Fermi National Accelerator Laboratory
Technical Division
PO Box 500 MS 316
Batavia, IL 60510

Proposed Upgrade of the MTF Helium Compressor System

R. Rabehl

Abstract:

The impact of one screw compressor characteristic, the internal volume ratio, on MTF helium compressor operations has been studied using manufacturer's data and archived MTF operations data. Based on these results, an MTF helium compressor system upgrade consisting of changing the internal volume ratio is proposed. The expected operational impact of this upgrade is presented.

1. Introduction

Commissioned in the late 1970's, the compressor system and cold box form the heart of the MTF cryogenics system. The system has provided reliable operation from the Tevatron superconducting magnet production testing up to and including today's LHC quadrupole production testing and high-field magnet R&D testing.

Today's high demand on the MTF cryogenics system has led to the observation of limits of the current system. The compressor skid was chosen as a starting point for investigating these limits and understanding whether they can be overcome.

This note describes screw compressor operations and compares MTF operating data with performance data gathered from the manufacturer. The internal volume ratio is identified as a characteristic of the MTF helium compressors that can be changed, resulting in a significant improvement in cryogenic operations.

2. Screw Compressor Operations

Rotary screw compressors operate by compressing gas between two rotors. The Sullair C25L compressors used at MTF use a four-lobe, male driven rotor and a six-lobe female rotor. At the low-pressure end of the compressor, gas fills the interlobe spaces (called flutes) of the rotors. As the rotors spin, the gas is trapped between the rotors and the compressor housing. The spinning rotors move the trapped gas toward the discharge end of the compressor. The gas is then compressed as the rotors mesh, and it is finally discharged at the high-pressure end of the compressor.

The Sullair C25L compressors have a slide valve for capacity control. When the compressor is fully unloaded, the gas trapped between the rotors is isolated from the suction volume after the rotors begin meshing. A significant fraction of the gas is then squeezed out from between the rotors and back into the suction volume. The mass flow rate reaching the discharge end of the compressor is reduced. As the compressor is loaded, the gas trapped between the lobes becomes isolated from the suction volume sooner and the mass flow rate is increased. The longitudinal position of the slide valve relative to the rotors therefore determines the mass flow through the compressor.

A characteristic of screw compressors called the internal volume ratio (VI) is also determined by the slide valve. The VI is the nominal ratio of the initial gas volume trapped between the rotors and the housing at suction pressure to the compressed volume of gas trapped between the rotors. The VI value of a slide valve is determined by the size of a notch machined into its end. A slide valve with a larger VI has a smaller notch because the rotors must rotate further and hence compress the gas more before the notch is seen and the compressed gas can escape from between the rotors.

Fig. 1 [1] shows a typical cross-section of a rotary screw compressor and a comparison of slide valves with various VI values.

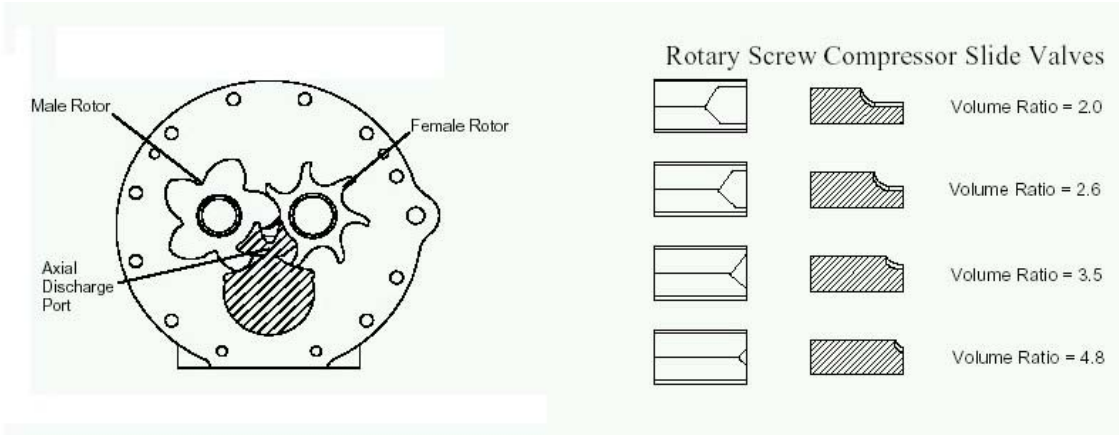

Fig. 1 Rotary screw compressor cross-section and slide valves.

How does one decide which VI should be used? A first approximation can be calculated by assuming either an isentropic compression process for oil-free screw compressors or an isothermal compression process for oil-flooded screw compressors like those used at MTF. Equation 1 can derived from basic thermodynamics, where n = k (ratio of specific heats, k = 1.667 for helium) for an isentropic process and n = 1 for an isothermal process.

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n = (VI)^n \tag{1}$$

Plotting the isothermally and isentropically ideal VI's as a function of pressure ratio results in the plot of Fig. 2.

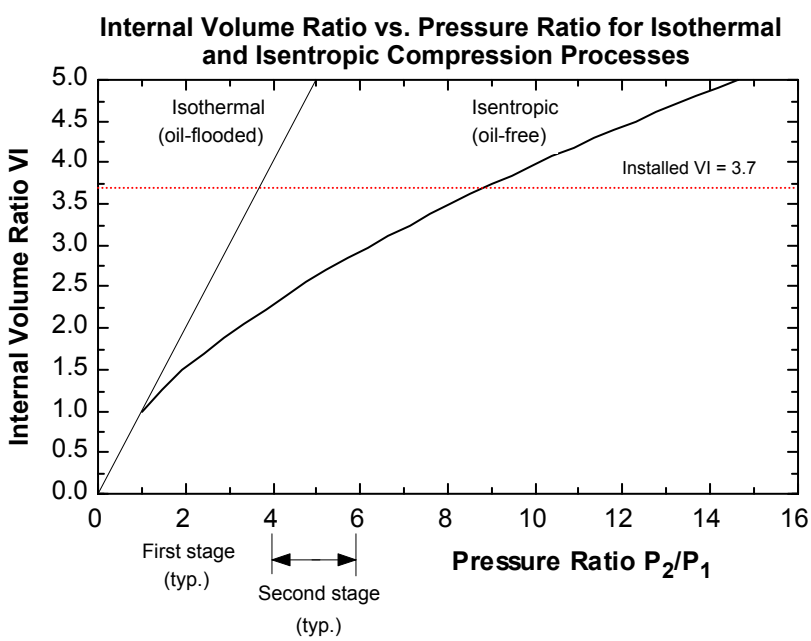

Fig. 2 Internal volume ratio vs. pressure ratio for ideal isothermal and isentropic compression processes.

On Fig. 2, typical pressure ratios of 2 for the first stage and 4-6 for the second stage are indicated. For an isothermal compression, an installed pressure ratio of VI = 3.7 is a good match for the second stage compressor. It is too high for the first stage compressor however, and the result is overcompession. Overcompression increases the input power and is inefficient because power is wasted in compressing the gas only to have it re-expand to the lower pressure of the discharge piping. Undercompression is not as inefficient as overcompression, although some power is wasted in moving gas that has backflowed from the discharge piping into the compressor.

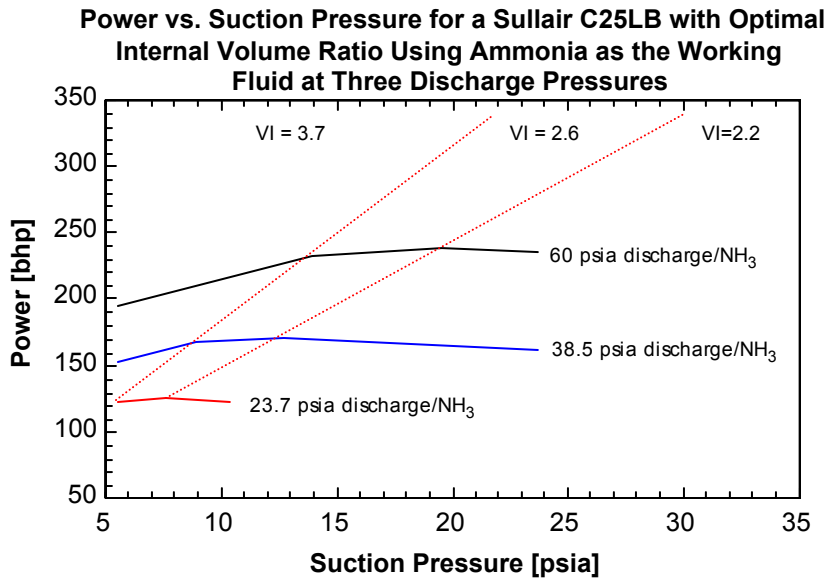

Fig. 3 Power vs. suction pressure at three discharge pressures for a VI-optimized Sullair C25L screw compressor with NH_3 as the working fluid [2].

Fig. 3 illustrates that for each of the discrete VI values offered by a manufacturer, there is an optimal range of pressure ratios. At higher pressure ratios, a higher value of VI is optimal in that input power is minimized. At lower pressure ratios, a lower value of VI is optimal.

3. MTF Screw Compressor Operations

The VI of a Sullair C25L compressor can be set to one of three values by changing the slide valve. These values are 2.6, 3.7, and 4.8. The VI = 2.2 shown in Fig. 3 is no longer available for the C25L [3]. The three compressors currently at MTF (in-service first stage, in-service second stage, and spare) have installed VI's of 3.7. This has been confirmed with the manufacturer [4], and VI values have not been changed at MTF.

Knowing that both MTF compressor stages use VI = 3.7, manufacturer's operating data (volumetric efficiency, isentropic (adiabatic) efficiency, and isothermal efficiency) were

gathered. Equations 2-5 were then used to calculate the motor current for a fully-loaded compressor. All manufacturer's data can be found at the end of this note [5].

$$\dot{m} = \eta_v \, \rho_{in} \, \dot{V}_{disp} \tag{2}$$

$$\dot{w} = c_p \, T \left[\left(\frac{P_{out}}{P_{in}} \right)^{\frac{k-1}{k}} - 1 \right] / \eta_s \tag{3}$$

$$\dot{w} = R \, T \ln\left(\frac{P_{out}}{P_{in}} \right) / \eta_t \tag{4}$$

$$\dot{W} = \dot{m} \, \dot{w} = \sqrt{3} \, V \, I \, (pf) \, \eta_{mech} \tag{5}$$

Equation 2 calculates the mass flow rate $\dot{m}$ in terms of the volumetric efficiency η_v, inlet density ρ_{in}, and volumetric displacement $\dot{V}_{disp}$ of the screw compressor. Equation 3 calculates the power per unit mass flow $\dot{w}$ in terms of the specific heat c_p (5192 J/kg-K), inlet temperature T (311 K, assumed by the consultant), discharge pressure P_{out}, suction pressure P_{in}, ratio of specific heats k (1.667), and isentropic (adiabatic) efficiency η_s. As a check, Equation 4 also calculates the power per unit mass flow but in terms of the ideal gas constant R (2077 N-m/kg-K), inlet temperature, discharge pressure, suction pressure, and isothermal efficiency η_t. Finally, Equation 5 calculates the full load current draw I in terms of the total power $\dot{W}$, voltage V (480 Vac), power factor pf of the motor (0.908 for the first stage motor), and the mechanical efficiency η_{mech} of the motor (0.936 for the first stage motor).

The expected motor current vs. suction pressure for a fully-loaded C25L with 35 psia discharge pressure is plotted in Fig. 4. The 219 amperes full load motor current for the in-service 200 hp/480 3-phase Vac motor is also plotted. It is interesting to note that with the installed internal volume ratio VI = 3.7, the full load current would be expected to be exceeded at all times under MTF operating conditions of 17.5 psia minimum suction pressure.

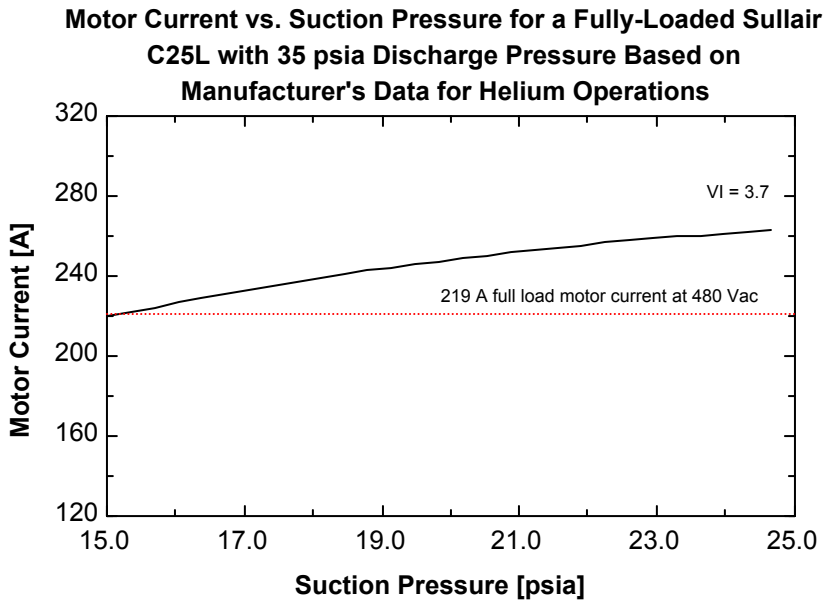

Fig. 4 Motor current vs. suction pressure for a fully-loaded Sullair C25L operating with helium, a 35 psia discharge pressure, and an internal volume ratio of VI = 3.7.

Fig. 5 overlays archived MTF operating data on Fig 4. This data was collected between June 4, 2004 and July 4, 2004, and the loading positioner feedback indicated nearly full (96-97%) loading for most of the data points. The operating data has the approximate shape of the VI = 3.7 curve as expected. However, the operating motor currents are 40-50 A less than the expected full load motor current. This indicates that the fully loaded position of the actuator is not the fully-loaded position of the compressor itself. It appears that the fully-loaded position of the actuator has been purposely set short of the compressor fully-loaded position in order to keep the motor current below the full load value. Scaling the mass flow directly with current, the partially-loaded machine in operation appears to be moving about 80% of its fully-loaded capacity. Operations data suggests the first stage compressor shortfall is even greater.

A better estimate of the first stage compressor throughput will be obtained at the start of the next maintenance period when the compressors can be isolated from the cold box, and only mass flows within the compressor skid need to be considered. The results of this and other compressor system tests will be documented in another technical note tentatively titled, “Studies of the MTF Helium Compressor System.”

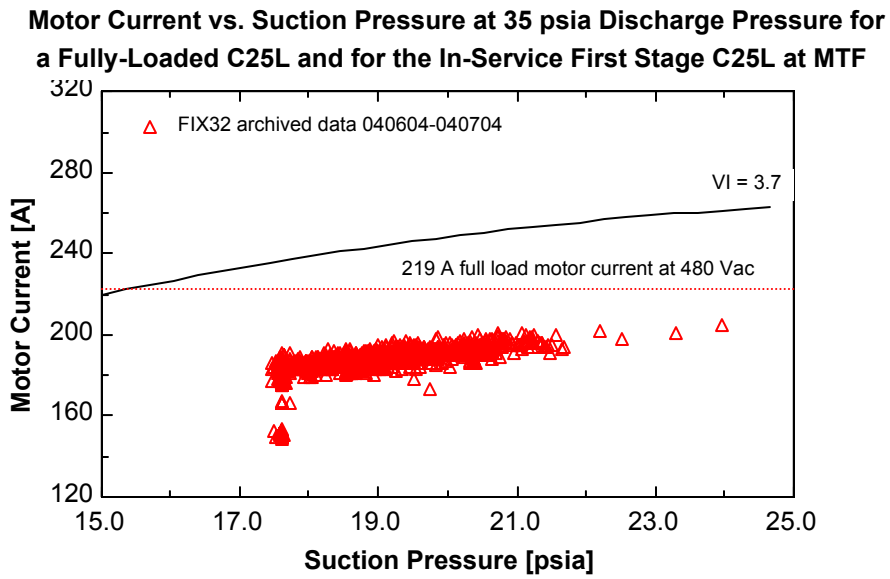

Fig. 5 MTF operating data and manufacturer's data for a fully-loaded Sullair C25L operating with helium.

A similar analysis can be done for a fully-loaded second stage compressor using Equations 2-5 to calculate the 1000 hp second stage motor current. The nameplate information for this motor indicates that at 460 Vac, the full load current is 1135 A. This means the product of the power factor and mechanical efficiency are 0.82. Using this information with the compressor manufacturer's data results in Fig. 6. Fig. 6 plots the motor current vs. discharge pressure for a fully-loaded C25L with a 35 psia interstage pressure and an internal volume ratio of VI = 3.7. The full load motor current is also shown but is well above any expected operating current.

Fig. 7 overlays second stage compressor operating data on Fig. 6. The plotted data are from February 25, 2004 through April 21, 2004 and June 4, 2004 through July 4, 2004. The scatter in the data is due to operating the compressor over a range of loadings. The operating data have the expected slope but show motor currents above the expected current for a fully-loaded machine. Operations data also indicate that like the first stage compressor, the second stage compressor throughput is less than would be expected based on manufacturer's data.

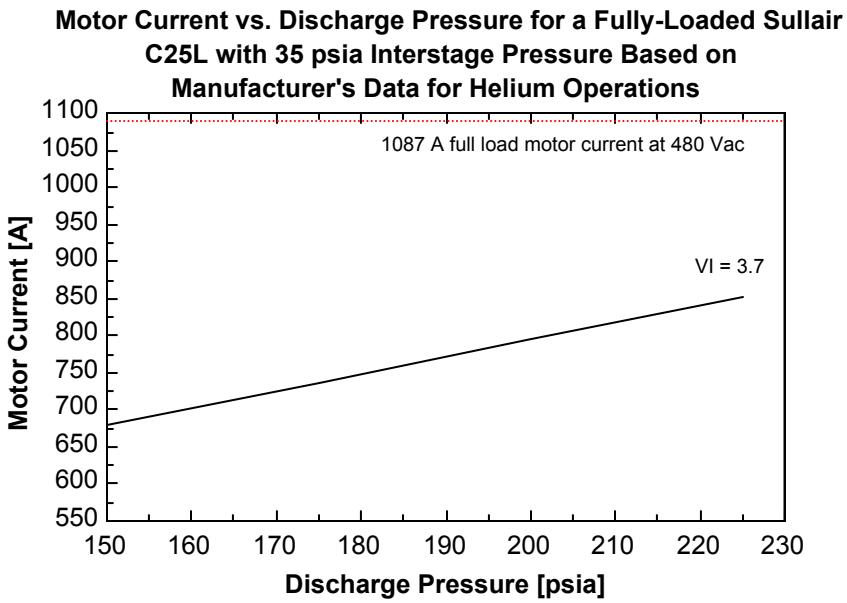

Fig. 6 Motor current vs. suction pressure for a fully-loaded Sullair C25L operating with helium, a 35 psia interstage pressure, and an internal volume ratio of VI = 3.7.

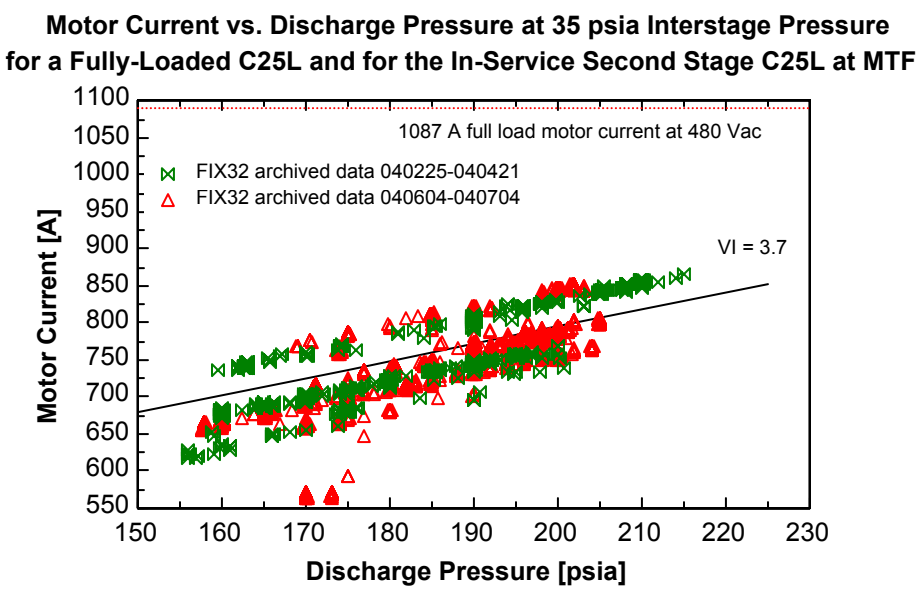

Fig. 7 MTF operating data and manufacturer's data for a fully-loaded Sullair C25L operating with helium.

4. Suggested Compressor Upgrade: Change in Internal Volume Ratio

The proposed upgrade will change the internal volume ratio of both in-service compressors and the one spare compressor from VI = 3.7 to VI = 2.6.

In addition to MTF operating data and manufacturer's data for VI = 3.7, Fig. 8 shows the expected operating current for an upgraded MTF first stage compressor with VI = 2.6. Increasing loading to increase the compressor throughput by an estimated 25% would bring the operating data up to the fully-loaded VI = 3.7 line, whereas increasing the compressor throughput by 25% and changing the slide valve would bring it down to the VI = 2.6 line. The compressor throughput will have increased, but the operating current will have decreased by reducing the overcompression of the gas.

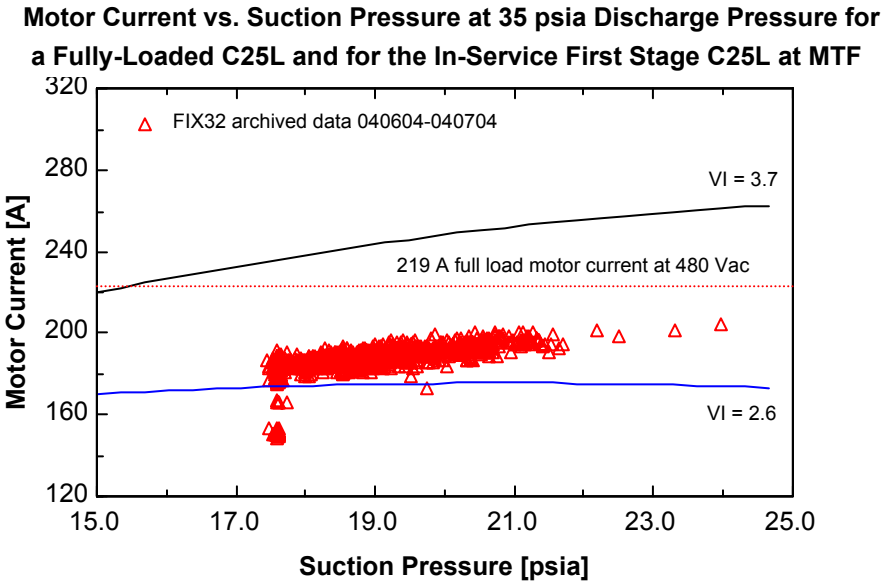

Fig. 8 Manufacturer's data for a fully-loaded Sullair C25L operating with helium and internal volume ratios of VI = 3.7 and VI = 2.6 and MTF operating data.

Fig. 9 shows similar data for the MTF second stage compressor. Up to a discharge pressure of 225 psia, a VI = 2.6 is preferred over VI = 3.7 because of reduced motor current. Above 225 psia, VI = 3.7 is more efficient but significantly so only when the operating discharge pressure is at least 250 psia.

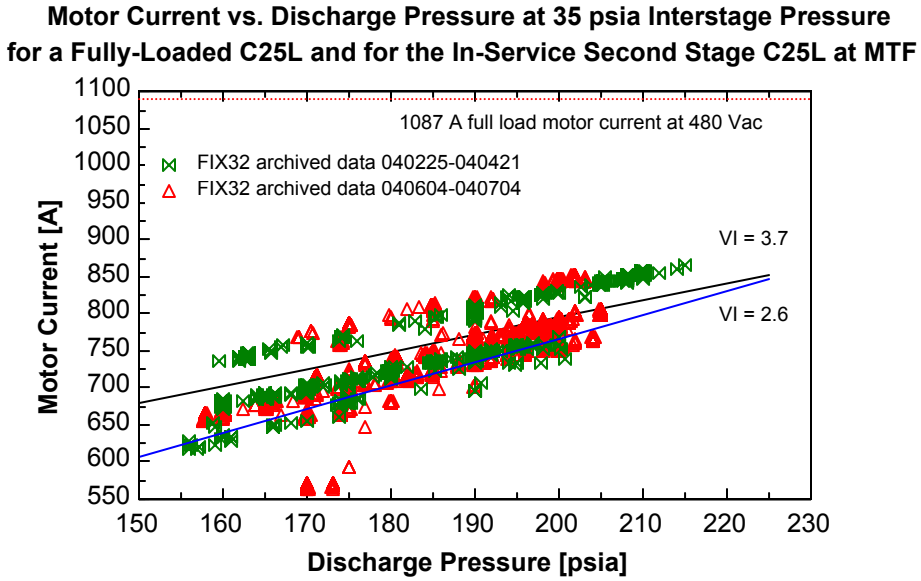

Fig. 9 Manufacturer's data for a fully-loaded Sullair C25L operating with helium and internal volume ratios of VI = 3.7 and VI = 2.6 and MTF operating data.

Changing the VI of the MTF compressors is clearly advantageous for the first stage compressor, but less so for the second stage compressor. However, keeping the same VI in all MTF compressors (in-service and spare) is operationally advantageous. The spare compressor can then be used in either the low pressure or high pressure position without requiring an invasive modification to change the slide valve. Old purchase requisitions indicate that until 1993, the first stage compressor did use a VI = 2.6 slide valve while the second stage used a VI = 3.7 slide valve. The first stage compressor was then switched to a VI = 3.7 slide valve so that only one spare compressor would be needed.

To upgrade the slide valves, the compressors do not need to be removed from the skid; the motors do not need to be moved, either. Each of the three compressors would require about $7,300 worth of parts: VI = 2.6 slide valve (P/N 022909, $4,416.50), shaft seal (P/N 001809A, $1,652.50), capacity control valve repair kit (P/N 001840A, $1,175.00), and outlet valve cover (P/N 001832A, $67.50). On-site installation of the components will be done by Mid-States Refrigeration Supply of LaPorte, IN. Mid-States is the local servicer and distributor of Sullair refrigeration compressors. Eight hours per compressor is required to install a new slide valve. The total cost of the upgrade is estimated to be $25,000, including parts, contractor labor, and travel costs. This upgrade can be accomplished during a multi-week refrigerator maintenance shutdown with a short commissioning period following startup.

5. Operational Impact of Internal Volume Ratio Change

Changing the internal volume ratio of the MTF helium compressors from VI = 3.7 to VI = 2.6 will have a significant impact on cryogenic operations. The increased throughput of the first stage compressor will allow suction pressure to remain much more stable.

Table 1 illustrates this. The first column gives the range of typical suction pressures observed during current MTF operations. The typical suction pressure during liquefaction is 17.6 psia, but this can rise up to 23 psia as test systems are brought online and discharge pressure is changed. The second column shows the corresponding suction pressure due to the upgraded first stage throughput. It is important to note that suction pressure will be maintained at 17.6 psia by the gas management system throughout most of the typical operating range.

Table 1 Comparison of suction pressures for the existing and upgraded first stage compressor.

Suction pressure – existing (psia)	Suction pressure – upgraded (psia)
17.6	17.6
18.0	17.6
19.0	17.6
20.0	17.6
21.0	17.6
22.0	17.6
23.0	18.4

The increased first stage throughput and better-controlled suction pressure have several implications to MTF cryogenic operations:

- As suction pressure rises, the storage dewar pressure must rise as well to maintain a differential pressure for liquid helium transfer to the test facilities. The ability to maintain lower storage dewar pressure will maximize the liquefaction rate.
- The compressor system will be capable of running at higher discharge pressure, assuming there are no limitations of the cooling system or the power system. Increasing compressor discharge is one of the most important factors in increasing the liquefaction rate of the plant and providing greater flexibility in cryogenic operations. There are two aspects that will allow operation at higher discharge pressure. The first aspect is that the increased first stage throughput will reduce the amount of helium that must be recirculated from discharge to the interstage by the gas management system. The second aspect is that increasing discharge pressure will push more helium to the cold box, and therefore more helium will return to compressor suction. This additional flow will now be able to be handled by the first stage compressor.
- Low suction pressure maximizes the volumetric efficiencies of the cold pumps used on the Tevatron magnet test stands and the Kinney pump skids used for subcooled operation at the Vertical Magnet Test

Facility and the LHC quadrupole test stand, reducing pumpdown times and quench recovery times.

Details of plant operations will be addressed in another technical note tentatively titled, "Modeling of MTF Cryogenic Plant Operations."

6. Conclusion

This note has demonstrated that changing the internal volume ratio of the MTF helium compressors from VI = 3.7 to VI = 2.6 offers several advantages. Liquefaction rates can be maximized due to better control of the storage dewar pressure as well as allowing increased discharge pressure. Volumetric efficiencies of the MTF subatmospheric systems will also be improved. This upgrade will be an important first step in improving cryogenic operations at MTF.

References

1. "Rotary Screw Compression Process," prepared for the Gas Machinery Research Council Conference, October 7, 2003, Salt Lake City, Utah, Jeffrey R. LaPlante, Ariel Corporation.

2. Sullair Refrigeration Compressors C Series Technical Manual, Form No. P00286, 1983.

3. Dan Mroz, Mid-States Refrigeration Supply, LaPorte, IN, private communication.

4. John Shoop, Sullair Corporation, Michigan City, IN, private communication.

5. Ken Puetzer, Cool Consulting Company, Inc., LaPorte, IN and former chief refrigeration engineer, Sullair Corporation, Michigan City, IN, private communication.

Appendix - Manufacturer's Data

COMPRESSOR: C25LB (FIRST STAGE)
VI = 3.7
INLET TEMPERATURE = 100 F
DISPLACEMENT = 1710 cfm @ 3600 RPM

DISCHARGE PRESSURE (PSIA)	EFFICIENCY	INLET PRESSURE (PSIA)		
		15	20	25
35	VOLUMETRIC	89.80	90.01	90.13
	ADIABATIC	48.69	35.87	24.29
	ISOTHERMAL	39.89	31.33	22.26

COMPRESSOR: C25LA (SECOND STAGE)
VI = 3.7
INLET TEMPERATURE = 100 F
DISPLACEMENT = 1710 cfm @ 3600 RPM

DISCHARGE PRESSURE (PSIA)	EFFICIENCY	INLET PRESSURE (PSIA)
		35
150	VOLUMETRIC	88.67
	ADIABATIC	73.76
	ISOTHERMAL	53.97
175	VOLUMETRIC	87.92
	ADIABATIC	77.03
	ISOTHERMAL	54.51
200	VOLUMETRIC	87.17
	ADIABATIC	78.99
	ISOTHERMAL	54.29
225	VOLUMETRIC	86.43
	ADIABATIC	80.03
	ISOTHERMAL	53.59

COMPRESSOR: C25LB (FIRST STAGE)
VI = 2.6
INLET TEMPERATURE = 100 F
DISPLACEMENT = 1710 cfm @ 3600 RPM

DISCHARGE PRESSURE (PSIA)	EFFICIENCY	INLET PRESSURE (PSIA)		
		15	20	25
35	VOLUMETRIC	89.80	90.01	90.13
	ADIABATIC	62.91	50.71	36.98
	ISOTHERMAL	51.54	44.29	--

COMPRESSOR: C25LA (SECOND STAGE)
VI = 2.6
INLET TEMPERATURE = 100 F
DISPLACEMENT = 1710 @ 3600 RPM

DISCHARGE PRESSURE (PSIA)	EFFICIENCY	INLET PRESSURE (PSIA)
		35
150	VOLUMETRIC	88.67
	ADIABATIC	82.61
	ISOTHERMAL	60.44
175	VOLUMETRIC	87.92
	ADIABATIC	82.70
	ISOTHERMAL	58.52
200	VOLUMETRIC	87.17
	ADIABATIC	81.89
	ISOTHERMAL	56.28
225	VOLUMETRIC	86.43
	ADIABATIC	80.56
	ISOTHERMAL	53.95