

Performance Testing of Jefferson Lab 12 GeV Helium Screw Compressors

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Abstract. Oil injected screw compressors have essentially superseded all other types of compressors in modern helium refrigeration systems due to their large displacement capacity, reliability, minimal vibration, and capability of handling helium's high heat of compression. At the present state of compressor system designs for helium refrigeration systems, typically two-thirds of the lost input power is due to the compression system. It is important to understand the isothermal and volumetric efficiencies of these machines to help properly design the compression system to match the refrigeration process. It is also important to identify those primary compressor skid exergetic loss mechanisms which may be reduced, thereby offering the possibility of significantly reducing the input power to helium refrigeration processes which are extremely energy intensive. This paper summarizes the results collected during the commissioning of the new compressor system for Jefferson Lab's (JLab's) 12 GeV upgrade. The compressor skid packages were designed by JLab and built to print by industry. They incorporate a number of modifications not typical of helium screw compressor packages and most importantly allow a very wide range of operation so that JLab's patented Floating Pressure Process can be fully utilized. This paper also summarizes key features of the skid design that allow this process and facilitate the maintenance and reliability of these helium compressor systems.

1. Introduction

Helium refrigeration systems are extremely energy intensive processes [1], especially 2-K helium refrigeration systems [2]. Even for large systems [1,3], which are typically more efficient [4], it is achievable to obtain 1 W of cooling at 4.5-K for ~250 W of input power; or, 1 W of cooling at 2-K (~30 mbar) for ~750 W of input power. The cold box, which is comprised of heat exchangers, turbines and process control valves, depends solely upon the availability provided by the helium compression system. Often, great effort is expended in the cold box design (i.e., how the supplied availability is utilized), but relatively little effort may be given to the helium compression system design, even though two-thirds of the input power losses are attributable to the compression system [5,6]. The reliability and efficiency of the entire helium system is largely governed by the reliability and efficiency of the helium compression system. Oil flooded screw compressors have improved the former without sacrificing the later [7]. A well matched compression system and cold box system is essential for efficient operation of helium refrigeration systems, especially since there is typically a considerable margin (usually, at least 50%) involved in estimating the loads on helium a refrigeration system. Additionally, these systems are usually commissioned in phases, requiring the refrigeration



system to support partial loads. And, often the load requirements can evolve considerably due to changes in the physics being supported and issues encountered during the project with accelerator and/or major beam end station components (e.g., magnets, targets, etc.). These factors, in addition to decreasing operating budgets, can require a considerable variability in the actual load being supported, even during a fiscal year.

2. Background

The Cryogenics Group at Thomas Jefferson National Accelerator Facility (JLab) has operated screw compressors, as part of helium refrigerators, from many manufacturers for 25 years. These compressors range from 100 to ~2000 kW and are from a wide range of compressor manufacturers and skid packagers. Some staff members have also worked in industry, supplying many of the helium refrigerator systems used at government laboratories and in industry, which includes the compressor and cold box systems. The limitations imposed by these compressor skid designs has become quite apparent to those staff members with a background of having specified their design in industry and then exposed to their operation and maintenance.

As mentioned previously, it is not common for helium refrigerators to run at a single operating condition for the majority of their life and many times these conditions may not often be the maximum capacity. A single operating condition can be more common in industry where the objective is the maximum capacity, except during maintenance shut-down periods. The realization of the now patented Ganni – Floating Pressure Process Cycle [8] during the design of the cryogenic system for the Superconducting Super-Collider Laboratory (SSCL) became the primary motivation to design a compressor skid capable of a wide range of operation of suction and discharge pressures.

It was fortuitous that the request by NASA Johnson Space Center (JSC) to the JLab Cryogenics Group to design and specify a 12 kW 20-K helium refrigerator [9] for the James Webb program provided an opportunity to develop a wide range helium compressor skid for that refrigerator which would use the Floating Pressure Process. Such a deviation from the ‘norm’ would not have been possible without the support of NASA-JSC project team. This development effort for NASA-JSC afforded an acceptable level of risk for the JLab 12 GeV project. Consequently, by the summer of 2013, the 12 GeV upgrade helium compression system was successfully commissioned, composed of compressor skids completely design by JLab and built-to-print by industry, demonstrating a very wide range of operation to allow full implementation of the Floating Pressure Process [3,10].

These compressor skid design packages are being used for the helium refrigerator system supporting the Michigan State University (MSU) – Facility for Rare Isotope Beams (FRIB) [11], having incorporated minor improvements realized during JLab 12 GeV commissioning. Salient 12 GeV compressor skid specifications are shown in table 1 and the high pressure (HP) stage skid is shown in figure 1.

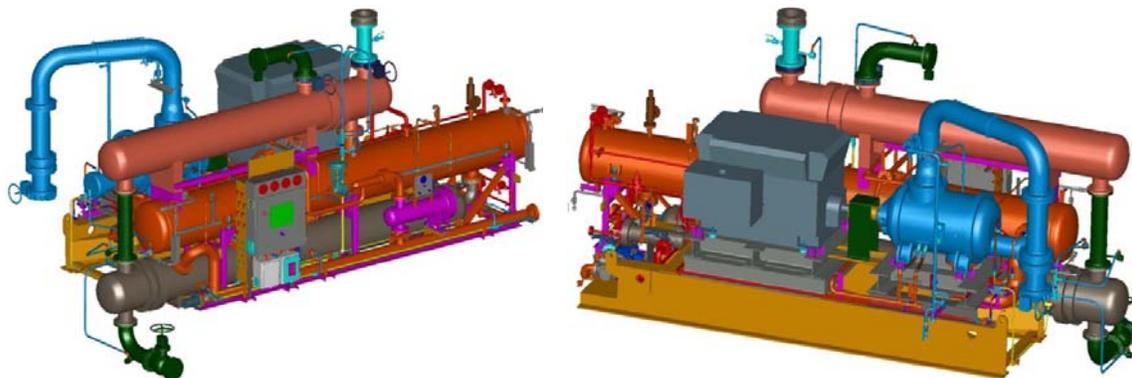


Figure 1. 12 GeV high pressure (HP) compressor skid

Table 1. 12 GeV compressor skid specifications (selected)

Stage	Units	HP	MP	LP	Notes
No. Units		1	1	3	
		Howden	Howden	Howden	
Compressor		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	[mm abs.]	8	8	8	(f)
Bearing Oil Injection Filter	[mm 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

3. Key design features

The JLab 12 GeV (and MSU-FRIB) compressor skids incorporate well proven components and have several key design features; a wide range of operation, good efficiency, considerations for maintenance technician and environmental safety, improved reliability, improved maintainability and lower cost.

3.1. Wide range of operation

Conventional oil-flooded rotary screw helium compressor skid packages can be rather limited in the range of suction and discharge pressure ranges that they can operate; especially in regards to their bulk oil separation (BOS). The helium cold box can function to serve as an ultimate purifier, since there are no known contaminants that will escape being condensed and/or solidified at 4.5-K. So, although dry rotary screw compressors have their appeal in this regard for helium refrigeration systems, they inherently require tighter tolerances and experience higher compression temperatures, especially when helium is the ‘refrigerant’. And, notwithstanding even these considerations, if oil is used to lubricate the bearings and timing gears, a comment attributable to Sam Collins (Professor Emeritus of the Massachusetts Institute of Technology, internationally known as the father of practical helium liquefiers) would seem to seal their fate; ‘It makes little difference whether it is one drop or one gallon of oil in the helium it still needs to be separated for helium liquefiers.’

The BOS’s for the 12 GeV (and MSU-FRIB) compressors are long horizontal vessels without flanges, composed of a short cyclonic separation section followed by an inertial separation section. The cyclonic and inertial sections are separated by a weir and the oil forms the ‘seal’ between the two sections. The entire vessel is sloped ‘downhill’ ~25 mm (end-to-end). The inertial section has two supply headers that direct the helium-oil mixture at a 15 degree angle down from the horizontal plane

into the vessel volume. A return header toward the top center of the vessel collects the helium before going to the after-cooler. There are no glass fiber coalescing elements in this vessel and unlike typical BOS's it has a considerably smaller diameter, but allows a reasonable transit time as the oil flows to the outlet, then onto the oil cooler below. Table 2 summarizes a number of the BOS design parameters (LP – low pressure stage, MP- medium pressure stage, HP – high pressure / swing stage). Figure 2 depicts the BOS cross-sections for the HP stage and a cross-section of the helium gas velocity distribution of the HP stage (calculated; velocity scale is logarithmic and ranges from 1 mm/s to 1 m/s with the mid-section ~20 cm/s). Table 3 shows the range of conditions tested during commissioning for the BOS's and table 4 shows the range of conditions that they have been operated at since being commissioned (the built-in-volume, BVR, setting during operation is: 2.4 for LP stages and 2.2 for MP and HP stages).

Table 2. Bulk oil separator (BOS) design parameters (selected)

		HP	MP	LP	Notes
Outside Diameter	[m]	0.762	0.762	0.610	
Overall Length (Head-to-Head)	[m]	6.78	6.28	6.23	
Minimum Oil Volume	[ℓ]	276	254	225	
Nominal Oil Volume	[ℓ]	407	374	330	(a)
Oil-Helium Surface Area Per Volume	[m ² /m ³]	10.9	10.6	10.5	(b)
Oil Transit Time	[s]	22	81	55	(b)
Max. Carry-Over Droplet Size	[μm]	<100	<100	<100	(c)
Total Oil Carry-Over	[ppmw]	500	500	250	(c), (d)

Notes: (a) 15.2 cm liquid level height; (b) At nominal volume; (c) To helium after-cooler; (d) Parts per million with respect to helium mass flow

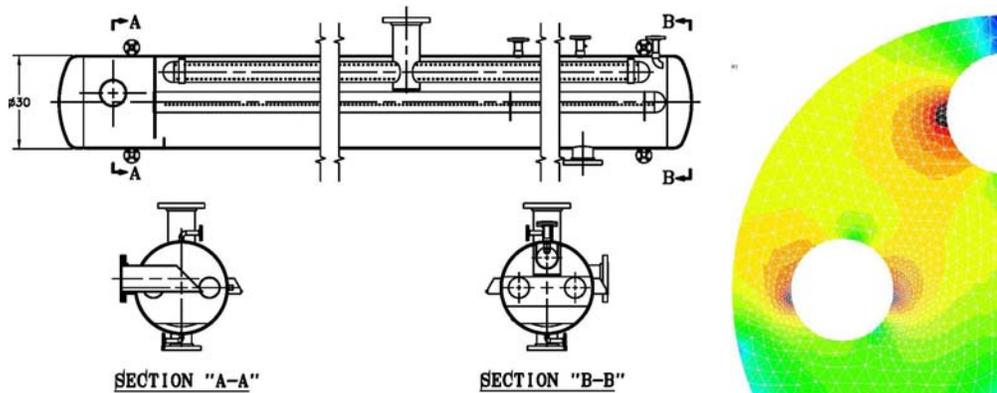


Figure 2. 12 GeV high pressure (HP) stage bulk oil separator (BOS) cross-sections and gas velocity distribution (calculated)

As there will always be oil that the BOS does not separate, the after-cooler heat transfer surface area was specified assuming 0.2% oil (by weight of helium; which marginally increased the surface area by ~10%) and designed so that coalesced oil can drain back to the compressor suction. After the helium leaves the after-cooler and the skid, there is an external coalescer dedicated to that skid. Referring to table 5, these coalescer vessels are substantial smaller in diameter than on other skids and are not subject to the high helium discharge temperatures (or high gas velocities). The compressor oil removal was found to perform adequately for all test and operational conditions listed. Additionally, except for the MP stage at high suction pressures (~3 bar), the external coalescer collected a minimal amount of oil.

Table 3. Range of conditions tested during commissioning of 12 GeV compressors

Compressor Stage	Suction Pressure [bar]	Discharge Pressure [bar]	Pressure Ratio	BVR (†)
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 (§)	1.07 to 5.47	13.37 to 18.75	3.00 to 16.98	2.2 to 5.0 (‡)

(†) BVR increments were 0.2

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

(§) HP stage can be operated as a (swing) LP or MP stage; under these conditions the suction pressure can be as low as ~1.03 atm, the discharge pressure as high as ~18.5 atm, and the pressure ratio ~18

Table 4. 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 (§)	3.19 to 5.78	7.25 to 20.04	2.27 to 3.70

(§) Same note as in table 3

Table 5. Size of selected BOS/coalescer vessels

Plant	Stage	Diameter [m]	Length [m]
CHL-1	LP	1.219	4.48
CHL-1	HP	1.219	5.40
SNS	LP	1.270	3.23
SNS	HP	1.067	3.37
CHL-2	LP	0.610	6.23
CHL-2	MP	0.762	6.28
CHL-2	HP	0.762	6.28

Note: CHL-1 is the old helium refrigerator installed at JLab in the 1990's for a 4 GeV program and used through the 6 GeV era. SNS is the helium refrigerator at Oak Ridge, TN for the Spallation Neutron Source which used compressor bodies essentially identical to CHL-1 except that they have a variable BVR; the skids are not identical to CHL-1. For CHL-1 and SNS, the BOS and coalescers are integral to each other. CHL-2 is the new helium refrigerator for the JLab 12 GeV upgrade. The BOS is part of the compressor skid, but the coalescer is external to the skid and is the same size for all stages; 0.914 m in diameter, 3.86 m long.

3.2. Efficiency

It has been found in practice that there is an optimum helium discharge temperature. Isothermal compression is the ideal compression process for helium refrigeration systems that use a Collins type process [12]. Increasing injected oil flow would tend to make the compression process more isothermal, assuming it mixes well with the helium during compression. However, oil viscosity increases exponentially with decreasing temperature. So, it has been found that allowing the helium discharge temperature to be as high as practically possible results in a minimum input power. To ensure good reliability of seals and bearings, the discharge temperature should not exceed 372 to 378 K, and normally 370 K is used for continuous operation. For the 12 GeV compressors, the oil injected into the rotors for cooling is not pumped, rather the differential pressure between discharge and the

injection point supplies the potential, and is regulated using a control valve that seeks to maintain a set discharge temperature. The oil used to cool the bearings, which is around 20 to 30% of the total oil flow, is supplied to the bearings using a positive displacement pump equipped with a variable frequency drive (VFD) that seeks to maintain a set differential pressure (between pump discharge and compressor discharge). As such, a considerably smaller oil pump is needed than if the full oil flow was being pumped and there is a minimum of excess oil flow handled by the pump that is bypassed back to the BOS, since a VFD is employed. Also, since coalescing elements are not incorporated into the BOS, there is a minimum amount of pressure loss in the bulk oil separation process.

During commissioning, the compressors were operated over the ranges indicated in table 3, and their performance was measured. Input power to the motor, helium mass flow, as well as, helium suction, helium/oil discharge and oil injection process conditions (i.e., pressure and temperature) were measured. Total oil flow was not measured, but can be calculated using an energy balance. The manufacturer's motor efficiency (vs. load fraction) was assumed to be correct, helium and oil were assumed to be at the same temperature upon discharge from the compressor and it was assumed that there was no helium dissolved in the oil injected. The motor speed was measured in some cases (using a stroboscope) and was found to be less than 1% greater than the motor nameplate data. The calculated volumetric and isothermal efficiencies for the LP, MP and HP stages obtained from the 12 GeV compressor commissioning data are shown in figures 3, 4 and 5, respectively. The actual (measured) mass flow is used for the isothermal efficiency. These efficiencies are applicable to the compressor proper (not the overall skid). Volumetric efficiency for the MP stage is ~5% too high due to calibration issues with the suction transducer and the HP stage isothermal efficiency is ~5 to 8% too high due to the calibration of the motor power monitoring or the turn-down assumed for the current or voltage transformers. This data shows similarities to previous work at the SSCL [13].

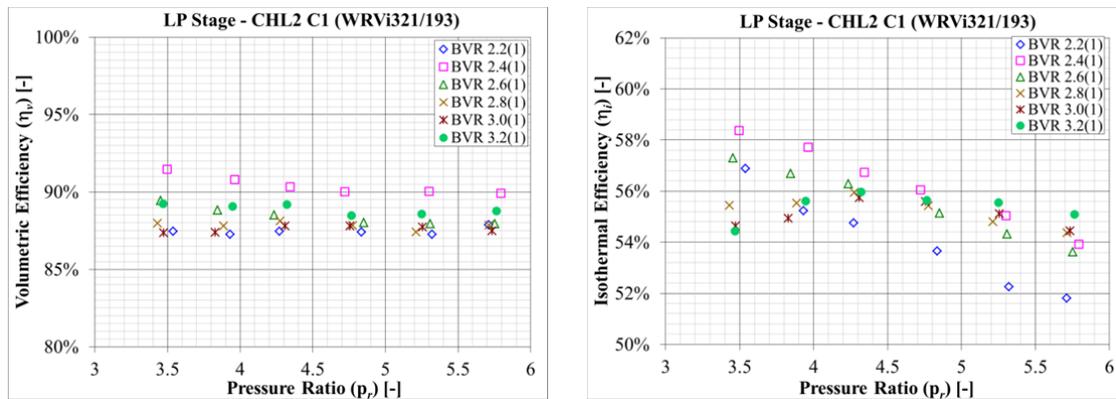


Figure 3. LP stage volumetric and isothermal efficiency for a range of BVR's

3.3. Safety, reliability, maintainability and cost

There are a number of previously mentioned factors that contribute to improved safety, reliability, maintainability and lower cost. The design of the skid is sized and arranged to allow the oil to drain first to the oil cooler and then up to the BOS. Except for a short "P" trap section of pipe between the compressor discharge flange and the BOS inlet, there are no components or sections of pipe that will hold the oil upon shut-down. This allows for ease in start-up and shut-down and the ability to verify the oil inventory, which is not the case in a number of typical designs.

Also, the skids designs allow a minimal oil charge (see table 1), as compared to typical helium compressor skid packages. This factor can be appreciated for those who have operated these machines and had to deal with the oil sprayed externally due to unexpected conditions that caused a relief valve to discharge or a line that developed a crack (such as a flex-hose or tube). The oil spray is not just messy, it can ruin the electric motors if ingested and can cause personnel health and environmental concerns.

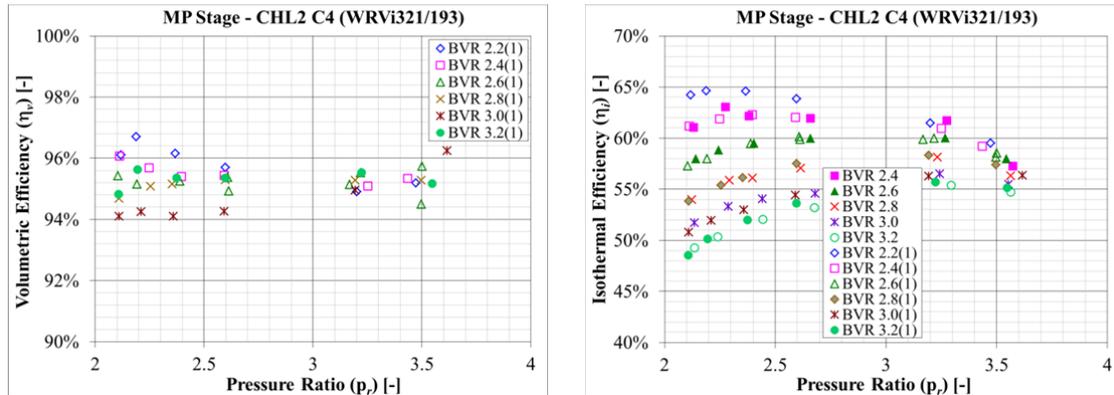


Figure 4. MP stage volumetric and isothermal efficiency for a range of BVR's

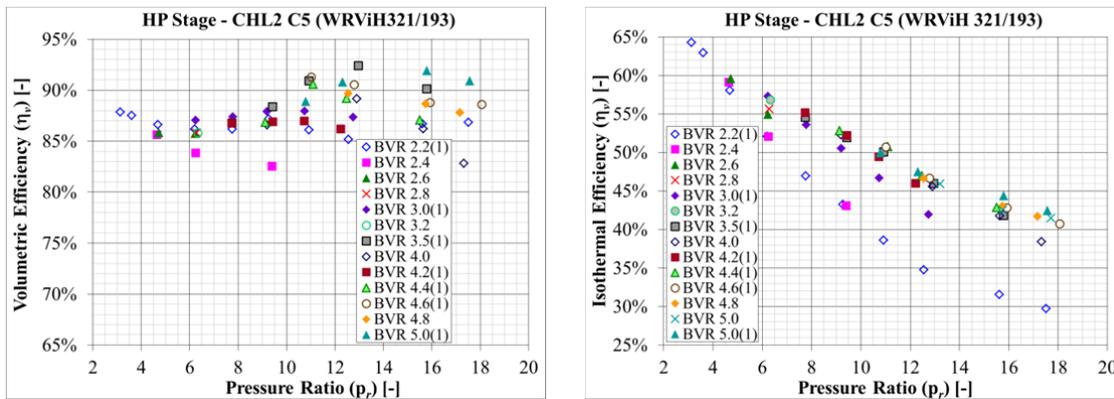


Figure 5. HP stage volumetric and isothermal efficiency for a range of BVR's

As mentioned, there are no coalescers in the BOS, and the external coalescer is not subject to high temperatures or high velocities. In typical designs, if operated outside their limited design range, not only is excessive oil carry-over an issue, these elements have been known to tear.

The BOS design in typical packages is quite large and usually flanged (see table 5). However, the BOS for the 12 GeV compressors is much smaller and has no flanges. This can tend to lower the cost of this component.

The oil pump for the 12 GeV compressor skids is downstream of the oil-cooler. In several typical designs the oil pumps are upstream of the oil-cooler (and are handling the total oil flow), which would seem better for minimizing specific pump input power (handling less viscous oil). However, if they are handling the total oil flow at high temperature, they need to be much larger (than if handling only the bearing injection flow) and they can tend to develop shaft seal leaks quite easily. The 12 GeV compressor pumps run cooler and have not developed shaft seal leaks. And, the fact that only 20 to 30% of the total oil flow is pumped allows for a much smaller pump and motor.

The entire 12 GeV compressor skid is design with minimal elevation for ease of access during maintenance and as mentioned the separate sub-skids isolating the rotating and non-rotating components minimizes the effect of vibration on a majority of the components and controls.

For those who have operated these machines, alignment of compressor and motor after maintenance can be an interesting challenge, which can be complicated by difficult access and thermal expansion effects. These can also affect their alignment over time while operating. For the 12 GeV compressors, both compressor and motor are mounted on frames with thick-plates (see figure 1). These, in turn, are mounted to the main structure below which is anchored to the floor. Additionally,

universal adjustable steel chocks and substantial tabs with adjustment bolts are incorporated to ease the alignment and help prevent a ‘soft foot’.

4. Conclusion

The development of the 12 GeV compressor skids has been part of an ongoing research and development effort at JLab. The present development has allowed a very wide range of operation and a full realization of the Floating Pressure Process [8]; and, after ~11,000 hours of operation (by the end of 2014 for each skid) they have proven to be efficient, reliable and easily maintained. However, there is still considerable research to be done to develop alternate BOS designs and investigate methods and processes to improve efficiency; especially 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.

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