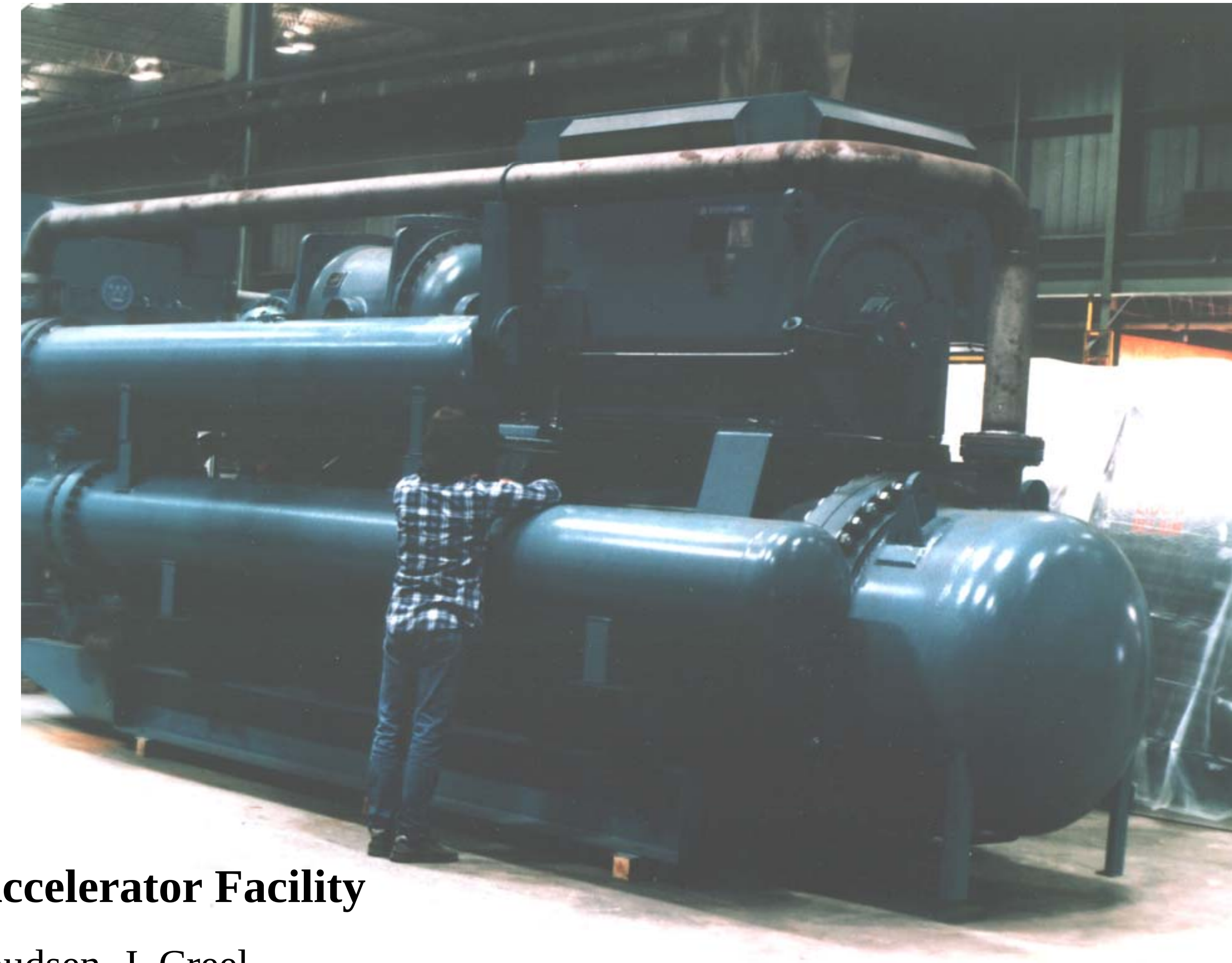
Cryogenics



Screw Compressor Characteristics for Helium Refrigeration Systems

Thomas Jefferson National Accelerator Facility

D. Arenius, V. Ganni, P. Knudsen, J. Creel

Spallation Neutron Source Project

F. Casagrande, D. Hatfield, M. Howell

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Present Compressor Features

Favorable:

The oil injected screw compressors have practically replaced all other types of compressors in modern helium refrigeration systems due to their large displacement capacity, minimal vibration, reliability and capability of handling helium's high heat of compression.

Unfavorable:

At the present state of compressor system designs, half the input energy is lost in the compression system. Therefore it is important to understand the isothermal and volumetric efficiencies of these machines as a guide for the full system design.

Presentation Focus

This presentation summarizes three separate tests that have been conducted on Sullair compressors at the Superconducting Super-Collider Laboratory (SSCL) in 1993, Howden compressors at Jefferson Lab (JLab) in 2006 and Howden compressors at the Spallation Neutron Source (SNS) in 2006. These introduce the effects of built-in volume ratio variation and pressure ratio to map out maximum compressor isothermal efficiency operating domains. These high efficiency domains then can be compared to efficiency domains of the refrigerator (turbines) for insight in the design approach of the entire compressor/refrigerator system.

The compressor pressure ratio and built-in volume ratio are the primary investigation parameters affecting the efficiencies of the screw compressor proper although it is evident from these tests that the compressor skid design can also strongly influences the overall efficiencies and performance of the compression system. This work is part of an ongoing task at JLab to understand the theoretical basis for these loss mechanisms and implement practical solutions for increased efficiency.

Compressor Variable Characterization

- **Stage Location: First, Second, etc. which determines the amount of mechanical loading**

- **Pressure Ratio for the stage location**

- **Built-In Volume Ratio of the Compressor (BVR)**
  - ❖ **Should Ideally be matched to prevent internal over and under compression compared to imposed external compressor pressure ratio (prevent irreversible inefficiencies)**

Q: But what BVR values are best? Suspect that these may be different based on operating pressure ratio and stage location. We'll address this by looking at the results of three tests.

Test Performance Parameters Definitions

- **Volumetric Efficiency**

  Ratio of the actual (measured) mass flow rate to the theoretical mass flow rate, as calculated using the swept volume (displacement) at the measured inlet conditions

- **Isothermal Efficiency**

  Ratio of the theoretical input power to isothermally compress the actual mass flow rate to the 'measured' shaft power being provided to the compressor (as given by the measured input power to the compressor motor and corrected by the motor efficiency)

Test #1: SSCL Compressor Fixed BVR

For the testing, each oil drain was carefully set (i.e., they were not shut-off or reduced so that there was no oil accumulation) to minimize the helium by pass. The calculated volumetric and isothermal efficiencies are based on pressures into and out of compressor skid (i.e., they include the bulk oil separator and helium gas cooler pressure drop which was roughly 0.2 bar for both stages. Built-in volume ratios (BVR) were 2.2 (1st stage) and 2.6 (2nd stage) BVR=constant

Several first and second stage Sullair compressors where tested in 1993 at the SSCL. TABLE 1.1 summarizes the compressor information.

TABLE 1.1 – SSCL Sullair Compressor Data

	1st Stage	**2nd Stage**
Model #	C25LB704-2.2-250Hp	C25MA704-2.6-700Hp
Max. Pressure	10.3 barg (150 psig)	20.7 barg (300 psig)
Motor Size	186 kW (250 Hp)	522 kW(700 Hp)
Rotor Diameter	255 mm	255 mm
Length to Dia. Ratio	1.70	1.25
BVR	2.2	2.6
Displacement @ 3550 RPM	2148 m^3/hr (1264 CFM)	2905 m^3/hr (1710 CFM)
Nominal Oil Charge	132 L (35 gal.)	227 L (60 gal.)

Test #1: SSCL Compressor Fixed BVR (cont.)

FIGURE 1.1 and FIGURE 1.2 show the 1st and 2nd stage volumetric efficiencies. As seem from these figures, the efficiencies are primarily a function of the pressure ratio (given stage and BVR).

Note the dependency on discharge pressure, in the operating range, is weak.

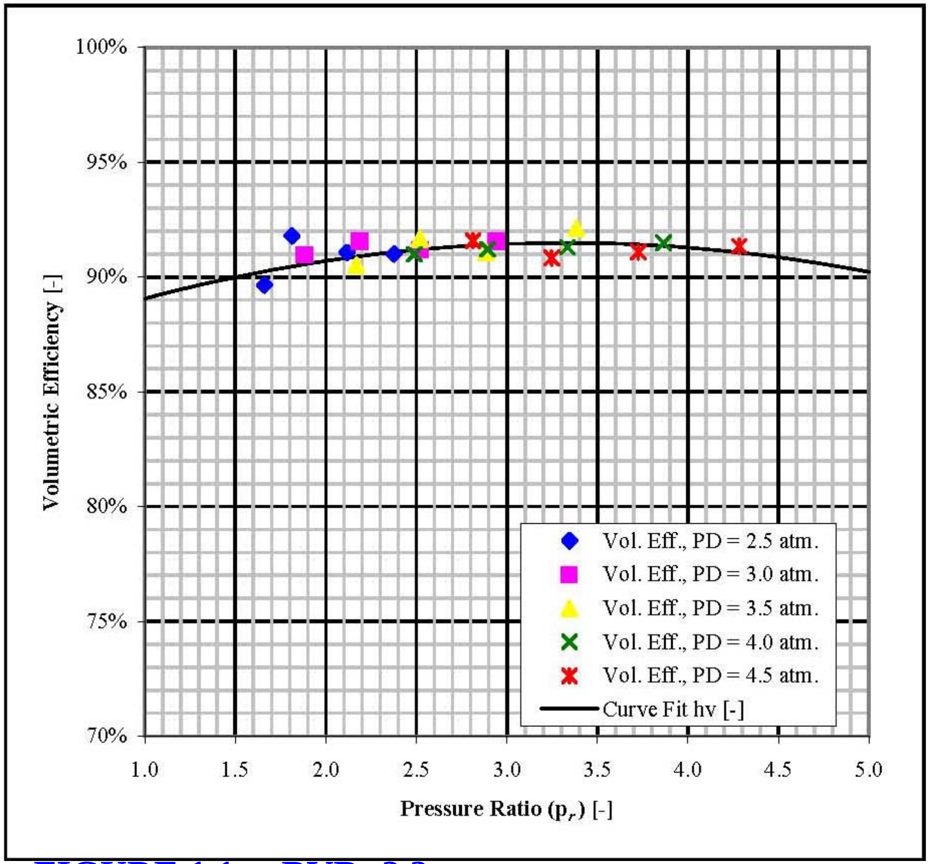

FIGURE 1.1 BVR=2.2

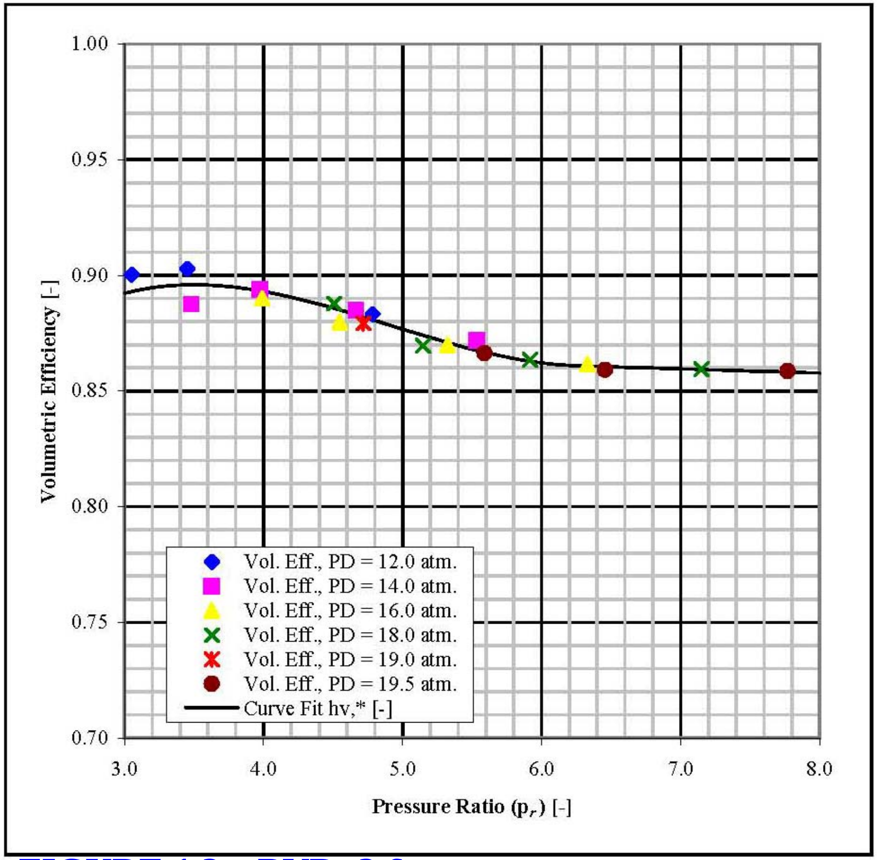

FIGURE 1.2 BVR=2.6

Test #1: SSCL Compressor Fixed BVR (cont.)

FIGURE 1.3 and FIGURE 1.4 show the 1st and 2nd stage isothermal efficiencies (for a given stage and BVR). The dependency on discharge pressure is also weak. Note values of pressure ratio at peak efficiency.

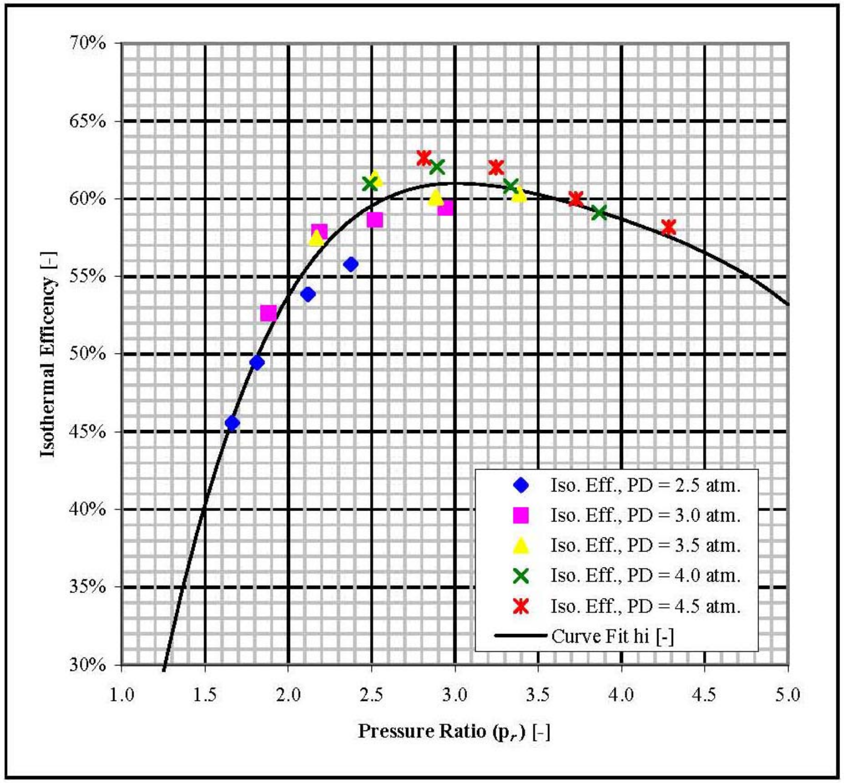

FIGURE 1.3 BVR=2.2 1st Stage

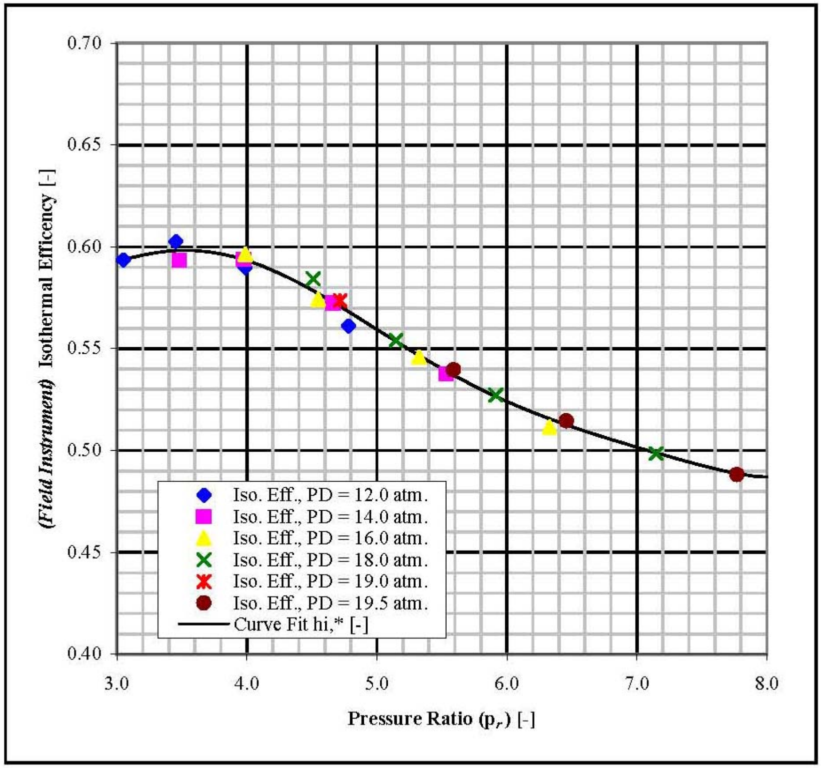

FIGURE 1.4 BVR=2.6 2nd Stage

Test #1: SSCL Compressor Notable Observations

- For fixed BRV, isothermal efficiencies are fairly <u>independent</u> of discharge pressures.
- For this case, isothermal efficiencies vary with pressure ratios .
- Peak efficiencies (at Pr~3-3.5) for 2nd stage occur at a lower pressure ratio than many existing systems (Pr=5 to 7). Many systems have 1,3,16/21 atm compressor header pressures.
- Possible efficiency improvement if the stage pressure ratios were balanced (Pr~ 3-3.5) and if the 1st stage suction pressure were allowed to float up as high as 1.8 atm.

Q: How does these observations change with varying BRV and fixed pressure ratio? We'll now look at Test #2

Test #2, Jlab 2nd Stage Compressor Pr=Constant=7.24

In January 2006 one of JLab's 2nd stage compressors was changed out and replaced with a new Howden variable BVR compressor. Some limited testing at a fixed pressure ratio was performed for several BVR settings. Variable pressure ratio testing was not practical due to restrictions on the refrigerator's required performance. The mass flow rate was estimated using an assumed volumetric efficiency (no individual flow meter is available for this compressor), so that the dependence of isothermal efficiency on the BVR setting could be examined. Also, the accuracy of power input measurement was not verified, but is believed to be close. To achieve the objective of the testing, it was not important to have accurate mass flow and power input measurements. Rather, it was important for these measurements to be consistent so that the behavior of the isothermal efficiency (as the BVR is varied) can be studied.

Test #2, Jlab 2nd Stage Compressor Pr=Constant=7.24 (cont.)

The compressor model number was a Howden MK6AS/WLVIH321165/685; which has a 321 mm male rotor with a length to diameter ratio of 1.65 and a swept volume of 94.61 m^3/s (3341 CFM) at 3550 rpm. The compressor is coupled with a 1678 kW (2250 Hp) Westinghouse motor. Typical suction and discharge pressures are 2.57 bar and 18.6 bar, respectively. (Pr=7.24)

The BVR setting was adjusted from 2.2 to 4.0. FIGURE 2.1 shows the behavior of the isothermal efficiency with respect to the BVR setting.

Test #2, Jlab 2nd Stage Compressor Pr=Constant=7.24 (cont.)

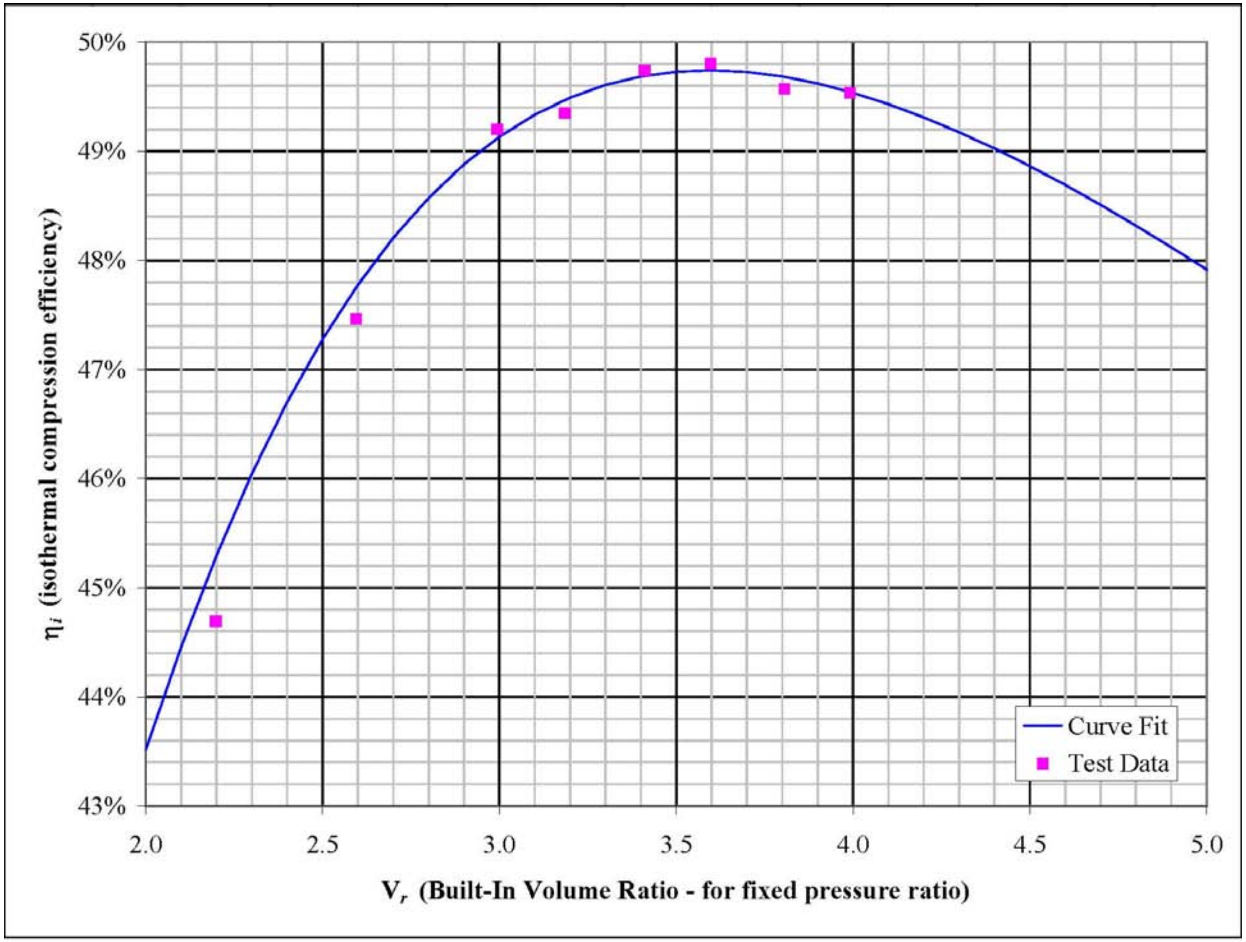

FIGURE 2.1, Jlab 2nd Stage Compressor, Pr=constant=7.24

Test #2, Jlab 2nd Stage Compressor Notable Observations

- Analysis of data revealed that the oil and helium were not in thermal equilibrium at the discharge. In fact there was probably a significant temperature difference between the helium and oil.
- The isothermal efficiency varies with the pressure ratio held constant and with variation of the BVR.
- The peak isothermal efficiency at a BVR of ~3.5.
- With a fixed pressure ratio of 7.24 and characterized as a 2nd stage, the resultant Howden isothermal efficiency is fairly close to the results of the SSCL 2nd stage Sullair compressor for a Howden BRV ~2.6, Refer to FIGURE 1.4

Q: With isothermal efficiency varying with both pressure ratio and BRV, what are the combined peak efficiency operating domains which take advantage of both in achieving peak efficiency? We'll take a look at the results of Test #3.

TEST #3, SNS Compressors Variable BRV and Pr

In March 2006, a series of tests were performed on the 1st and 2nd stage Howden compressors at SNS varying both BVR and pressure ratio. The tests were performed for BVR's ranging from 2.2 to 3.8. 1st stage pressure ratios were varied from 2.57 to 3.80 and 2nd stage pressure ratios were varied from 3.64 to 7.46. TABLE 3.1 summaries the compressor data.

TABLE 3.1 – SNS Howden Compressor Data

	1st Stage	**2nd Stage**
Model #	MK6S/WLVI321165/607	MK6S/WLVIH321165/604
Motor Size	447 kW (600 Hp)	1864 kW(2500 Hp)
Rotor Diameter	321 mm	321 mm
Length to Dia. Ratio	1.65	1.65
BVR	2.2-5.0	2.2 – 5.0
Displacement @ 3550 RPM	94.61 m^3/s (3341 CFM)	94.61 m^3/s (3341 CFM)
Nominal Oil Charge	594 L (157 gal.)	397 L (105 gal.)

TEST #3, SNS Compressors Variable BVR and Pr (cont)

FIGURE 3.1 and FIGURE 3.2 show the volumetric efficiencies for 1st and 2nd stage compressors. FIGURE 3.3 and FIGURE 3.4 show the isothermal efficiencies for the 1st and 2nd stage compressors. The coalescers in the bulk oil separation for these compressors were under-sized for some of the test conditions and consequently, there is a substantial amount of helium bypass is required to drain the oil. This is reflected by volumetric and isothermal efficiencies that are poorer than the SSCL compressor data. Additionally, oil drain settings had to be adjusted during testing since the oil carry-over increased at higher mass flows and lower discharge pressures.

TEST #3, SNS Compressors Variable BRV and Pr (cont)

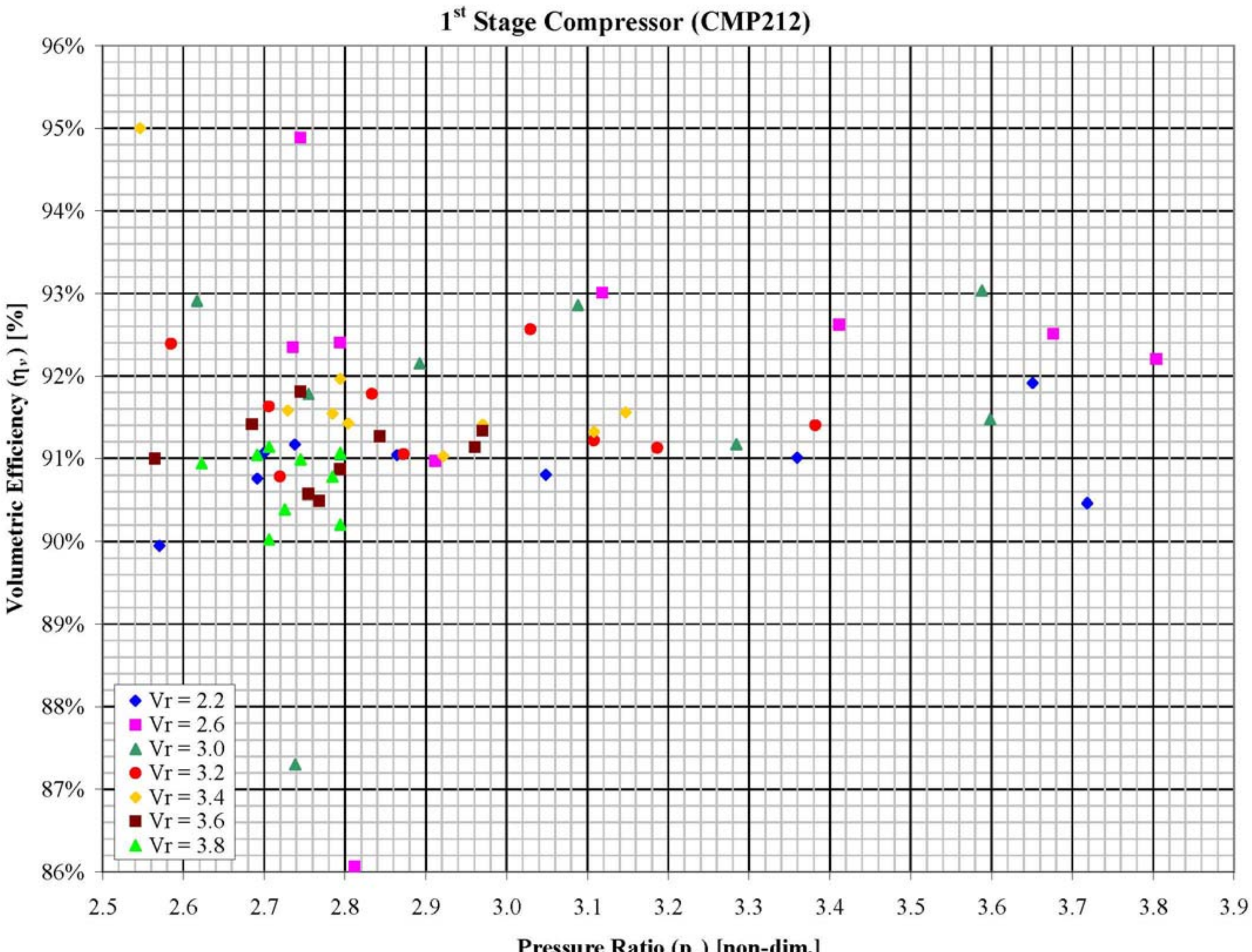

FIGURE 3.1 1st Stage Volumetric Efficiency

TEST #3, SNS Compressors Variable BRV and Pr (cont)

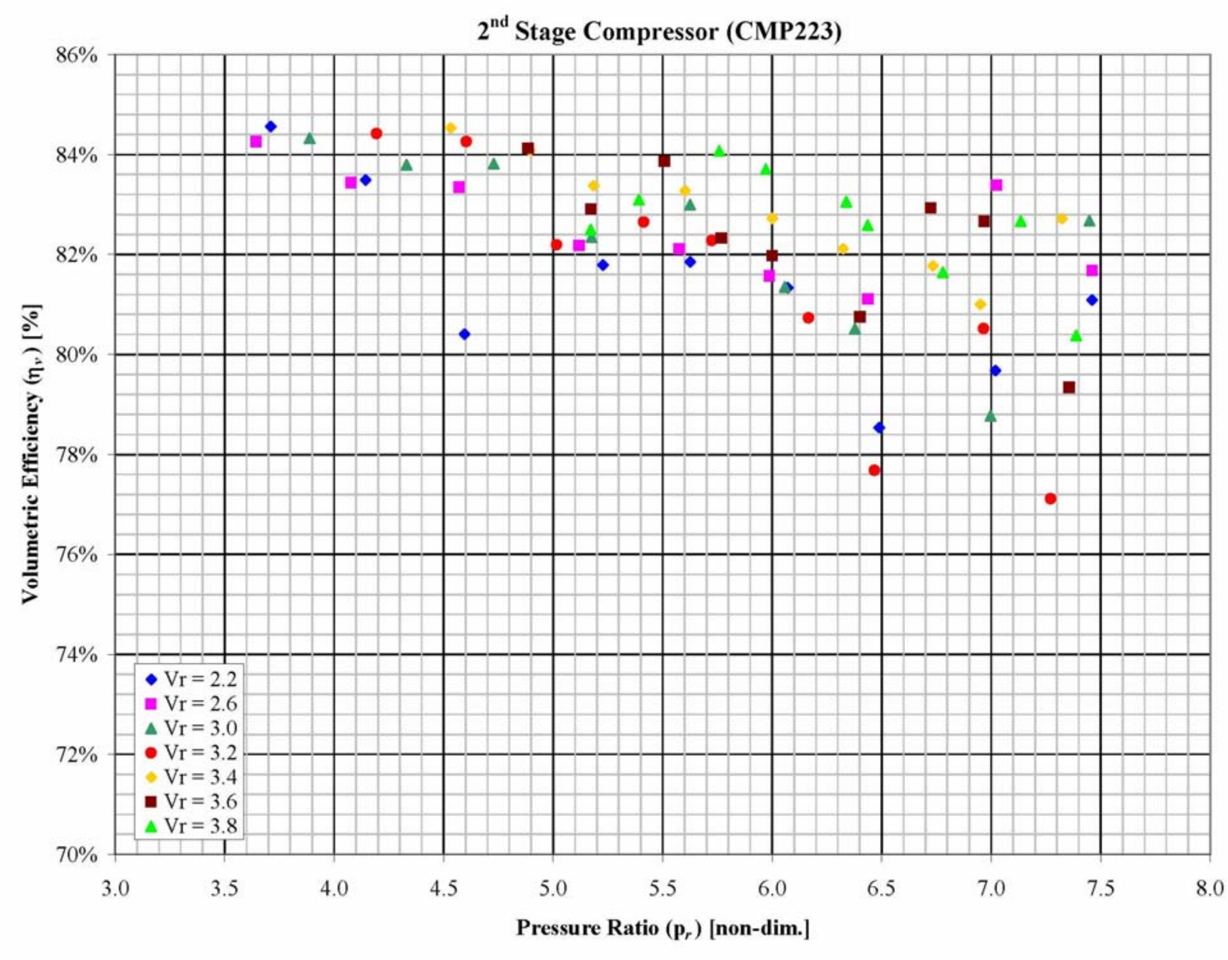

FIGURE 3.2 2nd Stage Volumetric Efficiency

TEST #3, SNS Compressors Variable BRV and Pr (cont)

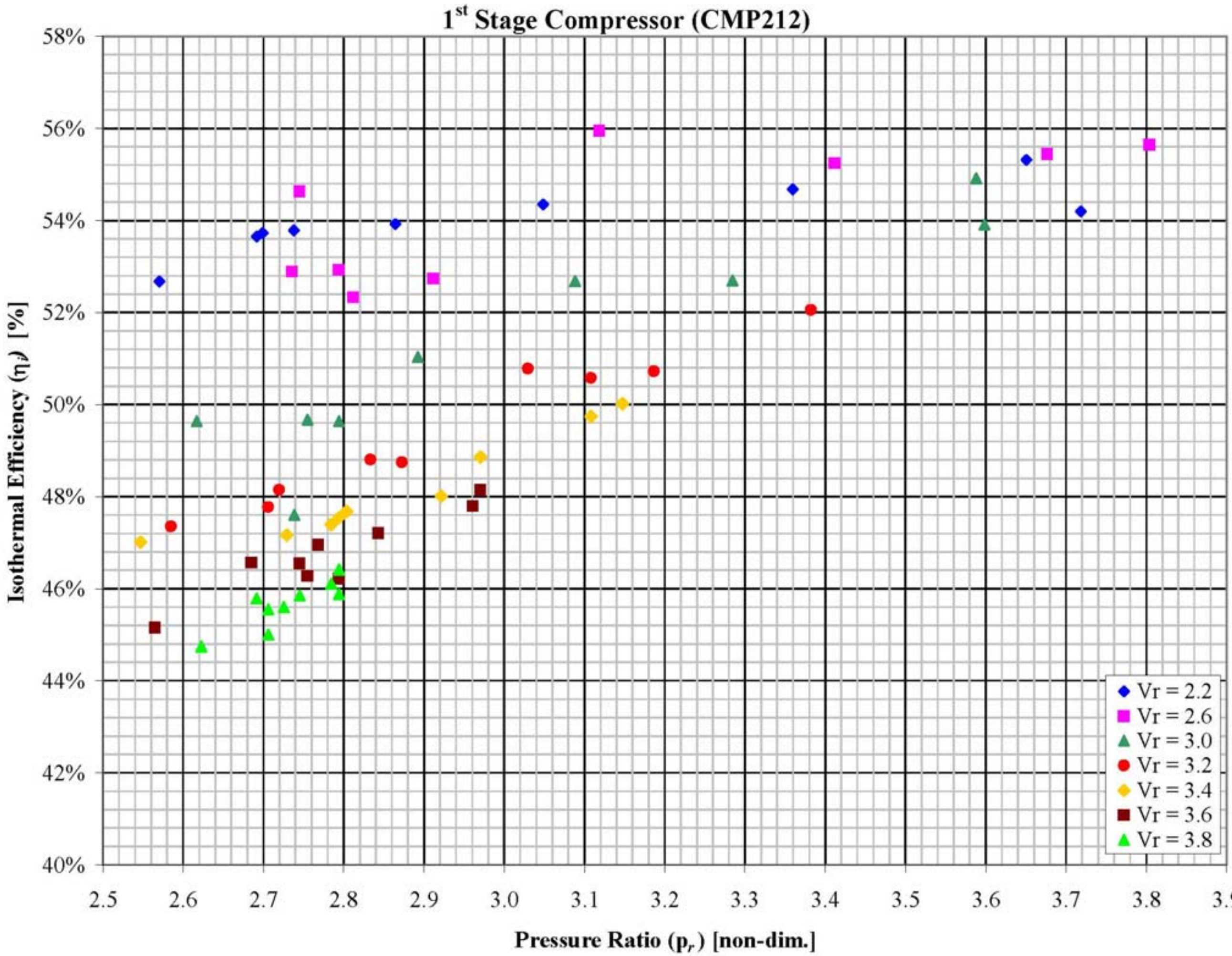

FIGURE 3.3 1st Stage Isothermal Efficiency

TEST #3, SNS Compressors Variable BRV and Pr (cont)

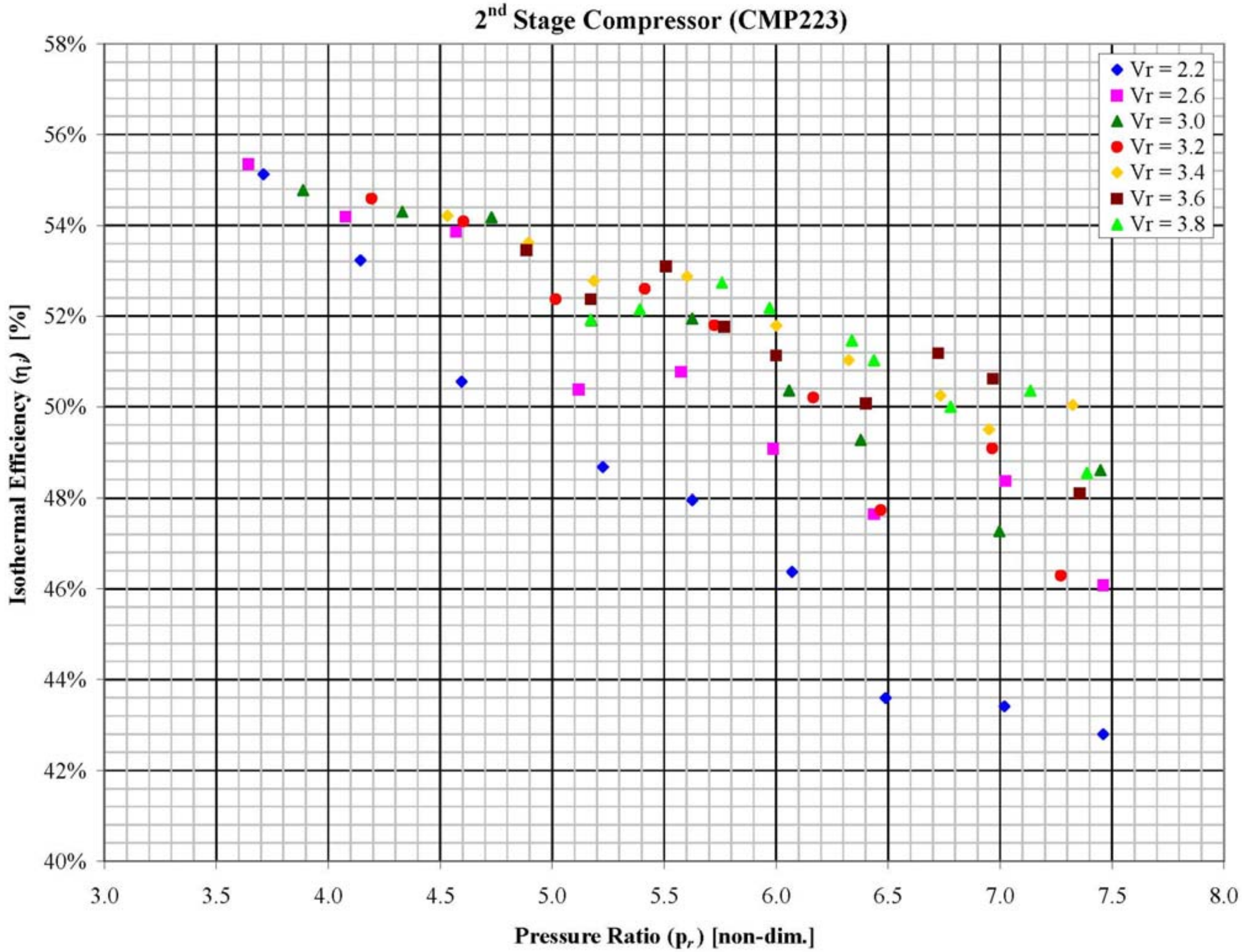

FIGURE 3.4 2nd Stage Isothermal Efficiency

TEST #3, SNS Compressors Notable Observations

- 1st Stage volumetric efficiencies were reasonably stable for low BRV values (2.2 to 2.6) for all pressure ratios
- 1st Stage isothermal efficiencies were reasonably high and stable for BRV values (2.2 to 2.6) for pressure ratios >3
- 2nd Stage volumetric efficiencies peaked for all BRV values and pressure ratios ~3.5
- 2nd Stage isothermal efficiencies peaked for all BRV values for pressure ratio ~3.5

Q: How could these peak efficiency domains affect new compressor designs? We will look at this in the Test Summary and Further Discussion on System Designs.

Test Summary

The SSCL compressor testing provides a good baseline for 1st and 2nd stage compressor performance for fixed BVR's. From the SNS compressor testing, the optimal BVR for a 1st stage compressor appears to be 2.2 to 2.6, for a 2nd stage compressor 3.0 to 3.4 (depending on the nominal operating pressure ratio). Since the SSCL 2nd stage compressor testing was done with a BVR of 2.6, it appears that with a more optimal BVR setting, the predicted 2nd stage compressor performance could be better than the test data by 1-2%. Further testing is required to confirm the actual improvement. Also, as indicated in Test #2, the overall performance of oil injected screw compressors can be significantly improved by injecting the oil into the helium in a more atomized fashion; so that the helium and oil are in better thermal equalibrium.

Further Discussion on System Designs

The tests opens new discussion for the selection of compressor system header pressures. There is a long design history of anchoring the first stage compressor suction pressure to ~1 atm to regulate load return pressures. This however places substantial limitations on the overall system efficiencies as to setting the system stage pressure ratios. One also must realize that the return load flows are the lowest of the entire system. In essence the pressure regulation of a small flow is restricting overall system efficiency. …… "The tail wagging the dog"

Solutions to this inefficiency can be resolved by application of new cycle designs such as the Jlab Ganni Cycle. Load return flows at 1 atm are compressed separately by a smaller compressor to the compressor system interstage header. Inefficiencies of the small flow are then confined to the smaller compressor. The main 1st stage compressor suction is then allowed to float up as high as 1.8 atm. Each of the main compressor stage pressure ratios are then balanced to ~3 to 3.5 each with the appropriate BRV values discussed. Lower 2nd stage efficiencies due to pressure ratios of 6-7 are eliminated. Traditional 2nd stage discharge pressure of 16-21 are easily reached but now with increased compressor isothermal efficiency. For system turn down efficiency, all main compressor header pressures are reduced to maintain pressure ratios. Load return pressure continue to be maintained by the smaller compressor. These new system pressure ratios fall within the normal high efficiency pressure ratios for turbines. In this sense, both high efficiency operation of compressors and turbines are realized and the two systems are matched over a large operating envelop of capacities and modes.

A Word of Appreciation

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