807 L/s @ 3600 RPM, motor power rating: 250 hp) and 2 Second Stage compressors (displacement rate: 595 L/s @ 3600 RPM, motor power rating: 750 hp). Each compressor is equipped with local oil separation and dedicated heat exchangers for cooling the helium and oil. A central cooling water system provides the coolant supply to the skids and maintains a water temperature near 300 K. Compressor Commissioning at ESR2 was performed in January of 2024. Testing consisted of initial calibration runs at low power and included adjustment of all associated subsystems: electrical, controls, cooling water, instrument air, helium gas management, and individual compressor skids. During each phase of commissioning, cooling water adjustments and gas management valve tuning were performed while the compressors were running. The end goal of the compressor system commissioning was to operate all compressors at several predetermined operating points from minimum and maximum capacity and compare the data gathered to previous literature published on these compressors when they were commissioned at SSCL in the early 1990s [2]. Additionally, a 24-hour run of each pair of first and second stage compressors, including a run at the expected highest operating point for the MOLLER experiment run (2 second stages and 1 first stage) was performed.

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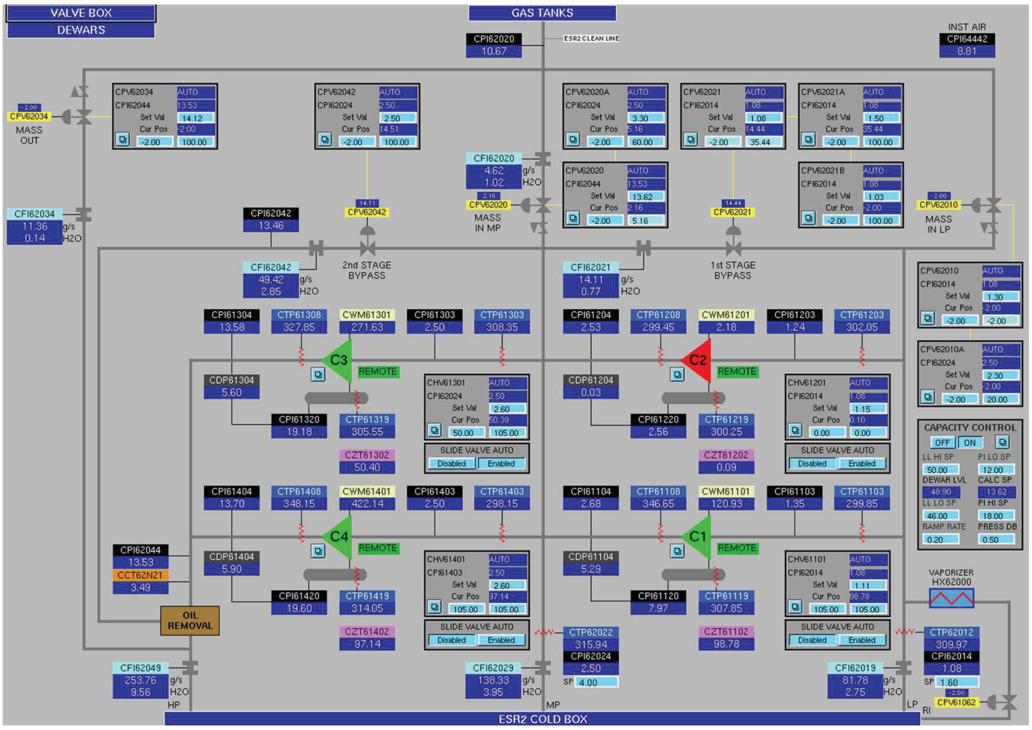

Figure 1. Layout of ESR2 compressor system. Values shown are not representative of commissioning runs.

2. Pre-Commissioning

Throughout the installation process, methodical pre-commissioning checks were performed on all components and subsystems. All piping was checked for potential helium leaks using a combination of pneumatic pressure testing and sensitive leak checking using a helium mass spectrometer leak detector. All electrical cabling for power transmission, control, and data communications was installed, functionality of all instrumentation and controls were verified by checking that all sensors reported sensible measurements and that all actuators responded properly when requested. Prior to initial operation of the system, process piping was cleaned by evacuating and refilling with clean helium gas and then circulating through the CHL helium purifiers while monitoring nitrogen and moisture levels. Testing commenced when contamination levels of nitrogen and moisture were favorable.

3. Results

Volumetric and isothermal efficiencies were compared to previously published data [2]. Volumetric efficiency, η_{Vol}, is a measure of how effectively a compressor draws in and delivers gas. It is the ratio of the actual mass flow rate of helium to the ideal, or theoretical, mass flow rate that

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the compressor should be capable of based on its displacement rate. The theoretical flow is calculated from the compressor's displacement rate and the helium density at the inlet. This relationship is shown in equation (1), where $\dot{m}_{out}$ is the measured helium flow rate at the outlet, $\dot{V}_{disp}$ is the displacement rate of the compressor, and ρ_{in} is the helium density at the inlet of the compressor based on temperature and pressure measurements.

$$\eta_{Vol} = \frac{\dot{m}_{out}}{\dot{V}_{disp}*\rho_{in}} \tag{1}$$

Isothermal efficiency, $\eta_{Isotherma}$, is the ratio of the actual mechanical input power, $\dot{W}_{Mechanical}$, to the theoretical ideal Isothermal compression work, $\dot{W}_{Isothermal}$. Ideal isothermal compression is shown in equation (2), and isothermal efficiency in equation (3), where P_{in} is inlet pressure, and $\rho_{out_{iso}}$ is the isothermal helium density at the compressor outlet pressure.

$$\dot{W}_{Isothermal} = -\left(\frac{P_{in}}{\rho_{in}}\right)\ln\left(\frac{\rho_{in}}{\rho_{out_{iso}}}\right)\dot{m}_{out} \tag{2}$$

$$\eta_{Isotherma} = \frac{\dot{W}_{Mechanica}}{\dot{W}_{Isothermal}} \tag{3}$$

$\dot{W}_{Mechanical}$, shown in equation (4), is determined by measuring the electrical input power, $\dot{W}_{Electrical}$ and applying an assumed motor efficiency, η_{Motor}, which is 95 % for all motors used in the tests.

$$\dot{W}_{Mechanica} = \dot{W}_{Electrical} * \eta_{Motor} \tag{4}$$

Inlet temperatures and pressures were measured at the suction of each compressor downstream of the inlet strainer and isolation valve to account for the pressure drop of those components. The outlet temperatures and pressures were measured at the discharge of the compressor, prior to any local oil separation or gas cooling on the skid. Electrical input power was measured via current meters installed on each motor control center, while power factor corrections were developed using upstream power monitoring equipment. Flow rates were measured using venturi style flow meters on each compressor's respective stage bypass line. During initial calibration runs it was discovered that the 1st and 2nd stage flow meter venturis were swapped during fabrication. The venturis are very similar externally. Each venturi has the same inlet diameter (2.157 in.), but their throat diameters are different. The flow meter calculations were updated with the proper venturi dimensions prior to gathering data. The Beta values for the as installed flow meters are: β= 0.595 for the 2nd stage, and β =0.659 for the 1st stage. The number of flow diameters upstream and downstream of each venturi was still within industry accepted standards. Venturi dimensions used in the flow calculation were checked against the as built dimensions provided by the manufacturer with no major discrepancies found. Differential pressure transmitters were calibrated prior to gathering data. Volumetric and isothermal efficiencies for the 1st and 2nd stage compressors are shown in Figures 2 and 4 respectively. Volumetric efficiency of the 1st stage compressors averaged about 2.6 % lower than previously published. This could easily be accounted for by the different flow measurement devices used, and the different piping systems that can introduce different leak paths from the MP header to the LP header, causing less flow to go through the flow meter on the 1st stage bypass. Compressor C1 and C2's average Isothermal efficiencies are 10.3 % and 15.6 % lower than those reported during the SSCL testing, respectively. The results are shown in Figure 2. As noted previously [1], the oil pumps on all compressors were replaced with newer, larger capacity models, it is possible this had an effect on the oil flow into the compressor thus affecting the isothermal efficiency as this would change the ratio of compressed helium to oil. The mass flow verses slide valve position was also measured and reported in Figures 3 and 5 for 1st and 2nd stage compressors respectively.

Table 1. 1st stage compressor C1 test data.

Suction Pressure (Atm)	Discharge Pressure (Atm)	Inlet Temp (K)	Outlet Temp (K)	Flow Rate (g/s)	Electrical Power (kW)	Pressure Ratio	Isothermal Efficiency	Volumetric Efficiency
1.094	2.686	291.3	344.3	131.6	149.2	2.455	0.504	0.891
1.095	2.625	296.3	343.7	130.4	148.8	2.396	0.496	0.896
1.109	2.817	296.8	344.5	130.3	158.3	2.541	0.498	0.887
1.110	3.106	297.4	346.2	130.0	173.3	2.799	0.502	0.885

Table 2. 1st stage compressor C2 test data.

Suction Pressure (Atm)	Discharge Pressure (Atm)	Inlet Temp (K)	Outlet Temp (K)	Flow Rate (g/s)	Electrical Power (kW)	Pressure Ratio	Isothermal Efficiency	Volumetric Efficiency
1.083	2.616	292.7	340.8	128.5	154.2	2.416	0.470	0.883
1.076	2.625	293.2	330.5	127.9	163.1	2.439	0.448	0.886
1.083	2.678	294.1	342.9	126.5	151.4	2.474	0.510	0.873
1.084	3.075	293.3	345.0	127.7	180.9	2.838	0.472	0.879
1.081	3.075	294.3	351.1	127.7	180.9	2.844	0.475	0.884
1.077	3.112	293.7	344.1	126.8	183.7	2.889	0.470	0.879

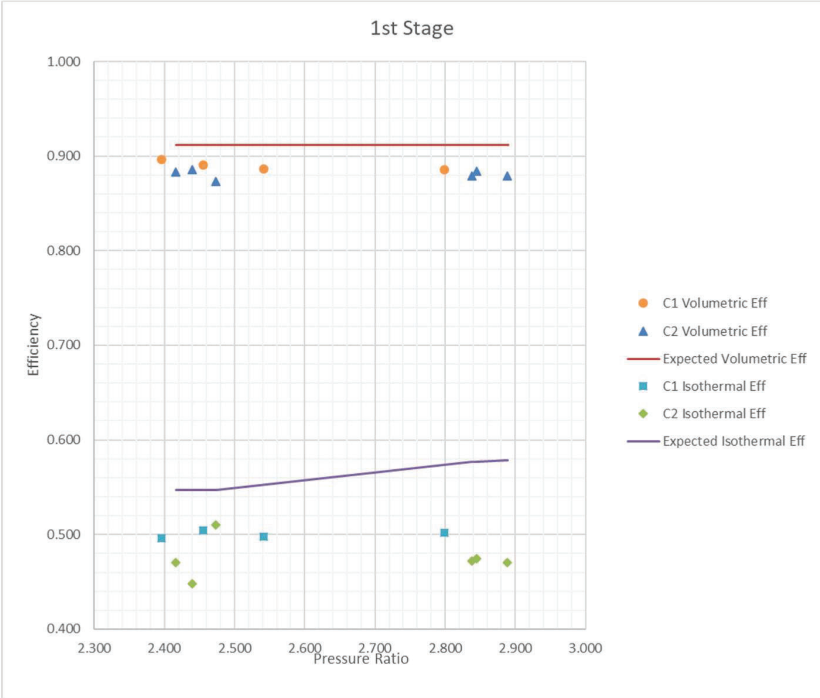

Figure 2. 1st stage compressor Volumetric and Isothermal efficiency. Expected efficiencies were derived from previous commissioning [2].

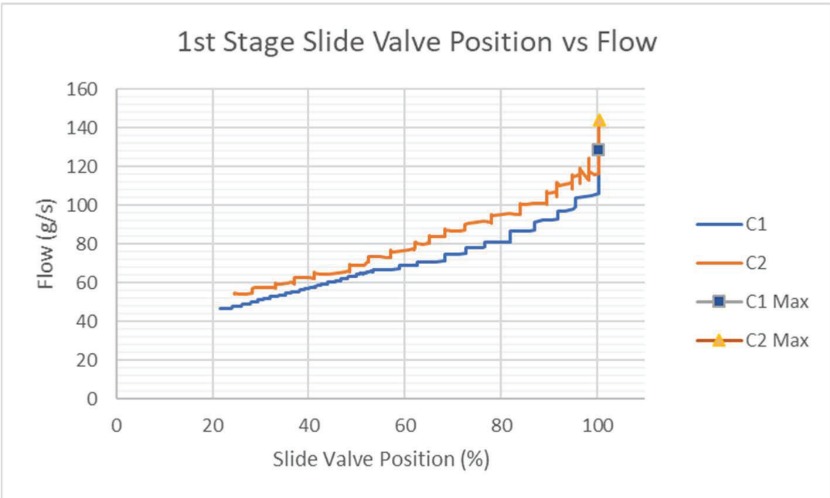

Figure 3. 1st stage compressor Flow vs. Slide Valve Position. Inlet conditions for C1 and C2 were 1.1 Atm, 296 K, and 1.08 Atm, 292 K respectively.

The volumetric and isothermal efficiencies are shown in Figure 4 and the mass flow verses compressor load for the second stage compressors are shown in Figure 5. The second stage compressors outperformed previously reported data [2] for both volumetric and isothermal efficiencies. C3 and C4 averaged a volumetric efficiency of 12.2 % and 9.5 % higher than expected, and the isothermal efficiency was 9 % and 7.7 % higher than expected respectively. The increase in the isothermal efficiency is similar to that of volumetric efficiency, which is most likely due to the increased flow that was measured. There are several possible explanations for this discrepancy. The position of the oil coalescer drain valves have a strong influence on the bypass mass flow. If the SSCL drain metering valves were adjusted further open than ours, it would allow more bypass flow, reducing efficiencies. In addition, the SSCL piping design included an additional piping circuit from the HP header to the LP header with a control valve called the "30 g/s bypass". The JLab design does not include this feature. If this SSCL control valve in this branch had any amount of leak through it would have reduced their efficiency results. The C3 inlet temperature sensor reported a consistently higher value than C4, this is likely an error in the temperature reading, this would reduce the calculated ρ_{in} used in equations (1) and (2) and as such result in fictitiously high volumetric and isothermal efficiencies.

Table 3. 2nd stage compressor C3 test data.

Suction Pressure (Atm)	Discharge Pressure (Atm)	Inlet Temp (K)	Outlet Temp (K)	Flow Rate (g/s)	Electrical Power (kW)	Pressure Ratio	Isothermal Efficiency	Volumetric Efficiency
2.984	12.282	308.6	347.7	264.4	414.2	4.116	0.608	0.943
2.482	12.215	308.3	340.0	218.9	403.9	4.922	0.581	0.938
2.976	16.276	308.9	349.9	261.6	523.4	5.469	0.572	0.937
2.990	18.201	310.6	352.9	259.1	563.2	6.088	0.563	0.928
2.976	18.195	308.7	350.7	260.6	568.8	6.114	0.558	0.932
2.487	16.277	309.4	348.4	217.7	506.0	6.545	0.546	0.934
2.493	18.180	308.4	340.0	216.8	575.2	7.292	0.504	0.925
2.484	18.152	310.8	351.2	215.4	546.0	7.309	0.532	0.930

Table 4. 2nd stage compressor C4 test data.

Suction Pressure (Atm)	Discharge Pressure (Atm)	Inlet Temp (K)	Outlet Temp (K)	Flow Rate (g/s)	Electrical Power (kW)	Pressure Ratio	Isothermal Efficiency	Volumetric Efficiency
2.978	12.258	299.1	347.1	267.2	406.6	4.116	0.607	0.926
2.450	12.197	298.4	345.1	221.9	399.2	4.979	0.581	0.932
2.981	18.190	300.0	352.8	261.4	560.8	6.102	0.552	0.908
2.482	16.173	296.8	348.8	220.0	499.3	6.515	0.535	0.907
2.493	18.159	299.9	351.5	217.7	544.6	7.283	0.519	0.903

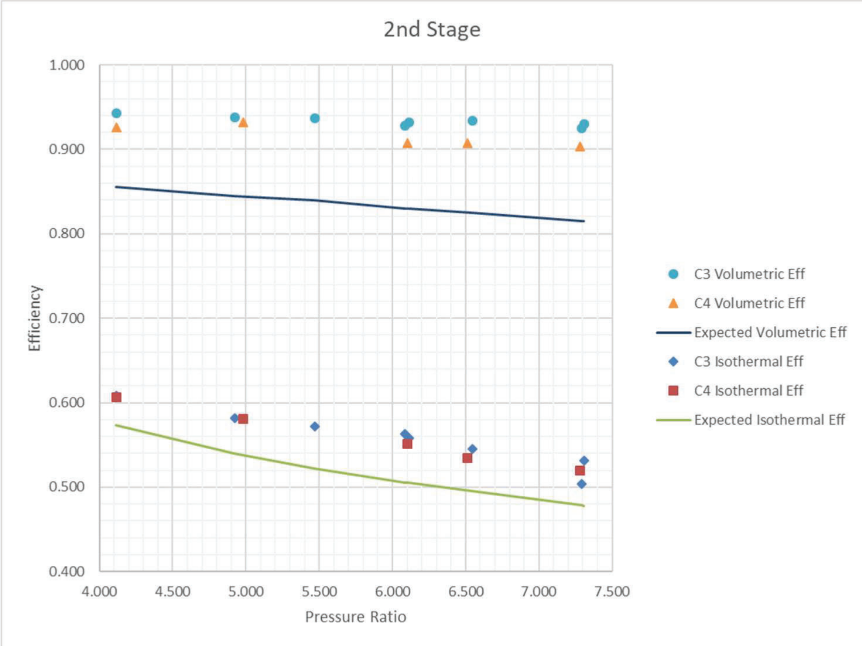

Figure 4. 2nd stage Volumetric and Isothermal efficiency showing better than expected performance.

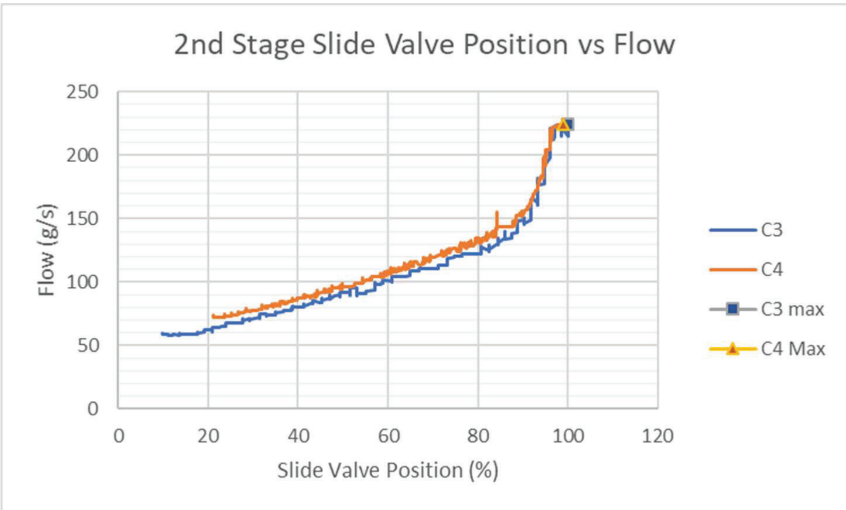

Figure 5. 2nd stage compressor Flow vs Slide Valve Position; exhibiting typical slide valve flow profile with rapid increase in flow starting at 90% slide valve position. Inlet conditions for C3 and C4 with inlet conditions at 2.49 Atm, 308 K and 2.49 Atm, 302 K respectively.

4. Conclusions

During commissioning, the ESR2 compressors proved to be reliable and robust machines, while achieving the necessary performance needed for the upcoming MOLLER experiment. Despite lower than expected 1st stage efficiency, the greater than expected 2nd stage efficiency resulted in an improvement in the total system efficiency, thus lowering power usage. Refurbishment and re-use of the SSCL compressors proved to be a worth-while venture as it reduced project capital cost and still resulted in the required performance despite residing in long term storage for over 30 years.

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References

[1] Bhattacharya R et al. End Station Refrigerator 2 Cryoplant at JLAB *IOP Conf. Series: Materials Science and Engineering* **1301** (2024) 012116

[2] Ganni V and Apparao T 1994 Design verification and acceptance tests of the ASST-A helium refrigeration system *Advances in Cryogenic Engineering* **39** 779–87