JLab's 12 GeV Compressor System and Development Work
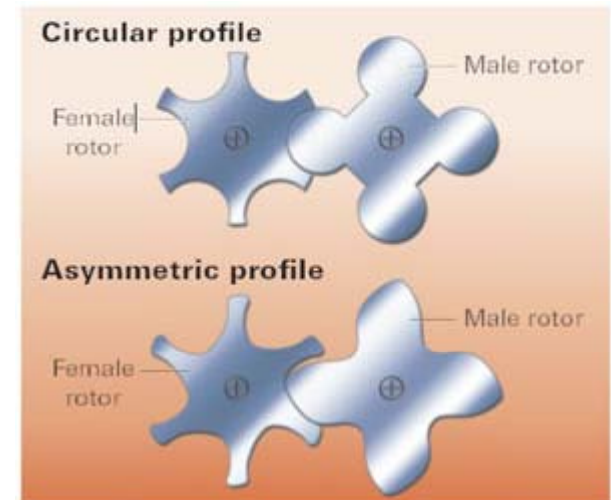
P. Knudsen

Cryogenics Group

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JLab's 12 GeV Compressor System

- Most helium refrigeration systems use rotary screw compressors

- What are they…

  - Also known as twin screw compressor

  - A pair of meshing helical rotors within a casing

  - Male, or drive, rotor has convex lobes

  - Female, or gate, rotor has concave flutes

  - (Modern) Lobes and flutes have an asymmetric profile

Courtesy of Howden Compressors

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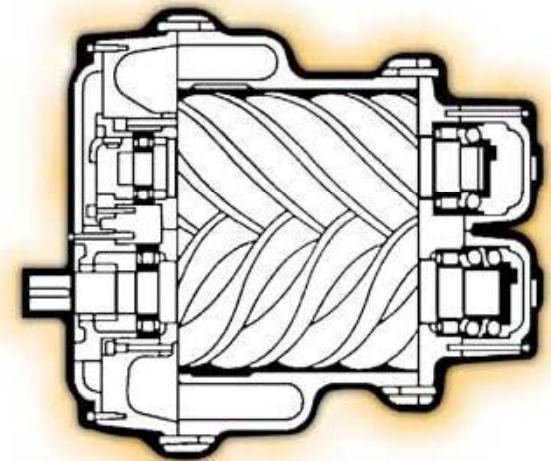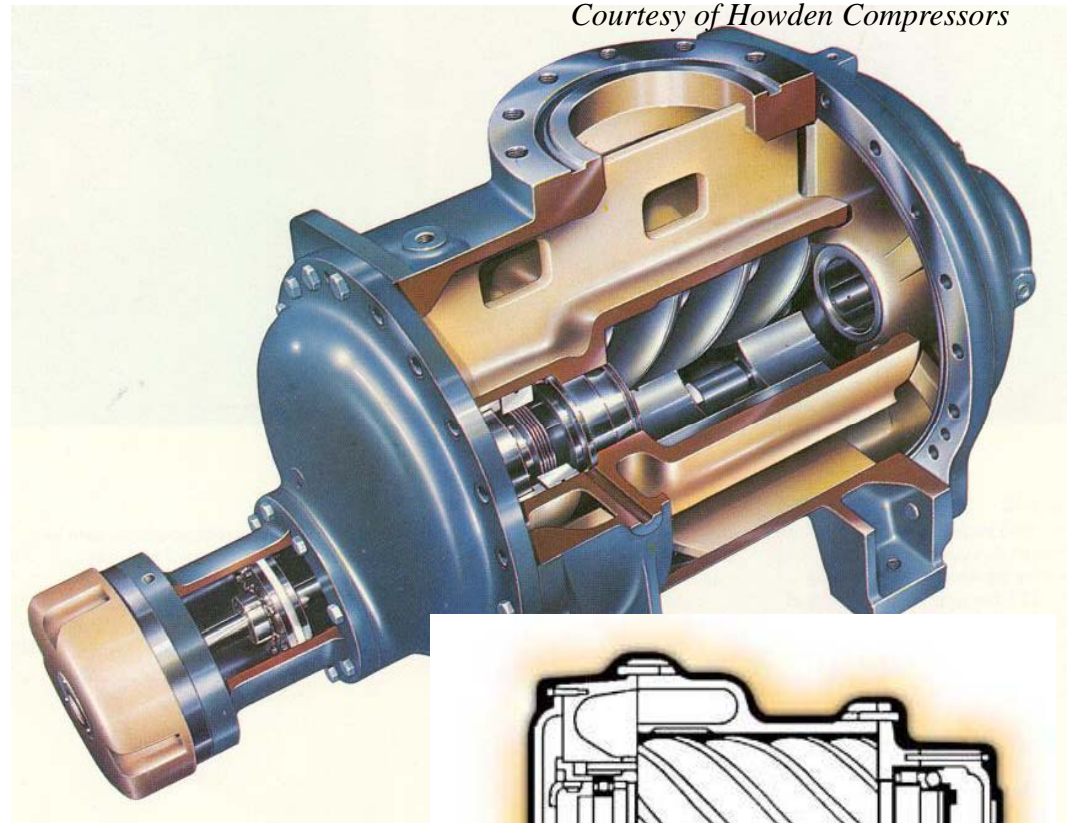
JLab's 12 GeV Compressor System

- ## What are they…
  - Basic rotor characteristics
    - \# male lobes, \# female flutes
      - » 4+6 (or 4/6) is common
    - rotor (male) diameter (D)
    - (L/D) – length to rotor diameter ratio
      - » ~1.4 to 2.1
    - Wrap (or helix) angle
      - » typically 300° for compressors)
    - Built-in-volume ratio (BVR)
      - » ~2.1 to 5.8
    - Usually directly coupled to an electric motor
      - » Requires ~80% starting torque
      - » This allows across the line starting (i.e., no transformer is required)

Circular profile — Male rotor, Female rotor

Asymmetric profile — Male rotor, Female rotor

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JLab's 12 GeV Compressor System

- ## What are they…

Courtesy of Sullair

Thomas Jefferson National Accelerator Facility

JLab's 12 GeV Compressor System

- Approximate size/performance range presently available,
    - Rotor diameters, ~50 to 800 mm (male)
    - Capacities, ~0.6 to 600 m³/min
    - Pressure Ratio, ~3.5 for dry, up to ~15 for oil-flooded
    - Nominal pressure difference, up to ~15 bar
    - Max. pressure difference, ~40 bar
    - Volumetric efficiencies ≈ 90%
- Used,
    - Widely, in the process industry; especially in commercial refrigeration and the oil & gas industry
    - As air compressors
    - For supercharging internal combustion engines
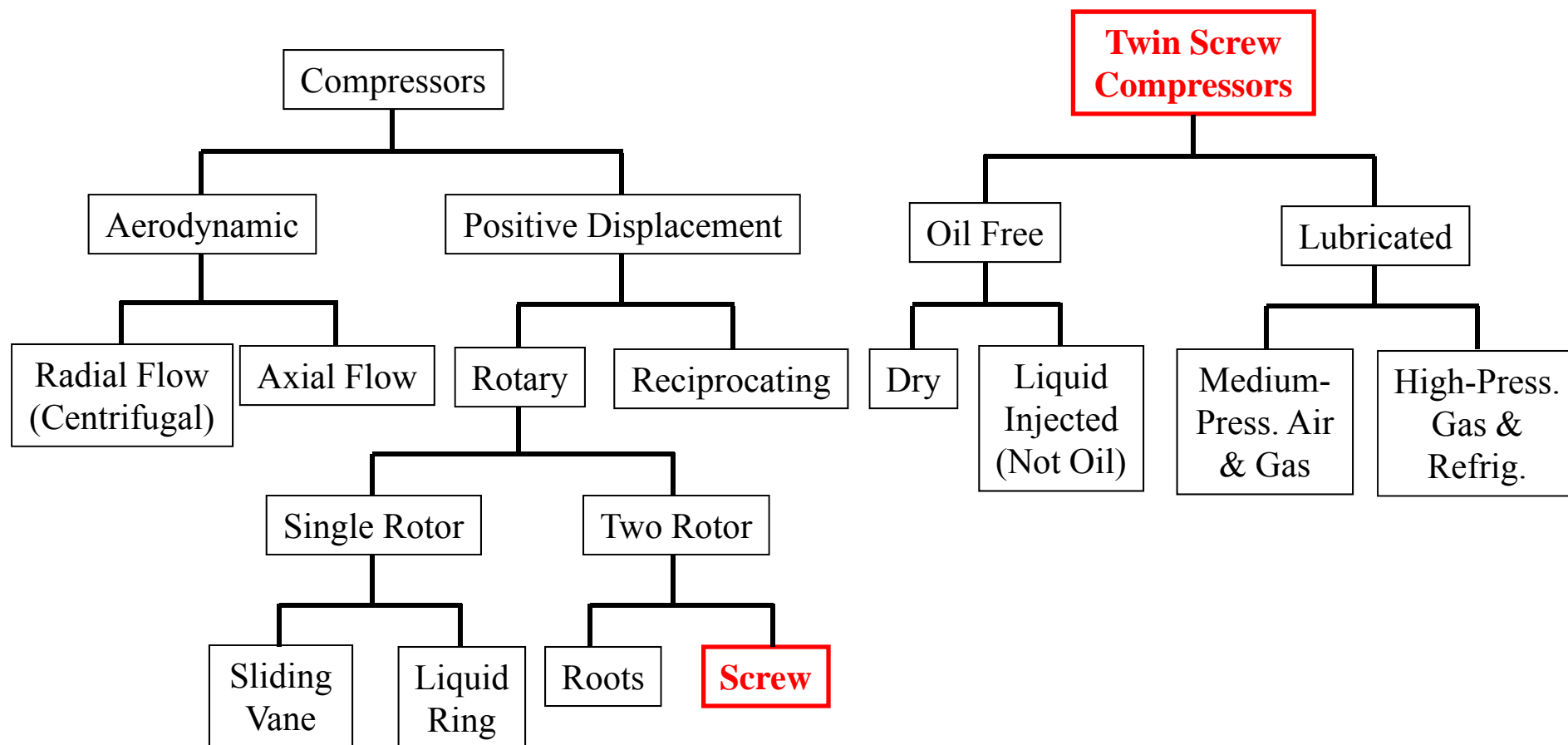   - As expanders

JLab's 12 GeV Compressor System

- Types
  - Dry, Oil-Free
    - One rotor drives other using timing gears (to prevent physical contact of rotors)
    - Use seals to prevent oil lubricating the bearings and gears from entering the process
    - Are considerably more expensive to manufacture than oil-flooded
  - Lubricated or Oil-Flooded
    - Oil enters through bearings and/or by direct injection into compression cavity
    - *Development of the successful lubricated screw compressor has been the most important development in its history* – by increasing obtainable pressure ratios with moderate discharge temperatures

JLab's 12 GeV Compressor System

- What are they…
  - They are positive displacement devices

```
                    ┌──────────────┐                          ┌──────────────────┐
                    │ Compressors  │                          │   Twin Screw     │
                    └──────┬───────┘                          │   Compressors    │
              ┌────────────┴────────────┐                     └────────┬─────────┘
     ┌────────┴────────┐    ┌───────────┴──────────┐        ┌──────────┴──────────┐
     │   Aerodynamic   │    │ Positive Displacement │        │ Oil Free │  │ Lubricated │
     └────────┬────────┘    └───────────┬──────────┘        └────┬─────┘  └─────┬──────┘
        ┌─────┴──────┐          ┌────────┴────────┐
```

Compressors				

Aerodynamic — Positive Displacement

Radial Flow (Centrifugal) · Axial Flow · Rotary · Reciprocating

Rotary → Single Rotor · Two Rotor

Single Rotor → Sliding Vane · Liquid Ring

Two Rotor → Roots · **Screw**

Twin Screw Compressors → Oil Free · Lubricated

Oil Free → Dry · Liquid Injected (Not Oil)

Lubricated → Medium-Press. Air & Gas · High-Press. Gas & Refrig.

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- What are they…
  - As compared to a piston compressor
    - Fewer moving parts
      - Typical MTBF for those used in helium systems is ~60,000 hrs
    - Dynamic balance and low vibration; since it is rotary and has effectively continuous flow
    - Higher operating speed
    - No clearance volume (at the end of the stroke)
   - Can tolerate ingestion of liquid in an amount that would damage a piston machine

JLab's 12 GeV Compressor System

- What are they…
    - As compared to centrifugal compressors
        - Does not surge
        - Can tolerate higher levels of liquid ingestion
        - Lower operating speed; screw compressors can be directly coupled to an electric motor
    - As compared to vaned compressors
        - Contact forces are low
    - As compared to piston, scroll and vaned compressors
        - Contact sealing line length decreases as the compression volume decreases; so as the pressure is increasing, the area available for gas leakage decreases

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- What are they…
    - Typical characteristic of positive displacement vs. aerodynamic (centrifugal) devices
        - Pressure vs. flow rate

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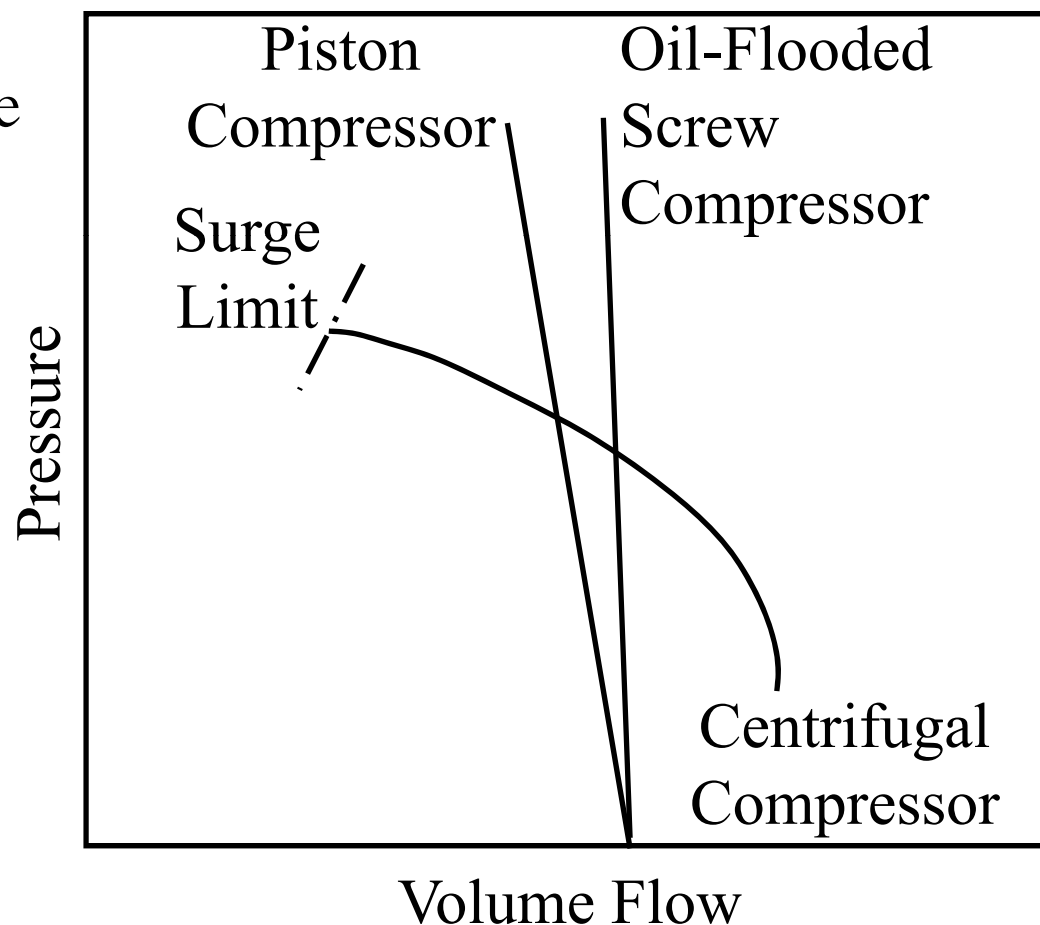
JLab's 12 GeV Compressor System

- 1878 German patent by Herr Krigar
    - 2 male lobes + 2 female flutes (2+2, or 2/2)
    - Produced ~1.5 psi and was 'quiet'

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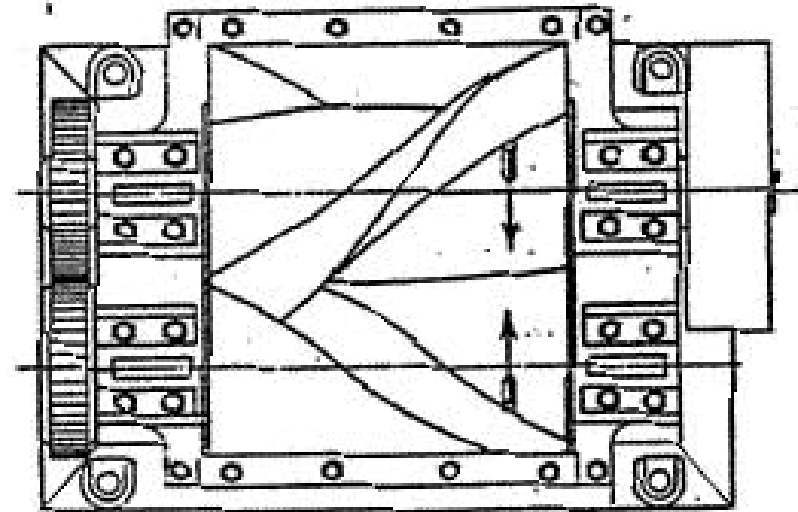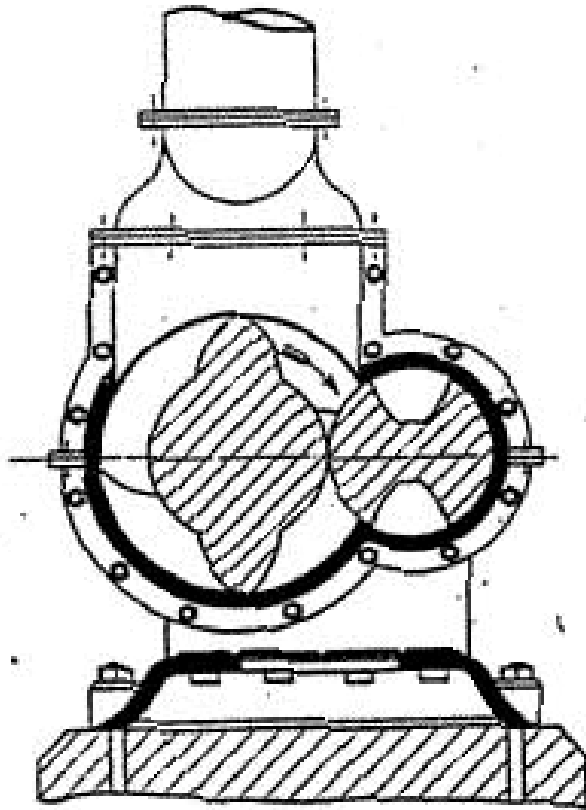
JLab's 12 GeV Compressor System

- Developed of modern screw compressor was done by Alf Lysholm in the 1930's, the chief engineer of the Swedish company then called The Ljungström Steam Turbine Company – now called Svenska Rotor Maskiner (SRM) AB

  - First Lysholm 3+3 rotor design, oil-free, achieved 2:1 compression ratio
  - First application was for turbo-prop engines
  - Tried asymmetric and symmetric circular rotor profiles
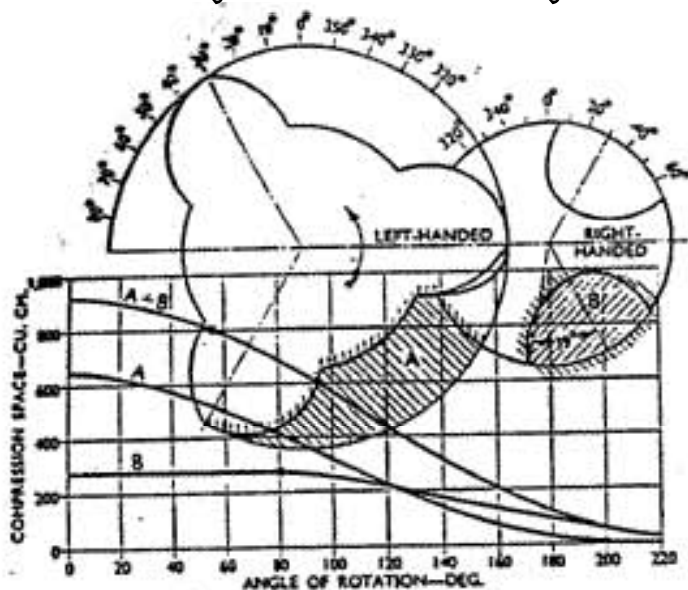
Fig. 4. Three-lobed Compressor
Diameter of male rotor, 150 mm.; of female rotor, 90 mm.; rotor length, 262 mm.
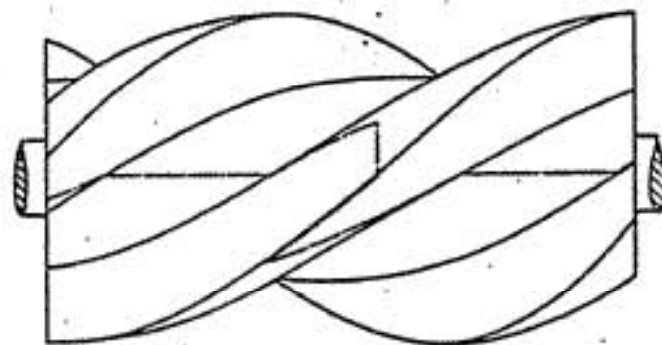
Fig. 5. Path of Sealing Line

between the lobes, but it is considerably shorter than that of the Roots type when higher compression is employed, owing to the shortening of the sealing line resulting from the reduced com-

JLab's 12 GeV Compressor System

- In 1946 SRM introduced symmetric circular rotor profile, which had less risk of damaging the sealing edges due to thermal distortion or errors in timing of rotors (still an oil-free design)
  - 4+6 rotor common, with both rotors of the same outside diameter

Female Rotor   Male Rotor
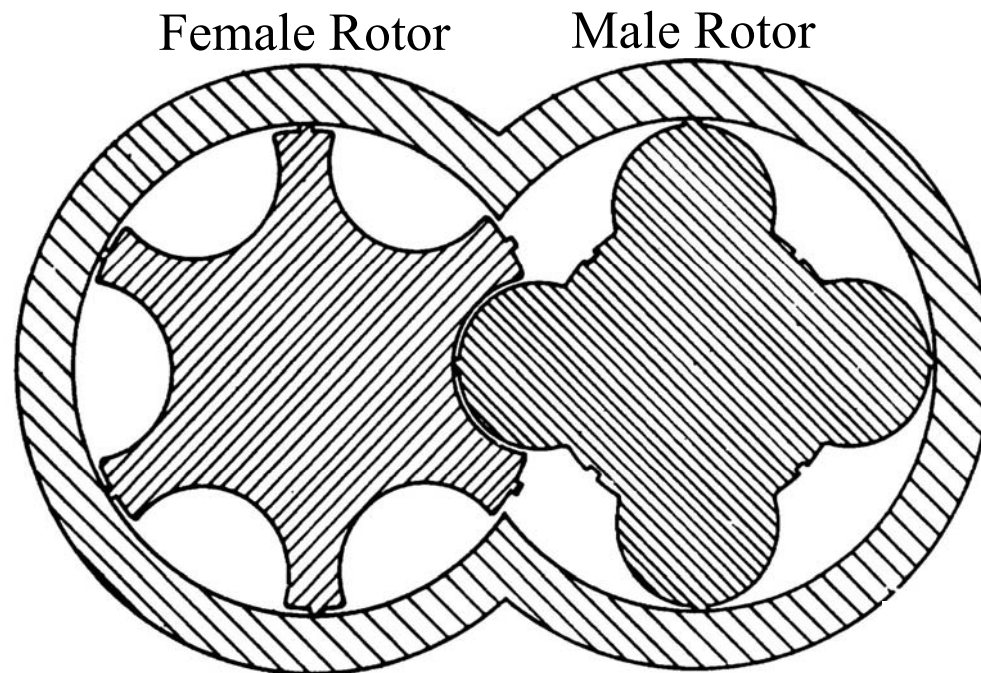
Ref. O'Neil, P., "Mechanical design and efficiency of screw compressors", presented at the Institution of Mechanical Engineers, London, 1966, p.4 Fig 1(b)

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- Oil-flooded compressors introduced in 1960's

- In 1960's SRM introduced the SRM asymmetrical rotor profile, which did not have the draw backs of Lysholm's asymmetric profile (abandoned in the late 1940's), resulting in 10-15% less power consumption than the symmetric circular profile

- Two factors accelerated the adoption of these machines

  - By the mid-1960's milling machines for thread cutting enabled the rotors to be manufactured accurately and with acceptable cost

  - In 1973, SRM introduced the SRM "A" profile, an asymmetric profile which reduced the internal leakage path area (known as the 'blow hole') by 90%.

    - This allowed the screw compressor to be built with efficiencies about equal to piston compressors and with stage pressure ratios of up to 8:1 in the oil-flooded type

    - This was unattainable using piston compressors
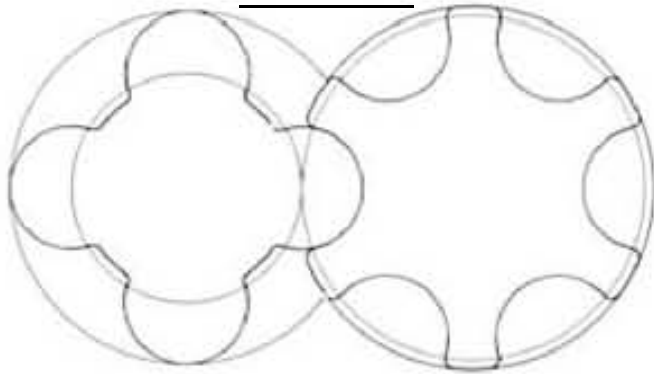
JLab's 12 GeV Compressor System

- Present machining accuracy allows tolerances in rotor on the order of 5 μm (0.2 mils), or better

- For an efficient screw compressor, rotor profile must have,
  - Large flow cross-sectional area
  - Short sealing line
  - Small "blow-hole" area

- Although 4+6 configuration with male and female rotors having equal outside diameters is very common
  - » 4+5 configuration appears the best for oil-flooded types at moderate pressure ratios
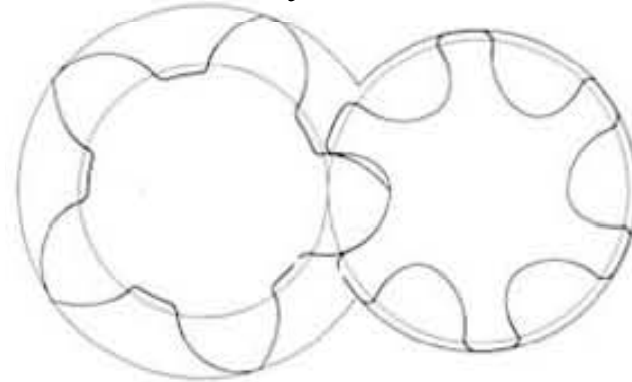 - » 5+6 configuration is becoming the most popular, both as an air compressor and for refrigeration and AC

JLab's 12 GeV Compressor System
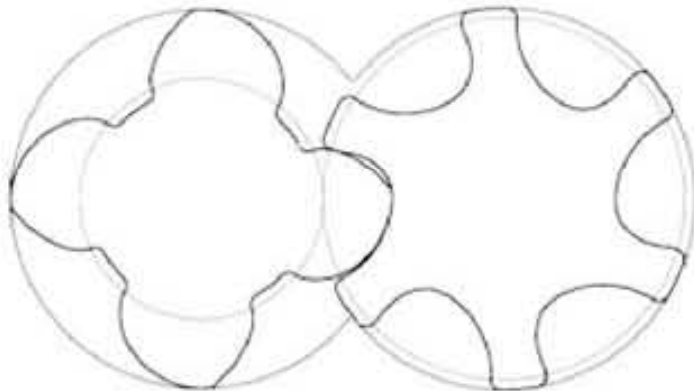
- Various rotor configurations
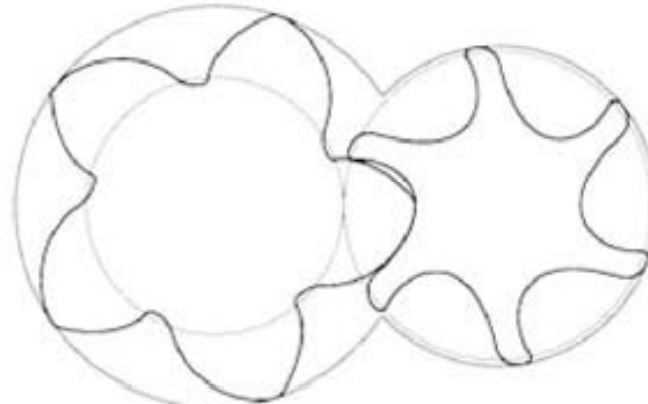
<center>

**SRM Symmetric
Circular**

**Lysholm's
Asymmetric**

SRM "A" Profile

Hyper Profile

</center>

*Ref. N. Stosic, I. Smith, A. Kovacevic, "Screw Compressors: Mathematical Modeling and Performance Calculation,"
Springer, 2009, p. 16, Fig. 1.6,*

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- For the oil-flooded type,
    - Contact force is small if the male rotor is driven; however, contact force is substantially higher if the female rotor is driven
    - Same oil used for bearing lubrication and oil flooding of rotors
    - Oil is injected where (calculations indicate) gas and oil temperatures are equal
    - Injection holes located so that oil enters tangentially in line with female rotor tip, in the direction of the advancing helix
    - Even though oil removes ~85% (in helium applications) of input power (as heat), the actual volume of oil as compared to the total displaced volume is less than 1%

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- Capacity control
  - Achieved using,
    - On/Off operation
    - Inlet throttling
    - Slide valve – shortens the effective length of the rotors
    - Variable speed (VFD)
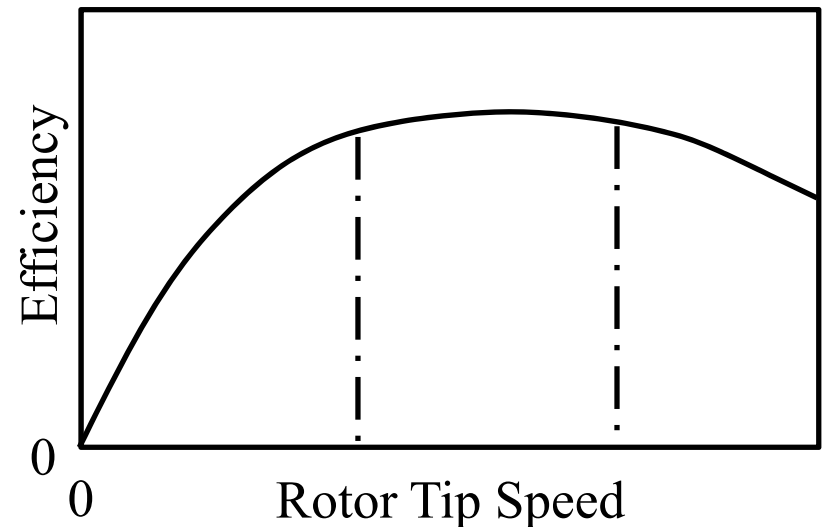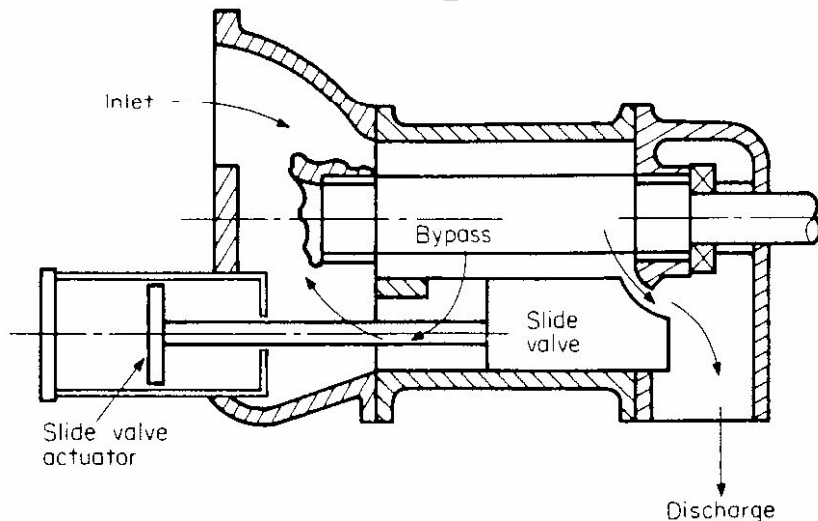
Ref. *E.A. Avallone, T. Baumeister III (Eds), "Mark's Standard Handbook for Mechanical Engineers,", 10th Ed., p.14-35, Fig. 14.3.17, McGraw-Hill*

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- Capacity control…
  - Effect of capacity control on input power required
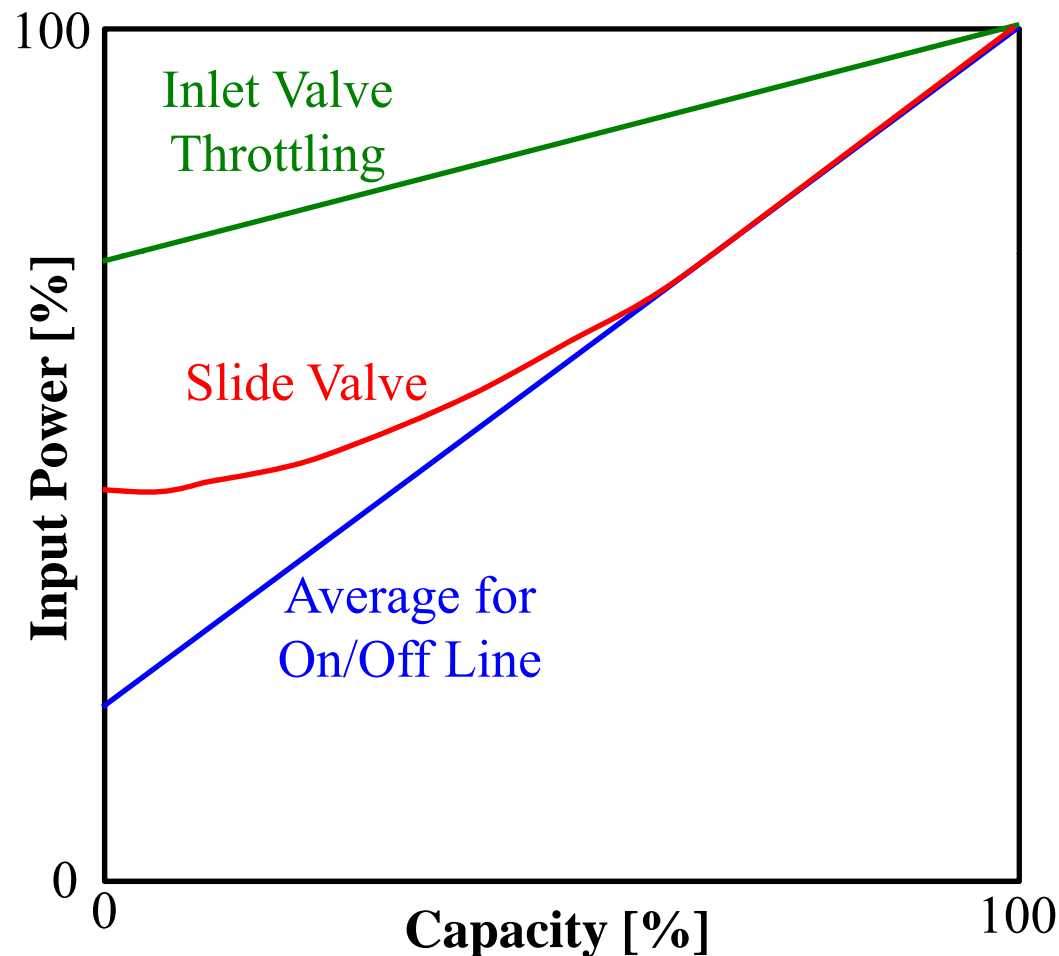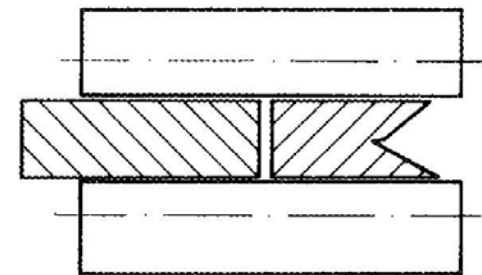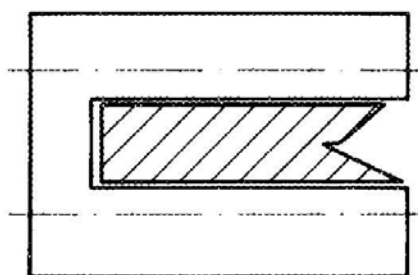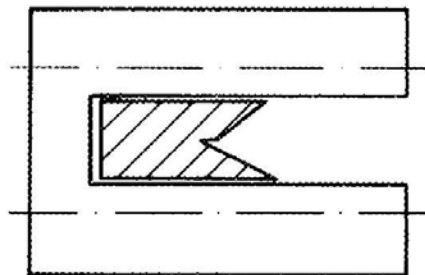
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- Built-in volume ratio (BVR) or "V_i"
  - It is common nowadays for this to be adjustable (though in some cases, not while running)
  - Three cases:
    - Over-compression
    - Under-compression
    - Matched-compression ▶ <u>minimum p–V work</u>
  - In general, it is better to under-compress,
    - Since the effect on the efficiency is less influential, and
    - To avoid extreme discharge temperatures (especially for helium, $\gamma = 1.67$)
    - Effect on the thrust bearings is less

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- Built-In Volume Ratio (BVR)
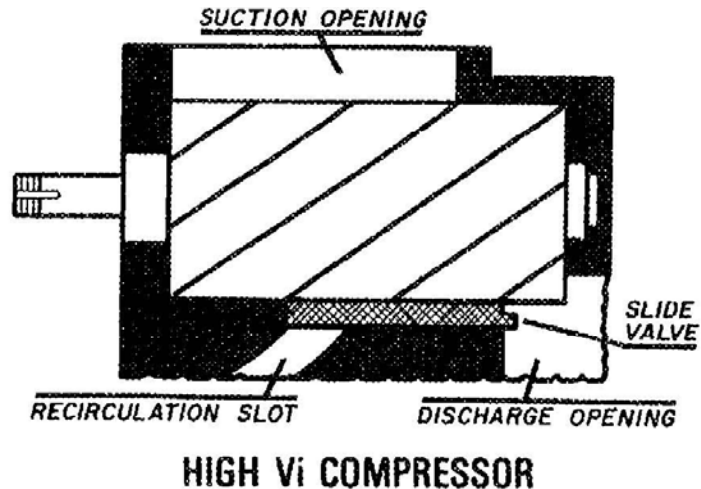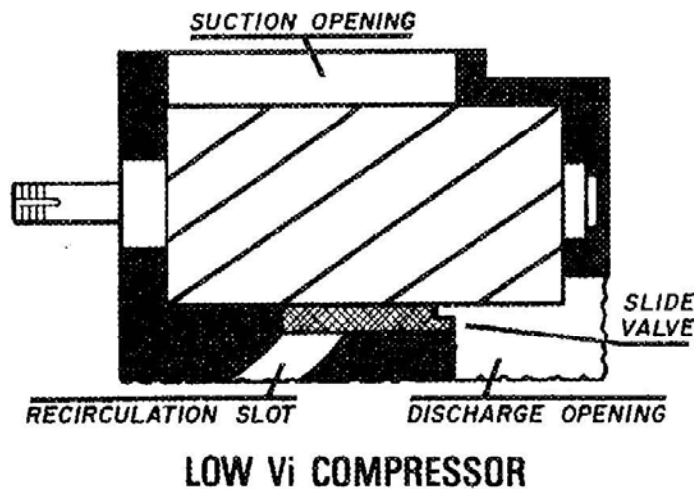
Ref. J.W. Pills, "Development of a Variable Volume Ratio Screw Compressor," IIAR Annual Meeting, April 17-20, 1983, p. 8, Fig. 5 & 6
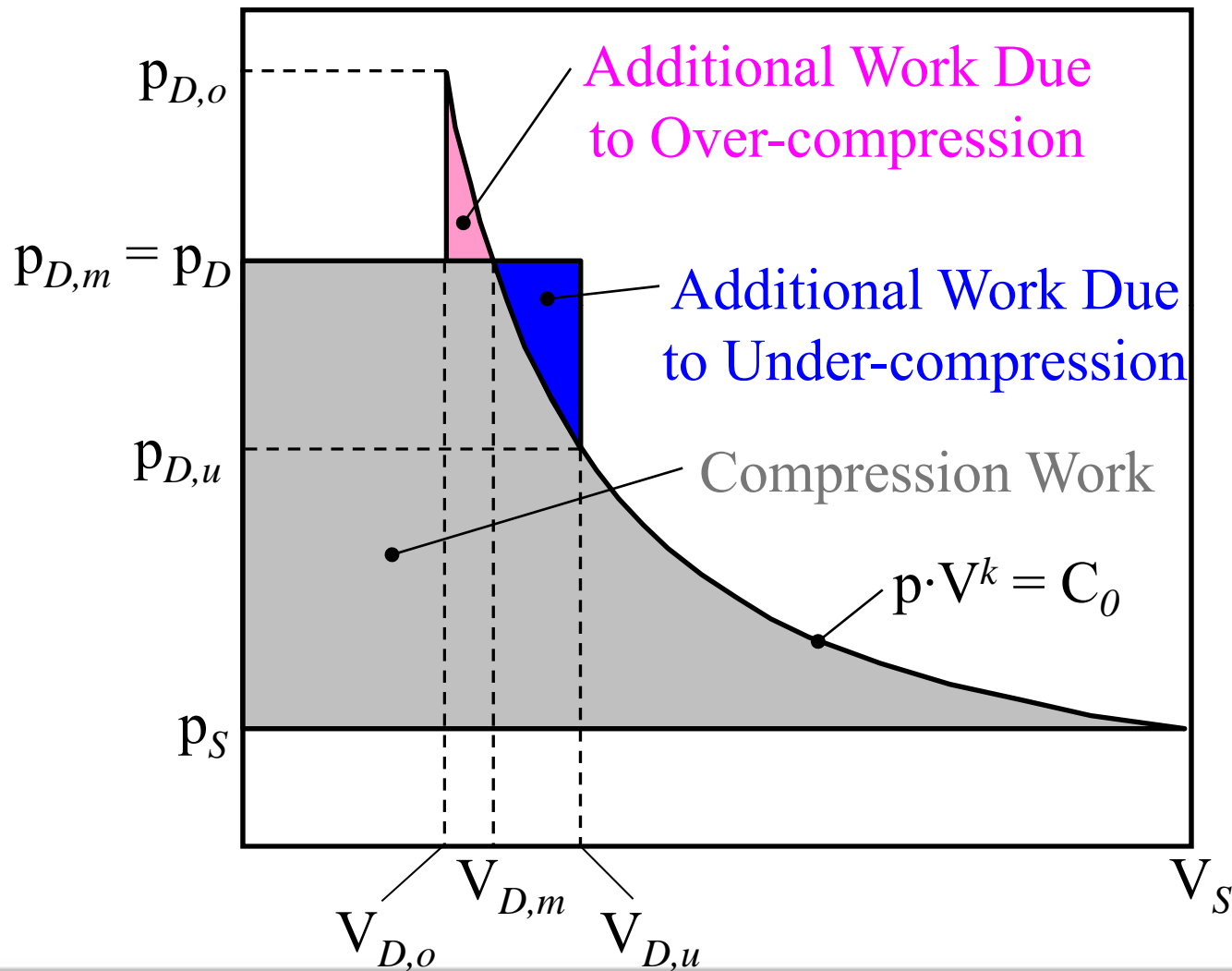
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- Effect of BVR adjustment on efficiency vs. pressure ratio
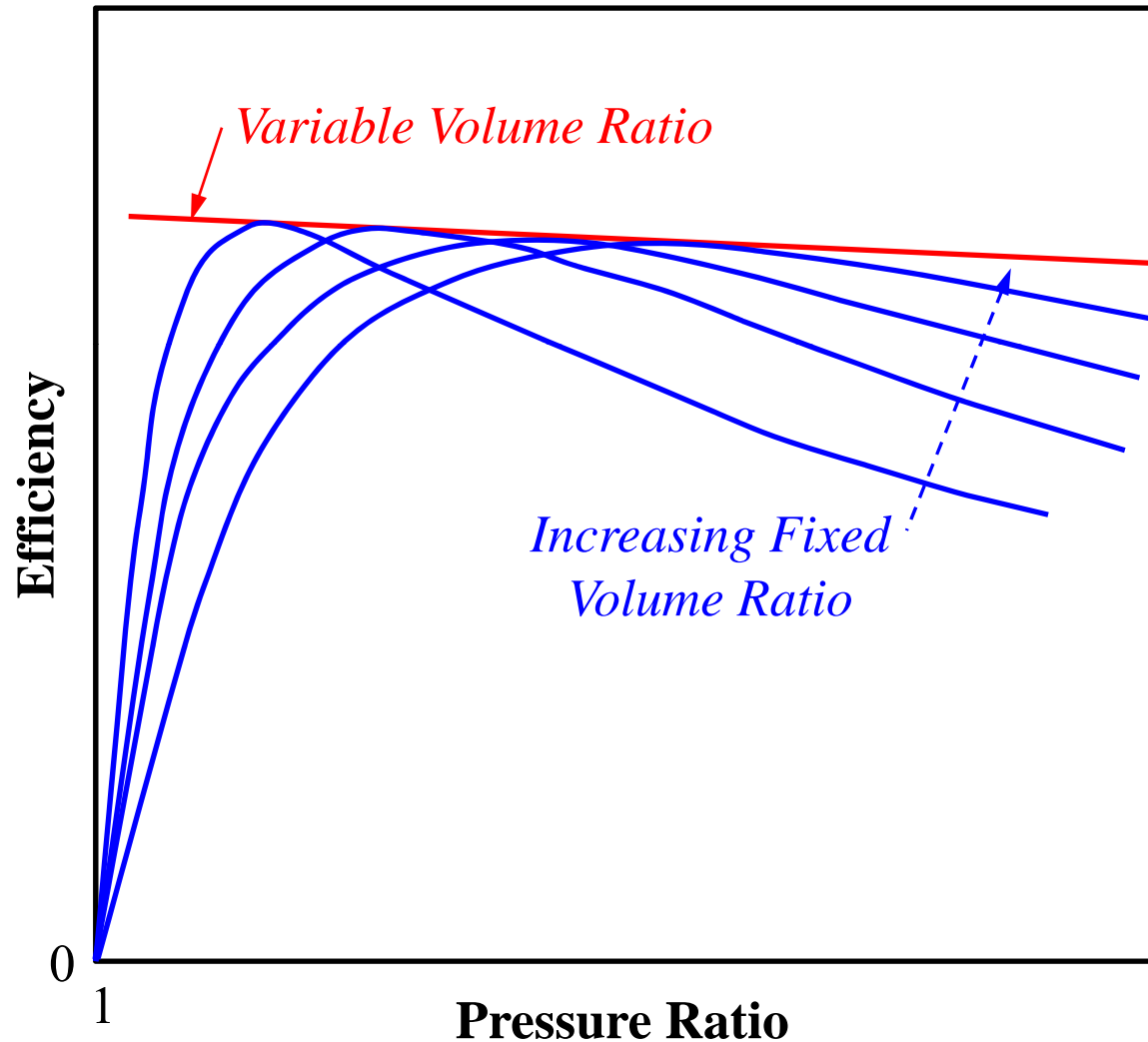
JLab's 12 GeV Compressor System

- Pressure-volume diagram of compression process

JLab's 12 GeV Compressor System

- Effect of polytropic coefficient (*k*) on final compression temperature
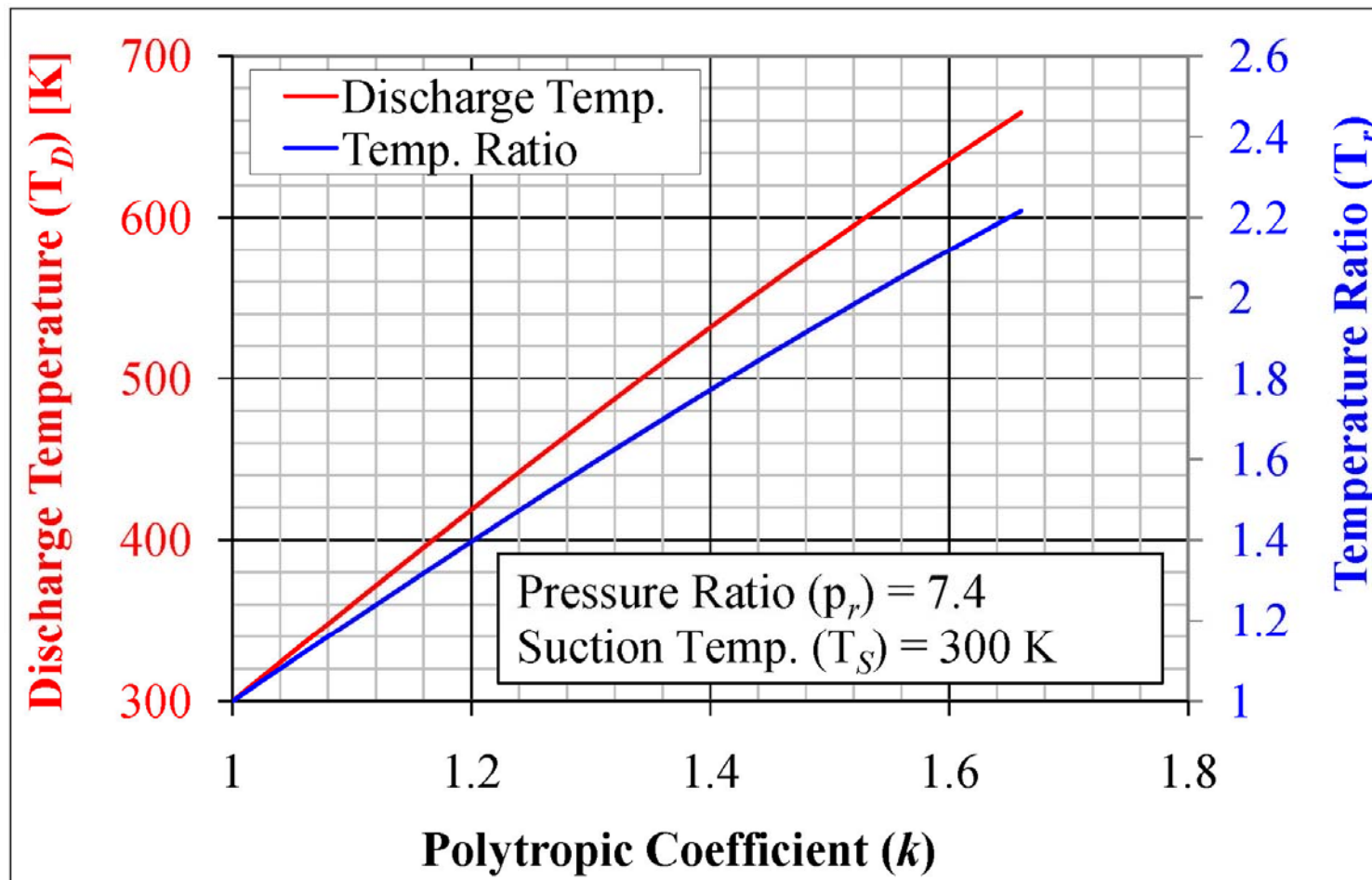
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- (Physical) exergy per unit mass is defined as,

  $$\varepsilon = h - T_0 \cdot s$$

  - where, T_0 is the reference temperature; i.e., environmental temperature; say, 300 K
  - exergy (ε) is an intrinsic fluid property (…like h and s)

- The minimum input power theoretically required; or conversely, the maximum power output theoretically possible is,

  $$\Delta E = W_{out,max} = -W_{in,min} = \Sigma \, m_{in} \cdot \varepsilon_{in} - \Sigma \, m_{out} \cdot \varepsilon_{out}$$

  - also, known as the reversible (input or output) power

$W_{out,max}$

m_{in}
ε_{in}

Process

m_{out}
ε_{out}
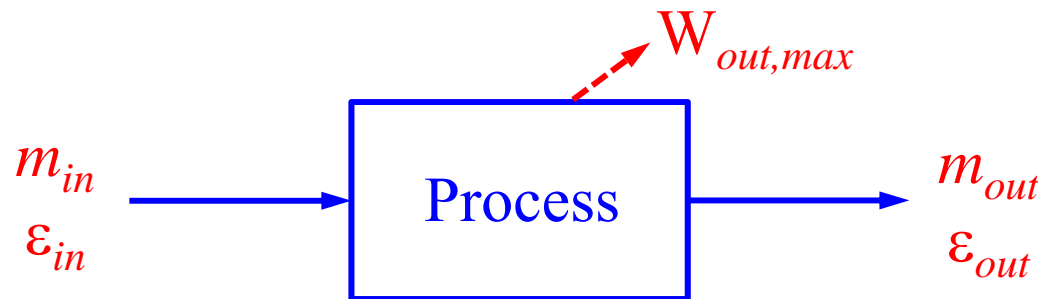
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- A common measure of process efficiency is the ratio of the ideal (or theoretical) input power to the actual (or real) input power required; known as the *overall exegetic efficiency*,

    $$\eta_C = \Delta E / W_{in}$$

    - Where, W_{in} is the actual (real) required input power

- For compressors, three efficiencies are commonly used,

    - *Isothermal efficiency*, T = constant

        $$\eta_T = \Delta E_T / W_{in} \; ; \text{where, } \Delta E_T \text{ is the exergy diff. at const. T}$$

    - *Adiabatic efficiency*, s (entropy) = constant

        $$\eta_s = \Delta E_s / W_{in} \; ; \text{where, } \Delta E_s \text{ is the exergy diff. at const. } s$$

    - *Volumetric efficiency*, ratio of measured (actual) mass flow to theoretical (as calculated from the displacement and speed)

        $$\eta_v = m_{actual} / m_{theoretical}$$

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- For an ideal gas, the ratio of the isothermal to adiabatic work is,

$$(W_T / W_s) = (\eta_T / \eta_s) = \phi \cdot ln(p_r) / \{p_r{}^\phi - 1\}$$

  - Where, $\phi = (\gamma - 1) / \gamma$ (= 0.4 for helium)

    γ is the ratio of specific heats (= 1.67 for helium)

    p_r is the discharge to suction pressure ratio

  - Note that $(W_T / W_s) < 1$ for $p_r > 1$; so, _isothermal compression is the optimum process_
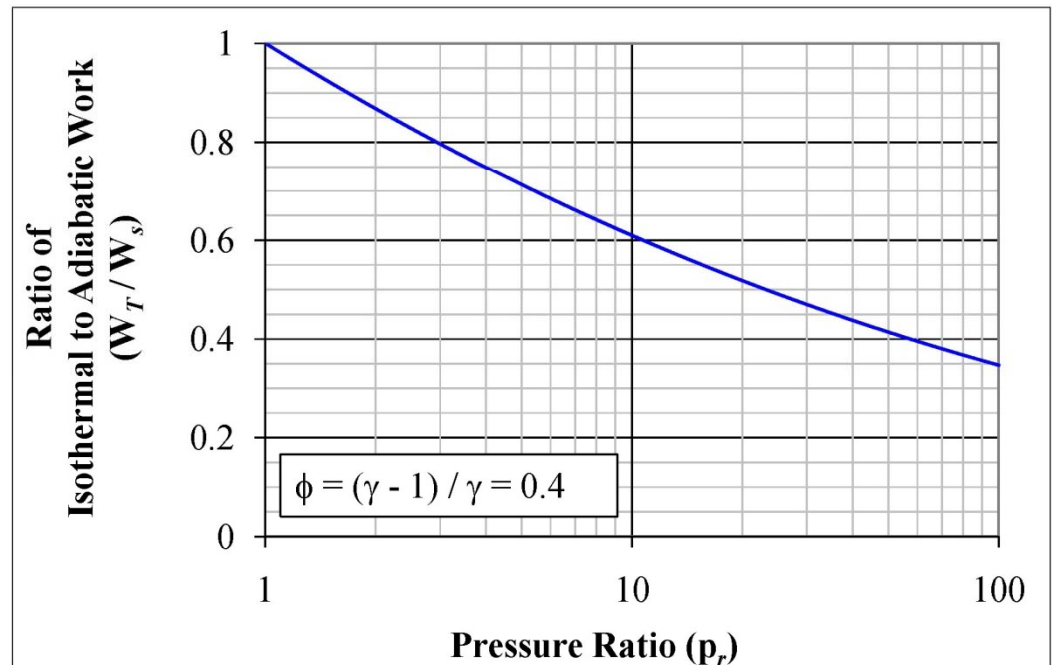
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- Several first and second stage Sullair compressors where tested in 1993 at the SSCL.   The table below summarizes the compressor information

	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 (Fixed)	2.2	2.6
Displacement @ 3550 RPM	2148 m³/hr (1264 CFM)	2905 m³/hr (1710 CFM)
Nominal Oil Charge	132 L (35 gal.)	227 L (60 gal.)

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- Volumetric efficiencies (η_v) – note the strong dependence on p_r, and weaker dependence on discharge pressure

Note: Optimum in the neighborhood of p_r − 3 to 4
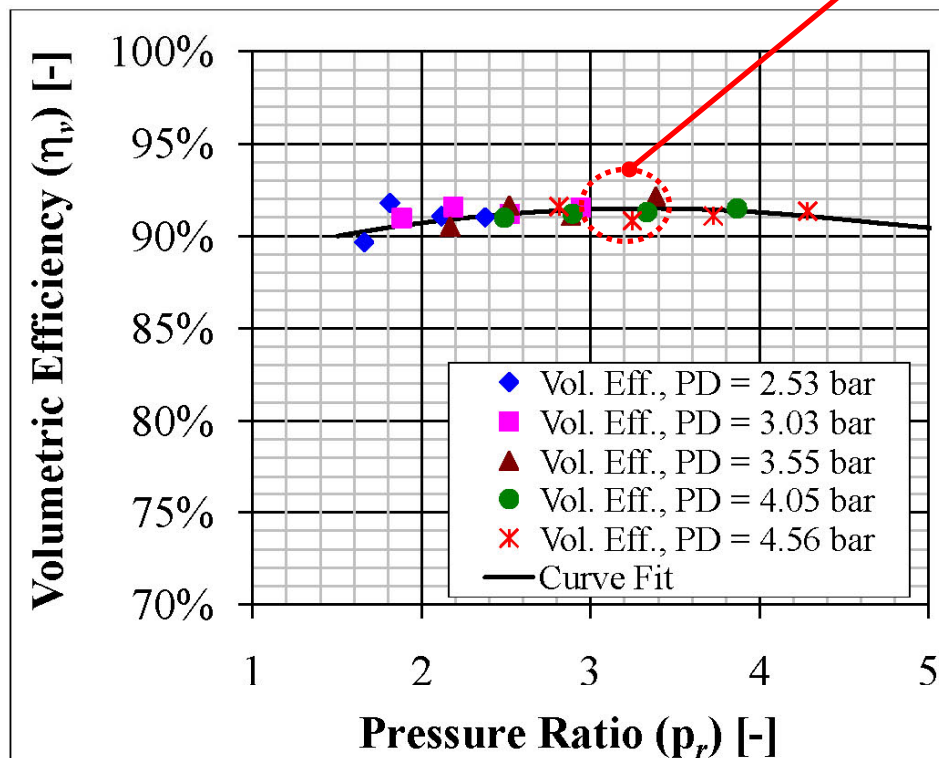
1st Stage   2nd Stage
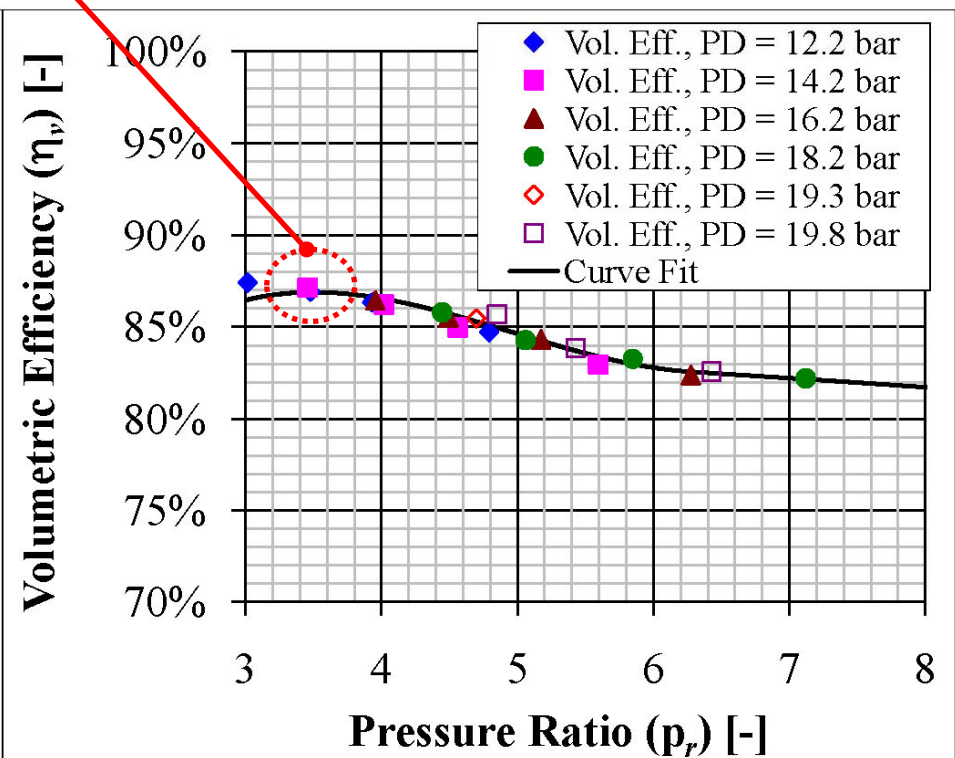
JLab's 12 GeV Compressor System

- Isothermal efficiencies (η_T) – note the strong dependence on p_r, and weaker dependence on discharge pressure

Note: Optimum in the neighborhood of p_r − 3 to 4
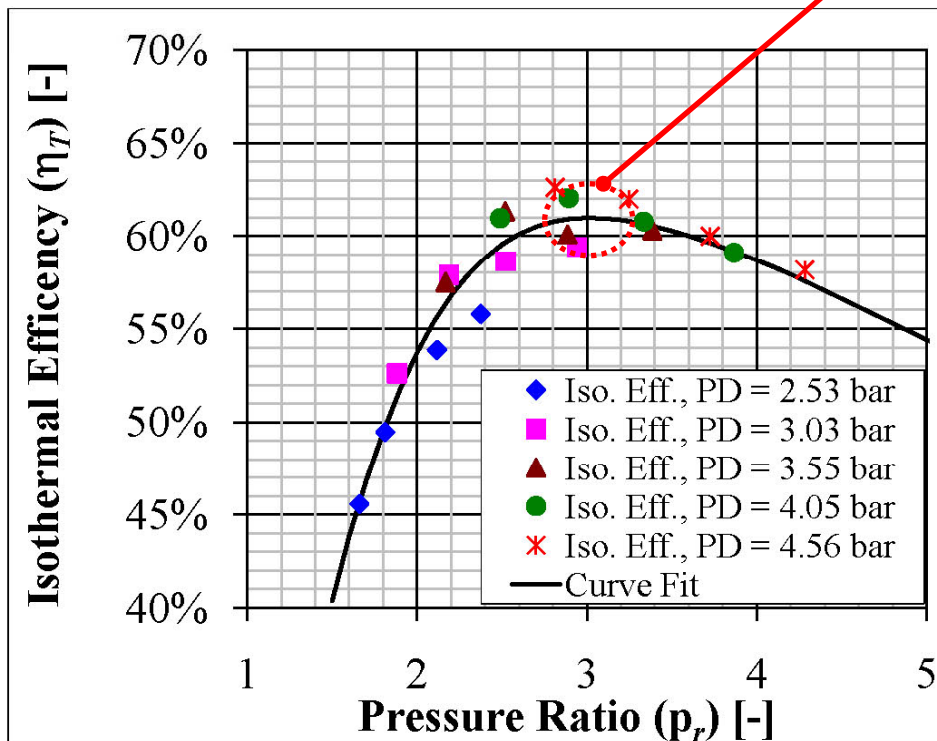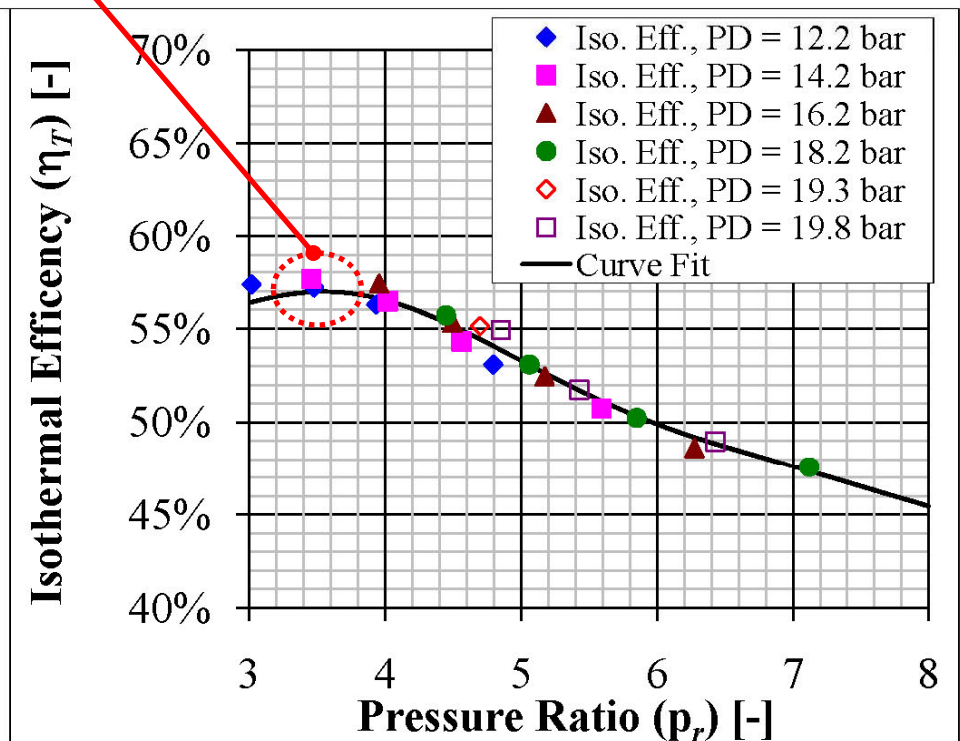
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- CHL-1 was designed with,
  - 2nd stage, $p_r = 7.34$; $\eta_v \approx 82\%$, $\eta_T \approx 47\%$
- CHL-2 id designed with,
  - 2nd stage, $p_r = 3.2$; $\eta_v \approx 87\%$, $\eta_T \approx 57\%$
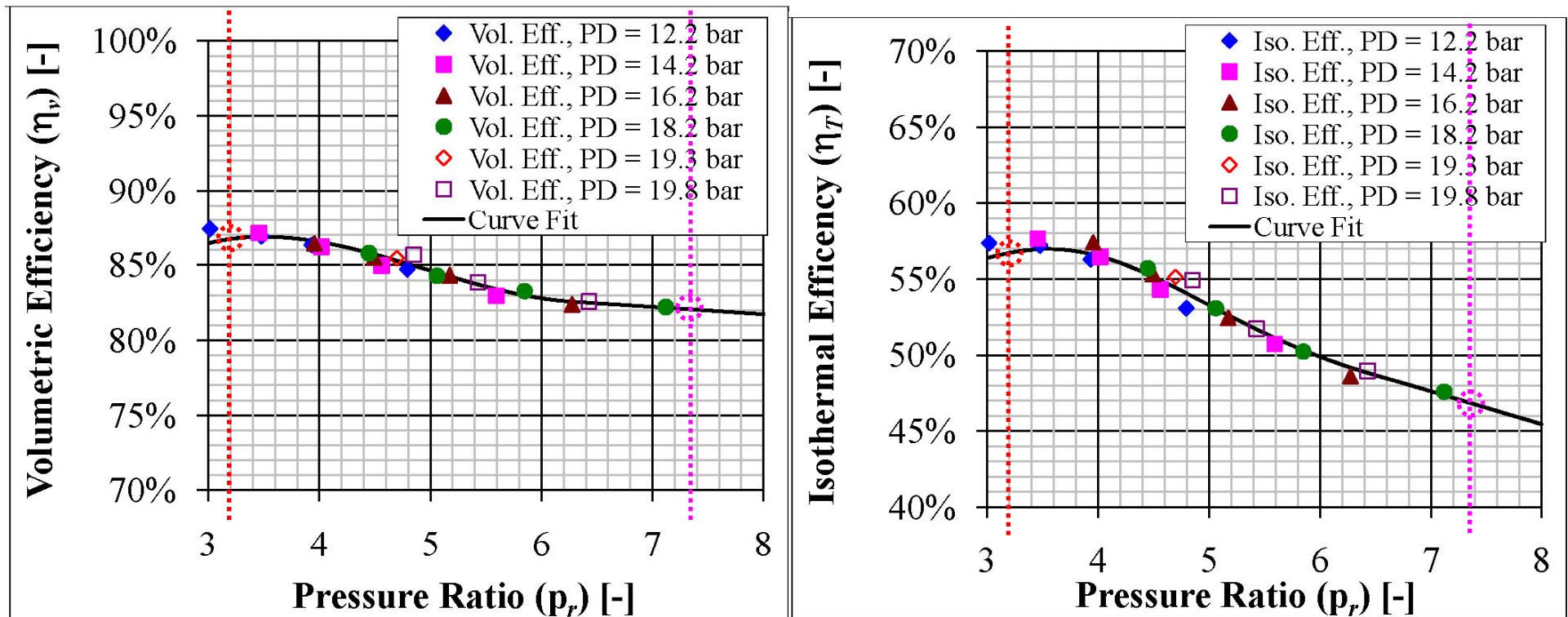
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- It's just a compressor…*right*?

- It's not even cryogenic…

- OK, we need one but…what does it really do?

Cold Box

Compressors

Increase Exergy

Load

Cold Box

Use Exergy

Compressors

Load
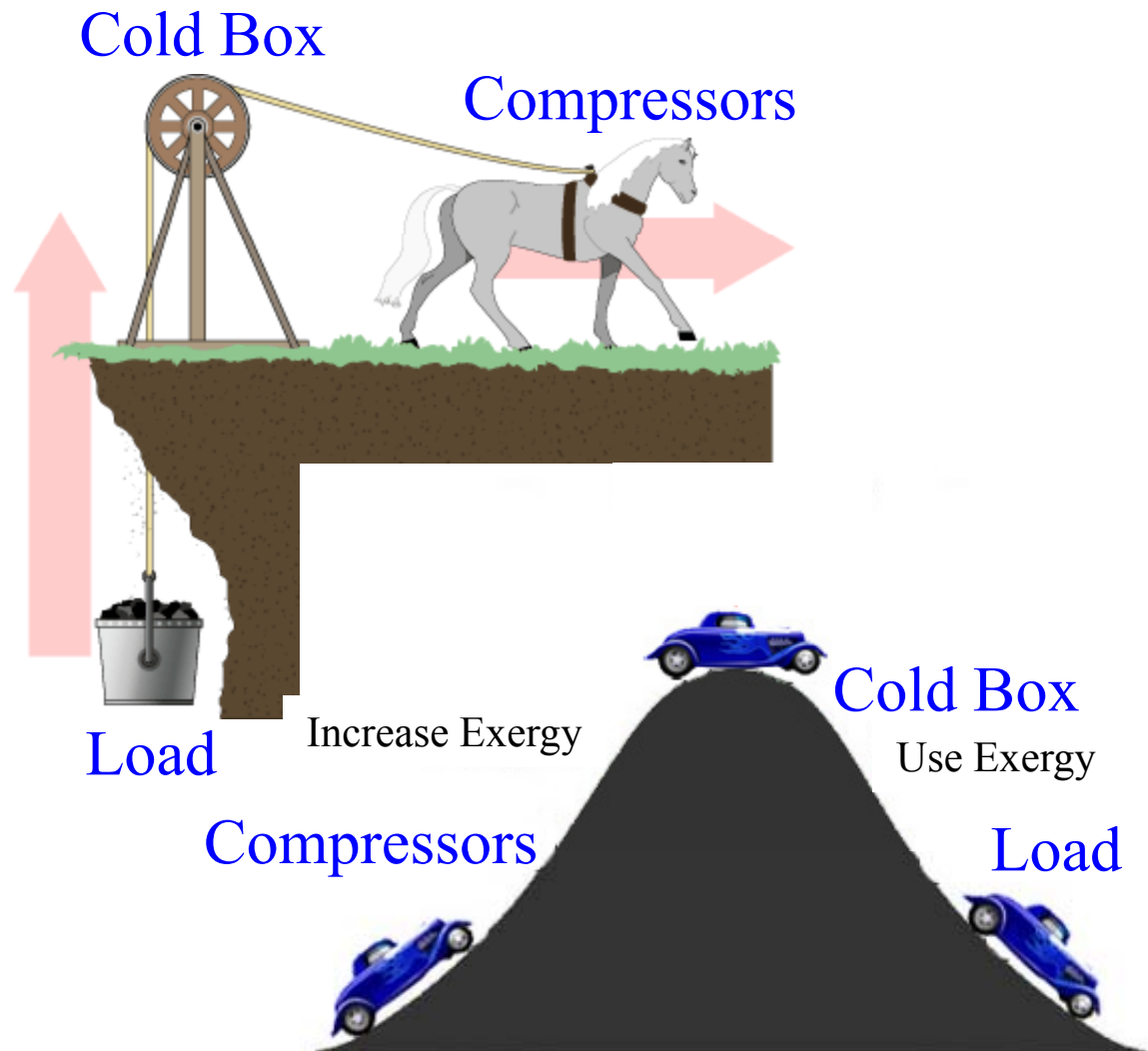
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- Example: CHL 2nd stage compressor (Howden WRVi321/165)
- A lot of power goes in…*but where does it go?*

1st Law:
$$W_i = m \cdot C_p \cdot (T_D - T_S)$$
1.4 MW = 0 ???

W_i = 1.4 MW

Discharge:
p_D = 18.5 atm
T_D = 310 K

Suction:
p_D = 2.5 atm
T_D = 310 K

$T_D = T_S$

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- The input power increases the "potential energy" of the working fluid, and is dissipated as heat to the ambient environment

- How does this work (with all the input power is dissipated as heat)?

$q = 1.4\ \text{MW}$ $W_i = 1.4\ \text{MW}$

1st Law:

$W_i = m \cdot C_p \cdot (T_D - T_S) + q$

$1.4\ \text{MW} = 0 + 1.4\ \text{MW}$

Discharge:
$p_D = 18.5\ \text{atm}$
$T_D = 310\ \text{K}$

Suction:
$p_S = 2.5\ \text{atm}$
$T_S = 310\ \text{K}$

$T_D = T_S \quad \blacktriangleright \quad W_i = q$

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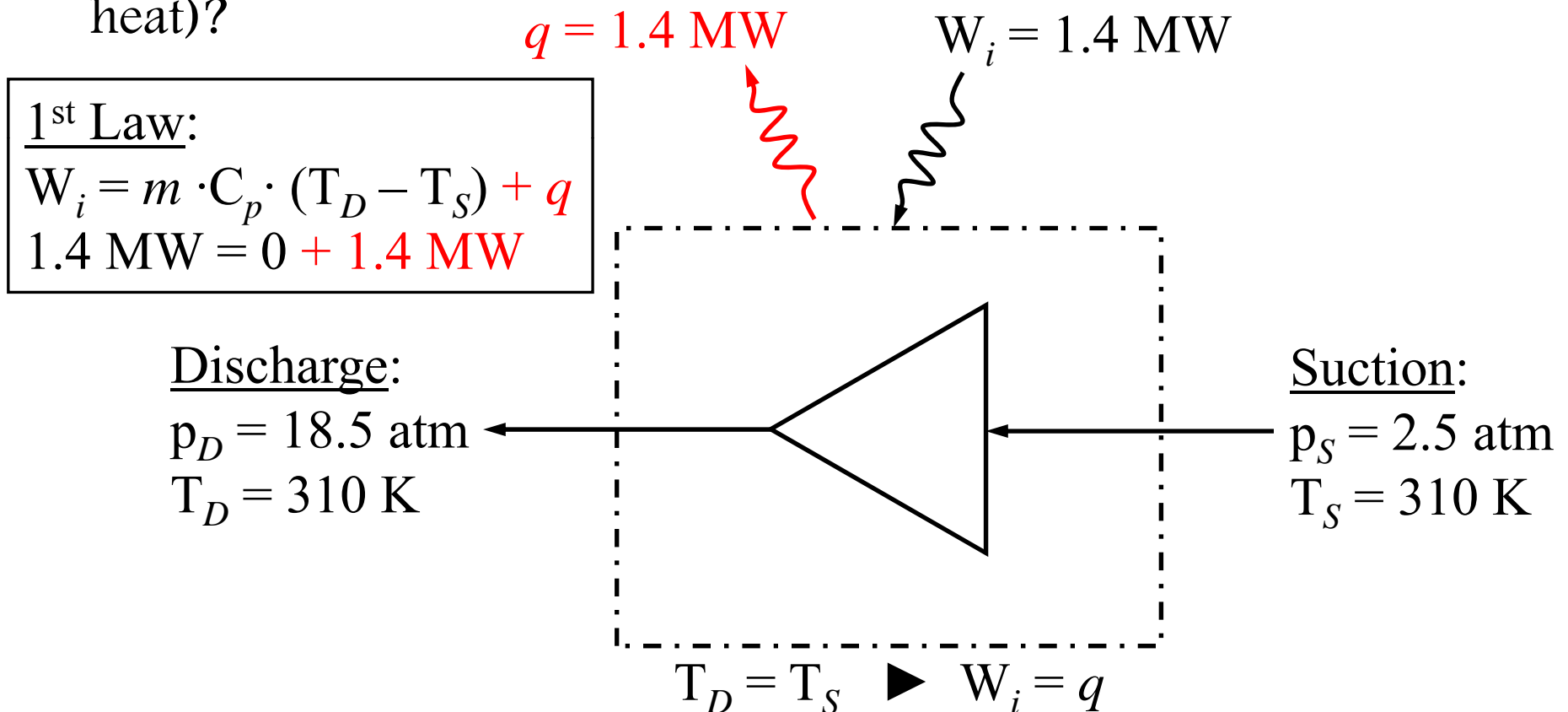
JLab's 12 GeV Compressor System

- 2nd Law of thermodynamics states,

  $dS \geq \delta q / T$ (in differential form, noting that 'δ' indicates path dependence & positive δq denotes heat "in")

- So, if T = constant,

  $\Delta S \geq q / T$

- That is, if heat is removed (–) then there is a entropy decrease (–); and visa-versa

- For an ideal gas (which a good approximation for helium at ambient temperatures and moderate pressures),

  $\Delta s = \Delta S / m = C_p \cdot \{ \textbf{\textit{ln}}(T_D / T_S) - \phi \cdot \textbf{\textit{ln}}(p_D / p_S) \}$

- Since, $T = T_D = T_S$

  $\Delta s = \Delta S / m = - C_p \cdot \phi \cdot \textbf{\textit{ln}}(p_D / p_S)$

- This equates the increase in pressure to the decrease in entropy

JLab's 12 GeV Compressor System

- So,

$$-q \geq -T \cdot \Delta S = m \cdot \phi \cdot C_p \cdot T \cdot \boldsymbol{ln}(p_D / p_S)$$

- Recalling the definition of exergy, and assuming an ideal gas,

$$\Delta \varepsilon = \Delta h - T_0 \cdot \Delta s = C_p \cdot (T_D - T_S) - T_0 \cdot \phi \cdot C_p \cdot \boldsymbol{ln}(p_D / p_S)$$

$$\Delta \varepsilon = -T_0 \cdot \Delta s = -T_0 \cdot \phi \cdot C_p \cdot \boldsymbol{ln}(p_D / p_S)$$

- If, $T_S = T_0,$

$$\Delta \varepsilon = -T_S \cdot \Delta s = -T_S \cdot \phi \cdot C_p \cdot \boldsymbol{ln}(p_D / p_S)$$

- And, using the 1st law and the definition of isothermal efficiency,

$$-q = W_i = \Delta E / \eta_T = m \cdot \Delta \varepsilon / \eta_T = m \cdot \phi \cdot C_p \cdot T \cdot \boldsymbol{ln}(p_D / p_S) / \eta_T$$

Heat dissipated
to environment

"Potential energy"
increase of fluid

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- Where did *all that input power* go, because of the 2nd Law of Thermodynamics...

CHL Evaporative Cooling Towers

JLab's 12 GeV Compressor System

- The following shows the input power losses per **1 kW** of cooling at 2.1 K (0.039 atm) for CHL-1 (existing) and (estimated) for the new CHL-2

- <u>Over half of the input power is lost in the compressor system!</u>

CHL-1
1000 kW Electrical Input for
1 kW of Cooling at 2.1 K

2.1-K Load, (142.0) 14%

2.1-K CBX & Distr., (92.8) 9%

4.5-K CBX, (165.2) 17%

Compressors, (600.0) 60%

CHL-2
710 kW Electrical Input for
1 kW of Cooling at 2.1 K

2.1-K Load, (142.0) 20%

2.1-K CBX & Distr., (92.7) 13%

4.5-K CBX, (107.4) 15%

Compressors, (367.7) 52%

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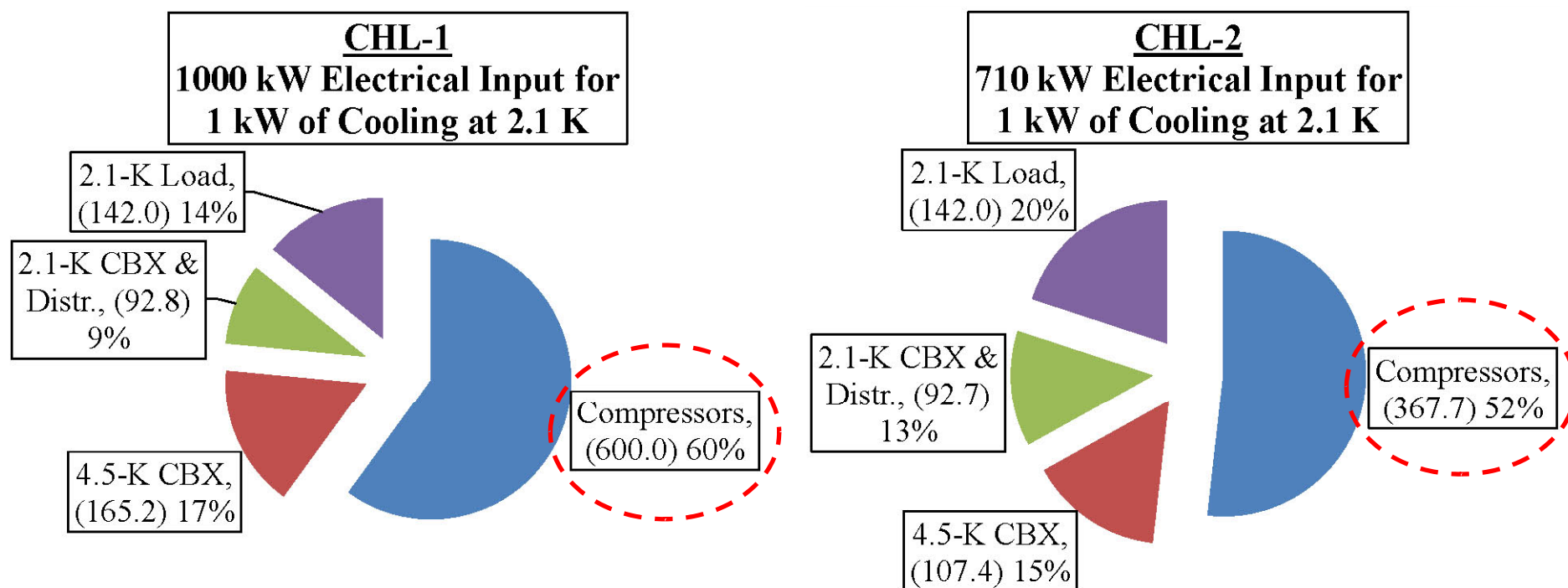
JLab's 12 GeV Compressor System

- As an important aside,
  - Assuming a thermal power plant to electrical conversion + distribution efficiency of 33% [†],
  - Every **1 kW** of cooling at 2.1 K requires,
    - ~17 metric tons/day of lignite coal [‡] using "a CHL-1"
    - ~12 metric tons/day of lignite coal [‡] using "a CHL-2"

Both providers and users of the cryogenic load need to be good stewards!

[†] *Assuming the coal power plant 'heat rate' (amount of fuel heat content in Btu per amount of electric energy produced in kWh) is 9300 Btu/kWh HHV (higher heating value; i.e., water vapor in combustion products) and the electric transmission loss is ~8%*

[‡] *Average in the U.S.; Lignite coal = 15 MJ/kg*

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- So, hopefully we now have an appreciation of the,
  - The role that the compressor system plays in a helium refrigerator – that it is the "heart" of the system, providing the rest of the system the "potential energy" (i.e., exergy) required for the process, and,
  - The relative portion of power lost in the compressor system due to its inefficiencies, as compared to the rest of the helium refrigeration system
  - The impact of the total energy demand required by the 2-K process

- What are the inefficiencies in the compressor system, and what can be done to reduce them…?

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- Obviously, effort can be given to improving the rotor design…but we will leave that to the manufacturers

- A compressor is more than just the compressor proper; i.e., it is a skid – a package, comprising a number of (major) components
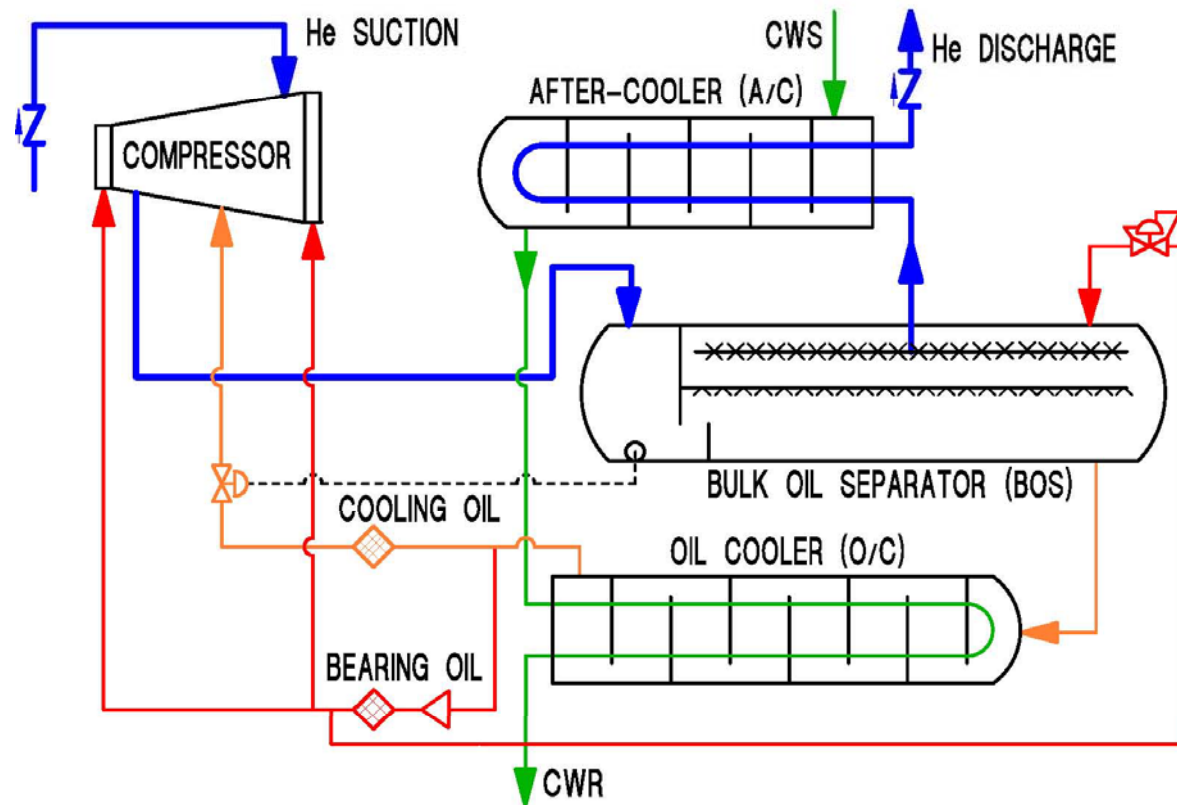
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- Often, a good starting point to answer such a question is to carefully examine existing designs
- So, based upon the experience from
  - (4) different compressor and skid manufacturers that support
  - (5) cryogenic plants (at JLab), as well as, from many other facilities,
  - A number of observations were made over the years of their operation
  - Each of these were addressed in the compressor system designed by the JLab cryo group for the James Webb telescope testing to be conducted at the NASA-Johnson Space Center
  - *JLab's 12 GeV compressor system directly benefits from this design effort funded by NASA!*

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- The following are (10) observations and how they were addressed for the 12 GeV compressor skids (to be used for the new cold box)
  - <u>Observation #1</u>: A proper alignment can be difficult to achieve due to,
    - An uneven or sloped compressor and/or motor mounting plate and due to the way in which the compressor is attached to the rest of the skid
    - Misalignment due to thermal expansion (i.e., mounting compressor and motor on bulk oil separator)
  - Of course, the life of the bearings is significantly affected by the quality and preciseness of the alignment

  - *<u>Response to #1</u>:*
    - *A universal adjustable steel chock is used under each foot for the compressor and motor mounting thereby eliminating parallel and/or angular soft-foot*
    - *Compressor and motor are physically separate from the BOS to reduce misalignment due to thermal expansion*

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- Observation #2: BOS's typically used for these packages have a large diameter and flanges and utilize fragile glass-fiber coalescing filters to remove oil from helium prior to entering after-cooler (presumably for improved heat transfer)

- However, even with the coalescing filters, the after-cooler heat transfer surface is coated with oil

BOS

BOS

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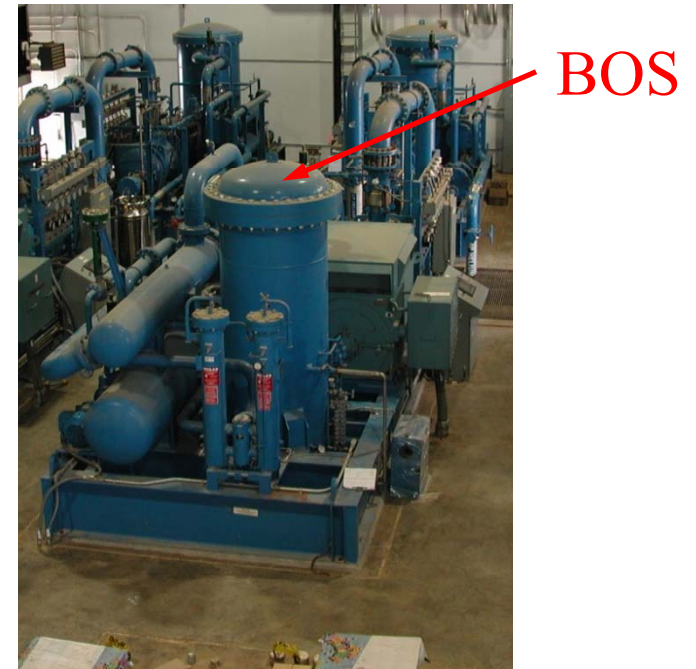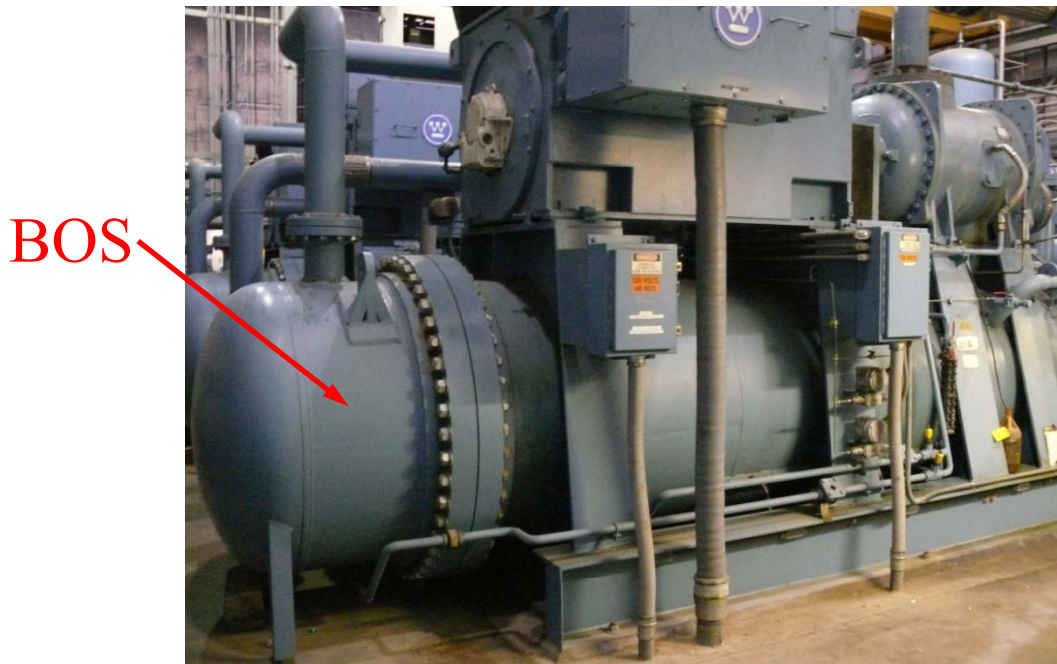
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- *Response to #2: BOS does not use glass-fiber filters. Referring to the figure below, the first stage separation is cyclonic as the helium enters the BOS*

- *Then the flow is distributed from horizontal supply header into the horizontal vessel at a slight downward angle, then gathered in the overhead return header*

Helium In (from Compressor Discharge)

Helium Out (to A/C)

Oil Weir

Return Headers

A

B

30

B

SECTION "A-A"

Oil Flow

Gravity Separation Section

Oil Out (to O/C)

SECTION "B-B"

Cyclonic Separation Section

Supply Headers
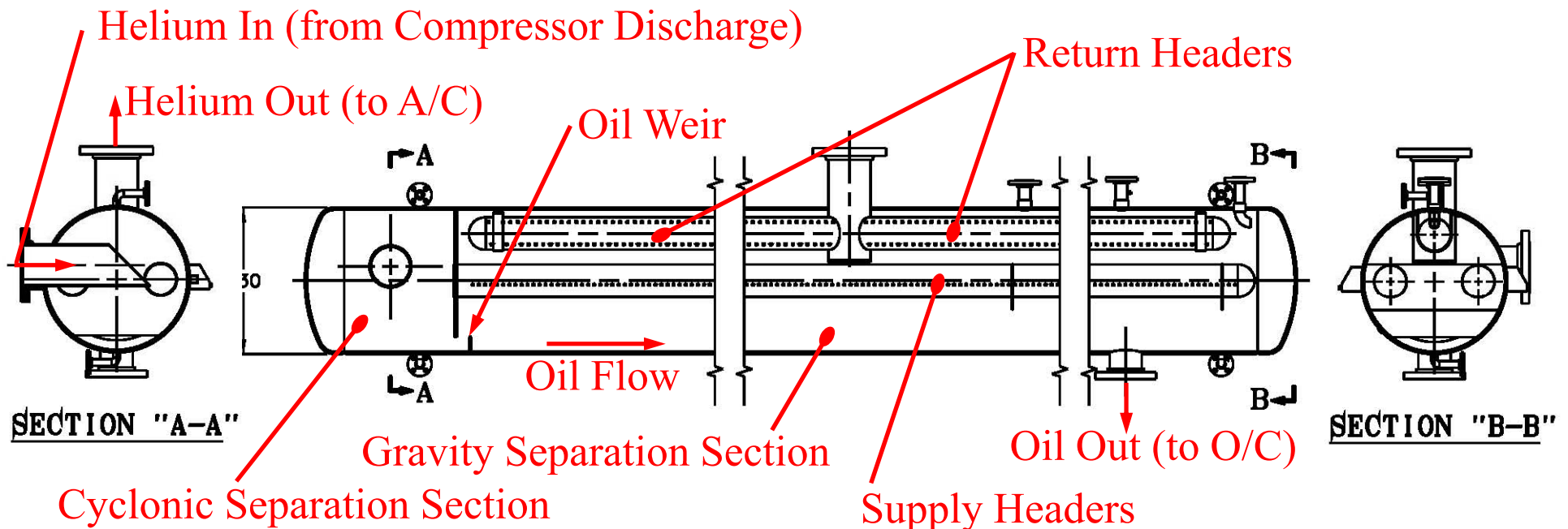
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- *Response to #2 (continued): Using this gravity separation, aided by the momentum change (to increase inertial impaction), the cross-section along the axis of the BOS is used rather than the vessel diametrical cross-section*

Helium In (from Compressor Discharge)

Helium Out (to A/C)

Oil Flow

Return Headers

Oil Weir

Supply Headers

Cyclonic Separation Section

Oil Out (to O/C)

Gravity Separation Section
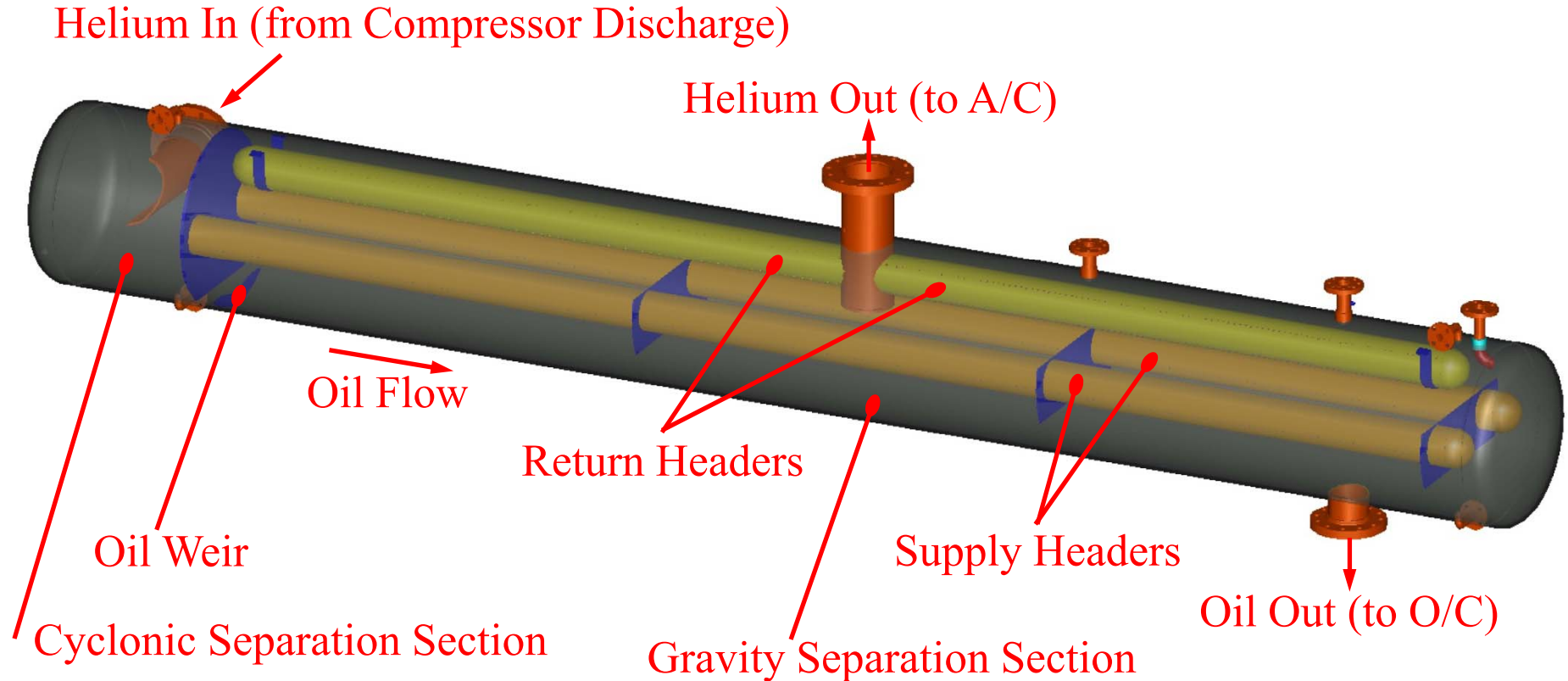
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- *Response to #2 (continued): Consequently, referring to the table below, the BOS diameter is quite small and does not use large flanges as compared to typical BOS's*

Bulk Oil Separator (BOS) Design Parameters (Estimated)

		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

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- BOS gravity separation section design goals
  - Minimize pressure drop, while
  - Keeping flow distribution as even as possible, and
  - Selecting an practical hole size and spacing
  - Keep total orifice area ≥ header cross-sectional flow area
  - Keep in mind that inlet static pressure difference in supply header will directly affect oil height difference between cyclonic and gravity sections (which are separated by a weir)
  - Orient header orifices to facilitate droplet inertial impaction by maximizing directional change in momentum

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- CHL-2 HP skid BOS gravity separation section, supply & return header flow distribution (design)
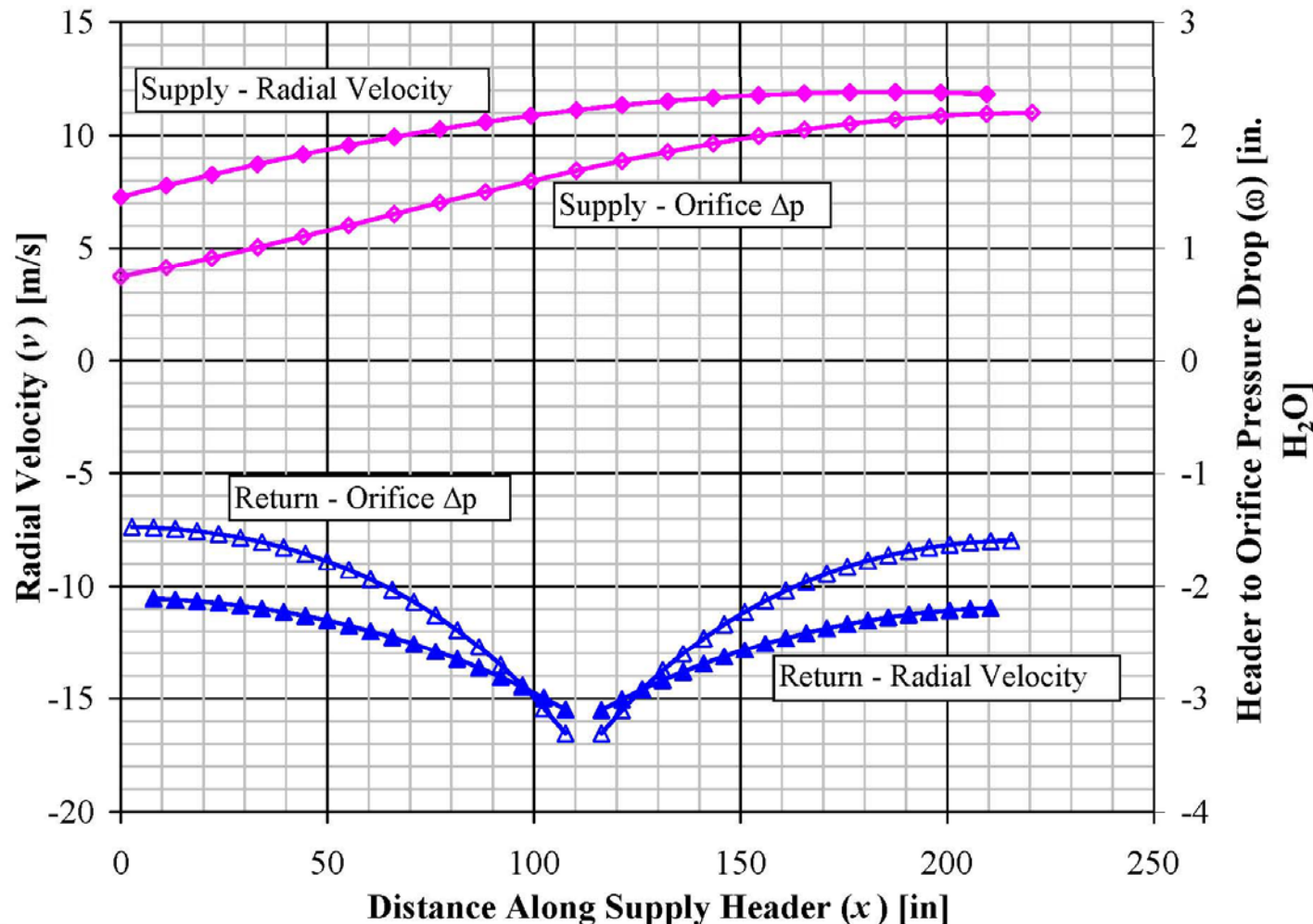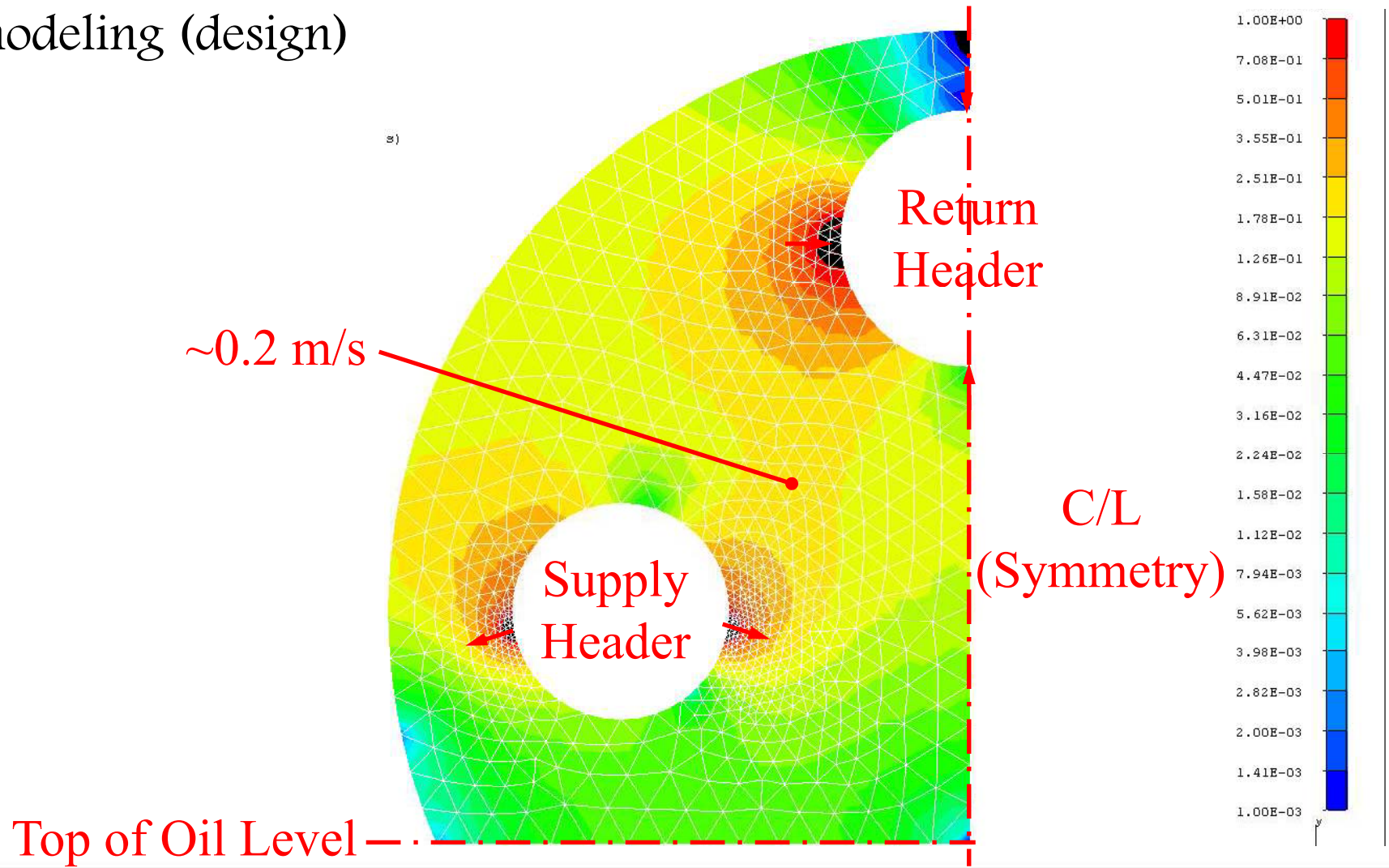
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- CHL~2 HP skid BOS gravity separation section header design – flow modeling, 1D with tangential mass subtraction (or addition)

  - Note that for the supply header, the effect of the mass leaving tangentially is known as 'wall suction'; and its net effect is to increase wall fraction

  - The converse is true for the return header (though it is known as "wall blowing")

<u>Non-Dimensional Variables:</u>

ξ	$= x / L$	*Position along header in the direction of (+) axial flow velocity*
Π	$= 2 \cdot (p - p_\infty) / (\rho \cdot u_0^2)$	*Orifice pressure drop*
u^*	$= u / u_0$	*Axial flow velocity*
ζ	$= \zeta(u^*/v^*)$	*Flow resistance coefficient*
v^*	$= v / u_0 = (\Pi / \zeta)^{0.5}$	*Radial flow velocity*
$(du^*/d\xi)$	$= -\lambda \cdot \alpha \cdot v^*$	*Gradient of axial flow velocity along header length*
Re	$= Re_0 \cdot u^*$	*Local Reynold's number*
f	$= f(Re, \varepsilon)$	*Local friction factor*
$(d\Pi/d\xi)$	$= -u^* \cdot \{4 \cdot (du^*/d\xi) + f \cdot \lambda \cdot u^*\}$	*Gradient of orifice presssure drop along header length*

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- CHL~2 HP skid BOS gravity separation section, inviscid flow modeling (design)

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- Estimation of oil carry-over from BOS; e.g., HP skid
  - Estimate Sauter droplet diameter

    D_{32} = 94 to 233 μm [<u>Ref</u>. Perry's, 7th Ed., equs. 14-199,198,200]
  - Estimate max. droplet size given velocity field (which is used as the terminal settling velocity)

    $D_{p,max}$ = 78 μm [<u>Ref</u>. Perry's, 7th Ed., equ. 6-228]
  - Assuming a droplet size distribution, the mass fraction of oil not separated can be estimated

    $\Delta = (D_{p,max}/D_{32}) = 0.824$ (using smallest D_{32})

    f_o (mass frac. of oil not sep.) $= 1 - exp(-c_0 \cdot \Delta^3) = 0.533$
    - with, $c_0 = \Gamma(5/3)^{-3} = 1.359$
    - <u>Ref</u>. C. Dumouchel, "The Maximum Entropy Formalism and the Prediction of Liquid Spray Drop-Size Distribution", Entropy, 11, 2009, p.713-747, equ. 27

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- – _Response to #2 (continued): Additionally, an intentional provision is made in the after-cooler to drain coalesced oil due to momentum changes in the boundary layer and increased coalescence as the oil surface tension and its viscosity increases as it is cooled_

- – _Since oil is present, with or without the coalescing filters in the BOS, the after-cooler is sized assuming 0.2% of oil (by weight of helium)_

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- *Response to #2 (continued): An external, but not integrally attached to the skid, vertical gas-fiber coalescer filter is located downstream of the after-cooler which is capable of being located in a convenient and unobtrusive location in order to maintain the skid compressor oil balance locally*

- *The coalescer efficiency is improved for the same size since the helium-oil mixture is at a lower temperature (and a higher density)*

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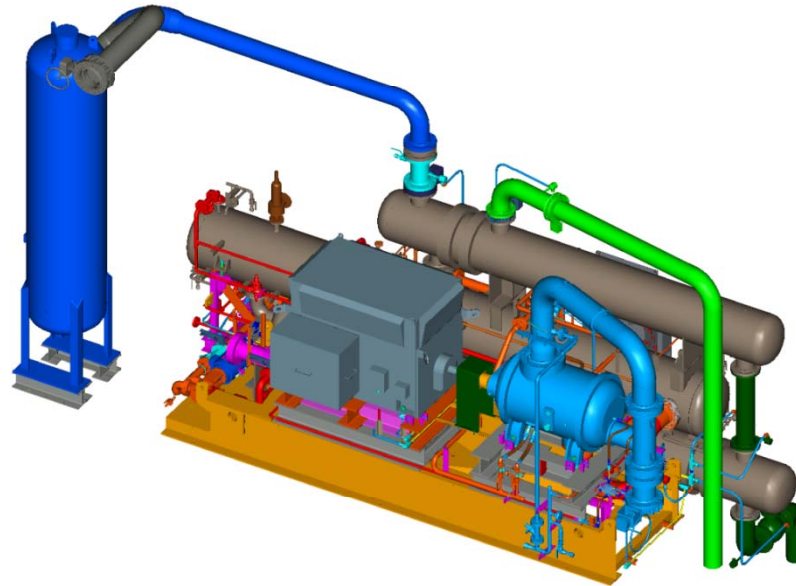
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- Observation #3: Some compressors require the entire oil flow, both for cooling and for the bearing lubrication and sealing, to be pumped

- *Response to #3: The selected Howden compressors require only the bearing oil to be pumped*

- *This (bearing oil) amounts to approximately 20% of the total required oil flow*

- Observation #4: A significant amount of oil is bypassed back to the oil separator due to over-sizing of the oil pump used to circulate the entire oil flow (used for both cooling, bearings and slide valve operation)

- *Response to #4: A variable frequency drive (VFD) is used on the pump, whose speed is adjusted to maintain a fixed differential pressure between oil injection pressure and helium discharge pressure*

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- <u>Observation #5</u>: Oil pump is often located upstream of the oil cooler to reduce the required input power by pumping the fluid when it is less viscous

- However, this results in handling the oil when it is still at its peak temperature

- Consequently, it is known that these pumps will develop more frequent shaft seal leaks

- *<u>Response to #5</u>: The oil pump is located downstream of oil cooler and is significantly smaller, handling only the oil required for the bearings (i.e.,~20% of the total oil flow)*

- *Also, the additional input power required for pumping the oil when it is cooler amounts to only a modest amount (perhaps 5% more than if it were pumped upstream of the oil cooler)*

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- Observation #6: Location of the hydraulic load valves for the compressor slide valve (for compressors of this configuration) is not usually easily assessable; this valve is a maintenance item

- *Response to #6: Hydraulic load valves and connecting hoses are located so they are easily accessible*

- Observation #7: Regulation of the oil injection temperature is done manually

- *Response to #7: The injection temperature of the (non-pumped) cooling oil is regulated using a control valve*

- *By properly regulating this temperature, the input power can be reduced by a few percent*

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- Observation #8: Oil coolers and after-coolers can have significant flow bypassing in the shell
- Also, the oil cooler can be undersized (which can result in a compressor shutting down on a hot day)

- *Response to #8: Both oil and after coolers are a TEMA AEU design; i.e., channel and removable cover, one pass shell, U-tube bundle*
- *The oil cooler is sized for approximately 90% of the total compressor power*

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- Observation #9: Inadequate oil filtration to the rotors
- This can significantly affect the life of the bearings and rotor tolerances
- The filtration should be at least 30 microns

- *Response to #9: Both cooling oil and bearing oil is filtered to 8 microns at 98% efficiency, with no significant pressure drop penalty ($\Delta p_D/p_D < 2\%$ for MP stage)*
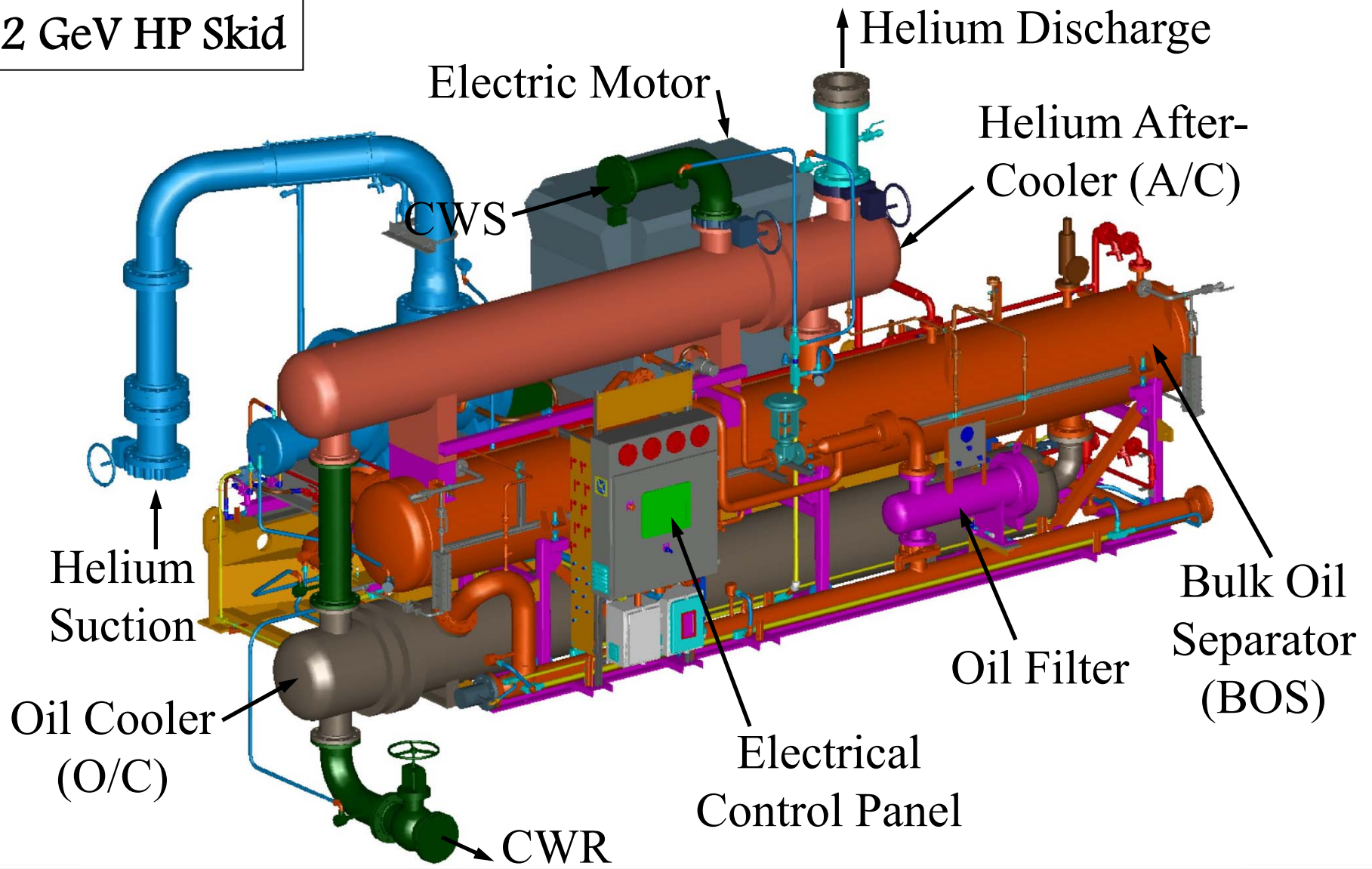
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- Observation #10: There can be an apparent loss and gain of oil in the BOS when starting and stopping the compressor

- *Response to #10: The oil cooler is at the lowest elevation and always kept full*

- *Skid components are arranged to allow the oil to completely drain by gravity to the BOS, which is immediately above the oil cooler*

- *Oil is not allowed to accumulate in other parts when the compressor is shut-down, so that there are no oil slugs upon start-up*
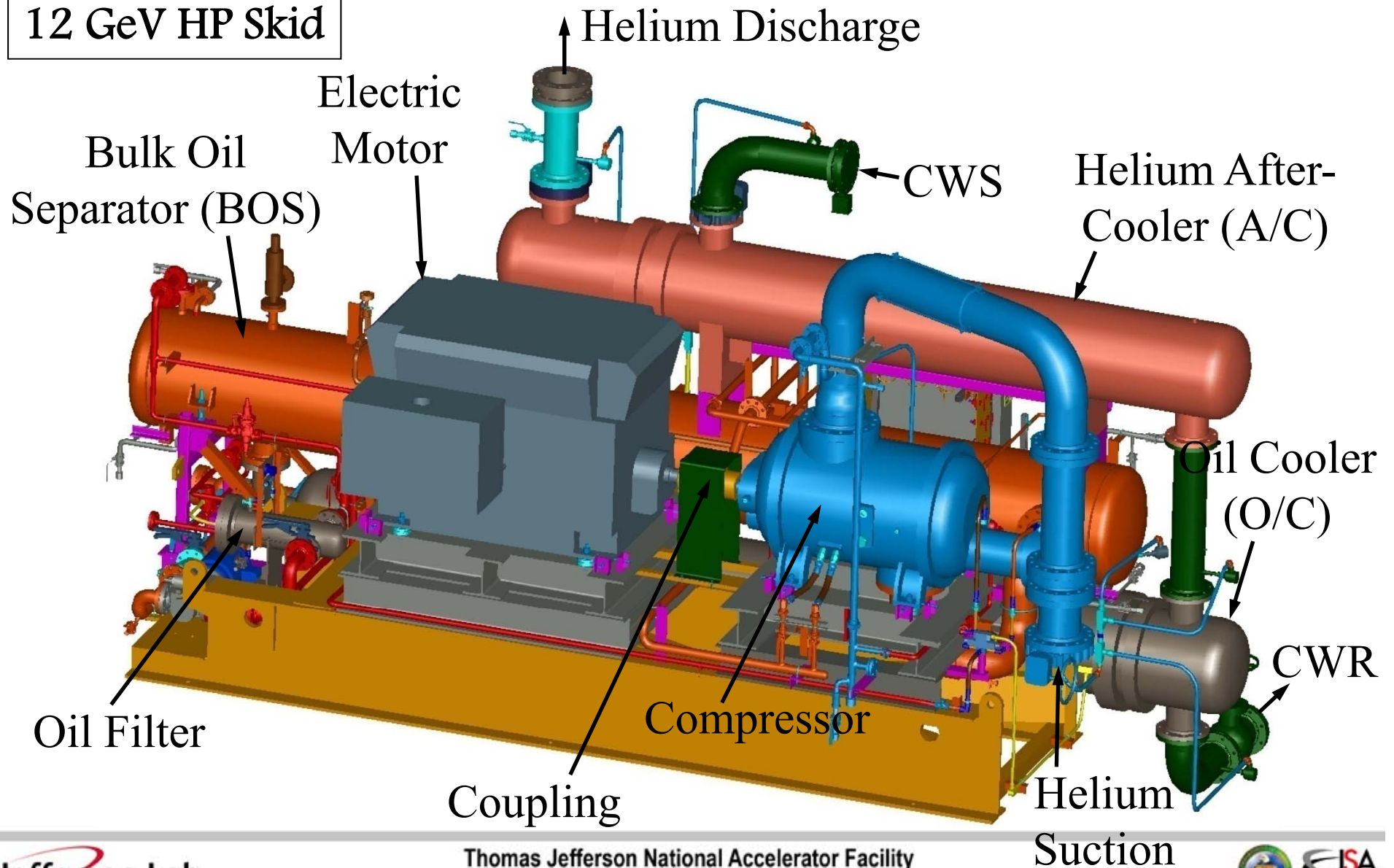
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12 GeV HP Skid

Helium Discharge

Electric Motor

Helium After-Cooler (A/C)

CWS

Helium Suction

Oil Cooler (O/C)

CWR

Electrical Control Panel

Oil Filter

Bulk Oil Separator (BOS)

JLab's 12 GeV Compressor System

12 GeV HP Skid

Helium Discharge

Electric Motor

Bulk Oil Separator (BOS)

CWS

Helium After-Cooler (A/C)

Oil Cooler (O/C)

CWR

Oil Filter
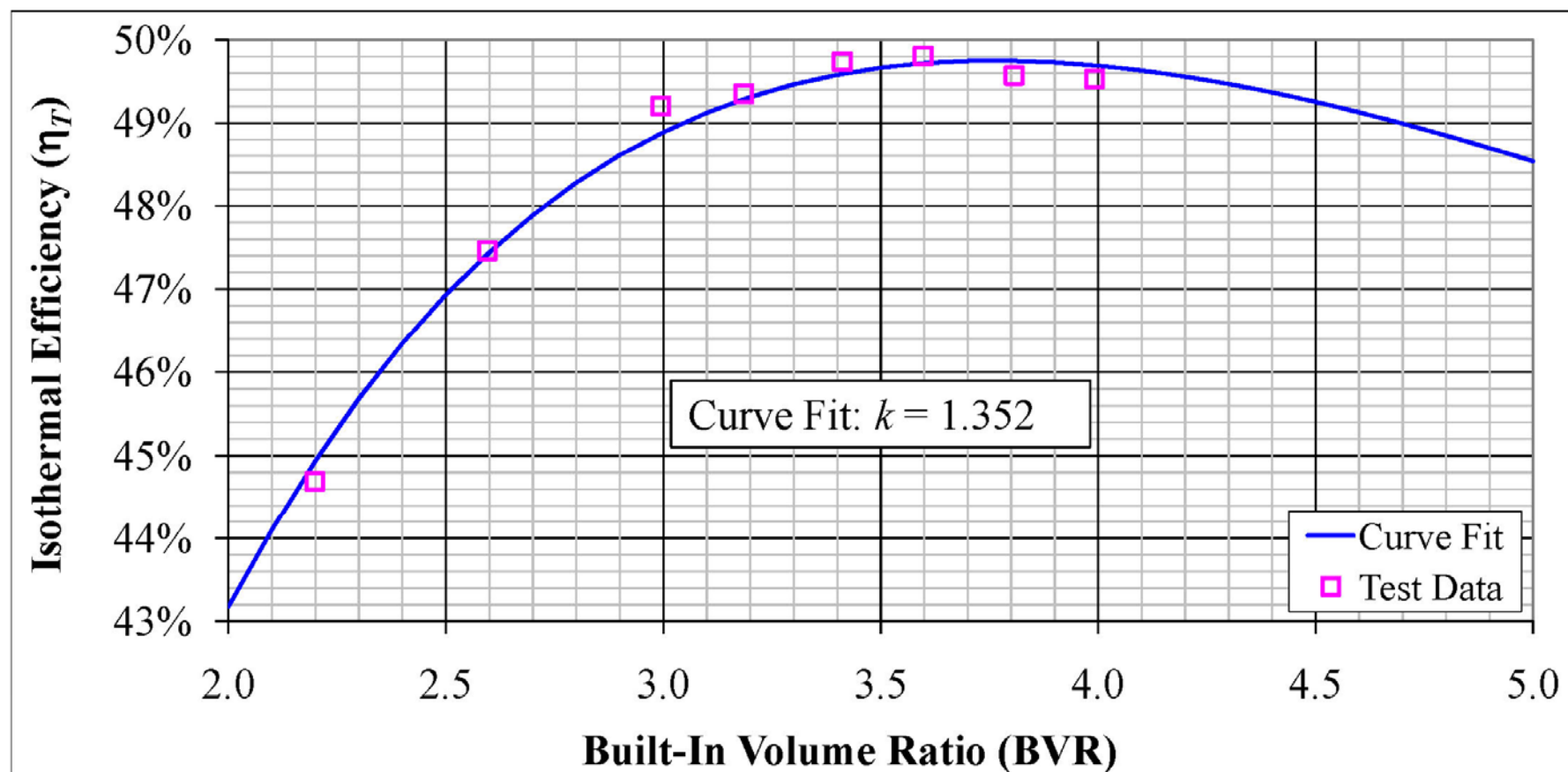
Compressor

Coupling

Helium Suction

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- JLab Compressor R&D ~ Focus areas
  - Power reduction by,
    - (~1 to 3%) Improving isothermal efficiency
      - Injection of fine spray of oil into suction line, promoting thermal equilibrium between oil and gas
      - <u>Ref</u>. Hartford screw compressor testing, CTI cir. 1983, Model 1215 (127 mm rotor, 1.50 L/D ratio)

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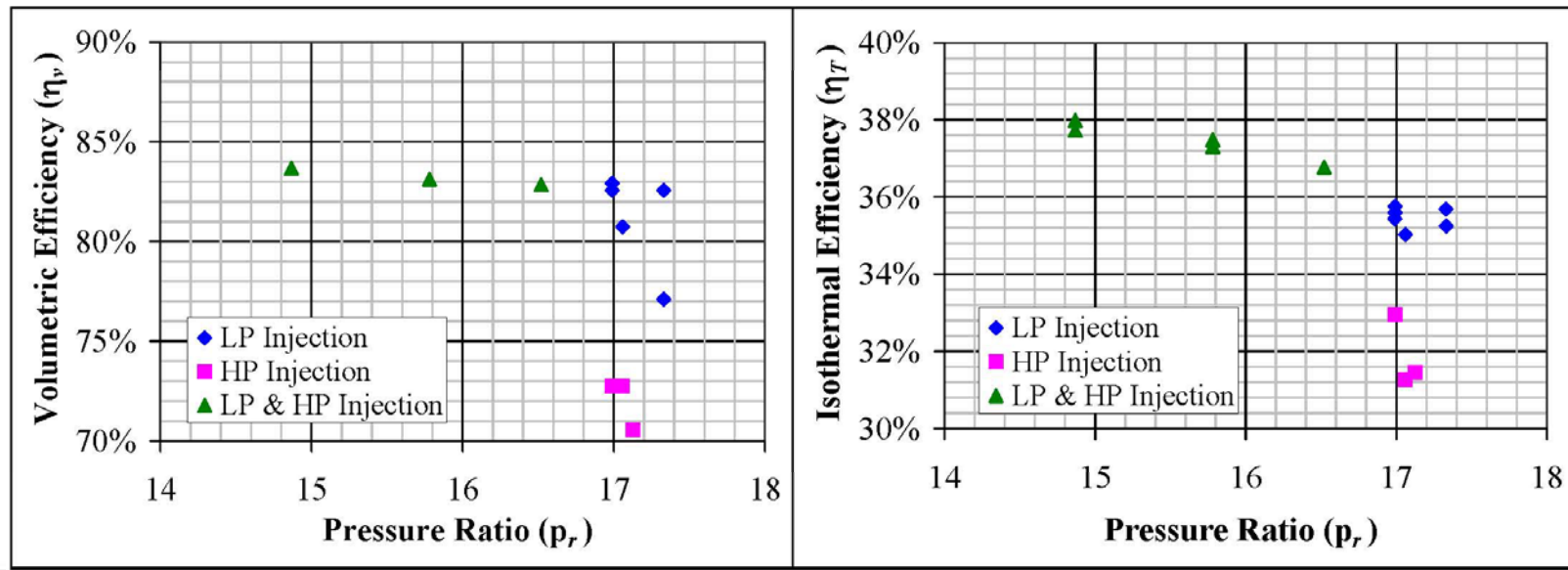
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- Testing on CHL-1 2nd stage (C5) WRVi321/165, Jan 2006
  - For perfect oil-helium mixing during polytropic compression, $k^o \approx 1.05$

Curve Fit: $k = 1.352$

Isothermal Efficiency (η_T) vs Built-In Volume Ratio (BVR)

Curve Fit — Test Data

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- JLab Compressor R&D ~ Focus areas
  - Power reduction by,
    - (~2 to 3%) Establishing optimum flow regulation of injected oil; i.e., increasing oil injection,
      - » Tends to improve the isothermal efficiency (i.e., the compression process becomes more isothermal), but
      - » Increases viscous friction (oil becomes cooler)
      - » So, there is an optimum oil injection; this usually corresponds to roughly an oil injection temperature of ~322 K (120 °F)
  - Note that using very high oil injection pressure to achieve high atomization (of oil) does not improve the efficiency, since the gains in making the compression more isothermal are outweighed by the additional power required to pressurize the oil
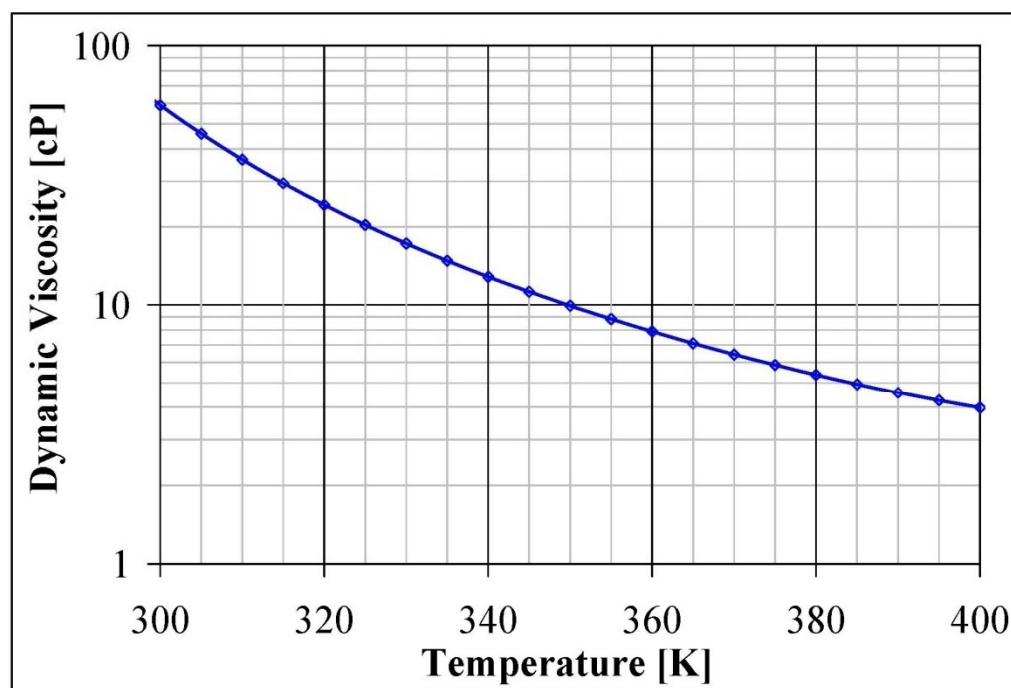
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- Equation for power dissipated by a journal bearing, can be used to estimate oil viscous-friction power loss

$$P = N^2 (\pi D)^3 L \mu / \delta$$

  – Where,
    - N – rotational speed
    - D – rotor diameter
    - L – rotor length
    - μ – oil dynamic viscosity
    - δ – clearance between rotors and housing

UCON LB-170X

Note: Dynamic viscosity is a Log scale

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- JLab Compressor R&D - Focus areas
  - Power reduction by,
    - (~1%) Reduction of wasted pressure drop in BOS
      - Promote increased inertial impaction in the separation of the oil from the helium without using coalescing filters (in the BOS and at elevated temperatures).
      - Use, BOS with combined centrifugal (cyclonic) and gravity separation sections; gravity separation of oil from helium accomplished in a compact and efficient manner by using supply and return headers in a small diameter vessel; or by using multiple 'trays'

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- Effect of pressure drop due to valves, BOS and A/C downstream of compressor discharge is to increase the isothermal power required

  - e.g., CHL~1, 2nd stage; $p_r = 7.2$, $\Delta p_D > 0.5$ atm, $(\Delta p_D/p_S) > 0.2$
  - so, ~1.5% additional isothermal power is required due to the pressure drop
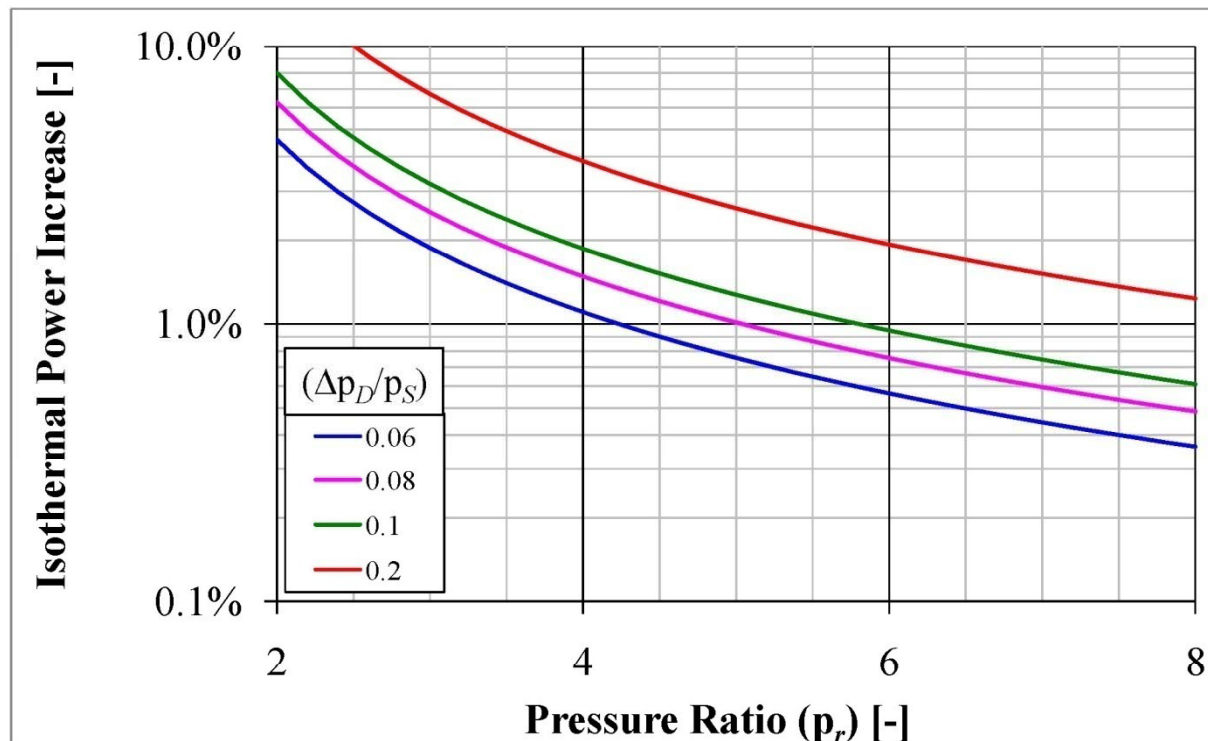
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- JLab Compressor R&D ~ Focus areas
  - Increase capacity by,
    - Improving volumetric capacity
      - Provide more phase interface surface area for at least the same contact time to promote the dissolution of helium from oil
      - This is done in the BOS

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- JLab Compressor R&D ~ Focus areas
  - Increase performance by,
    - Using coalescing elements at lower temperatures; rather than at the highest temperature (i.e., compressor discharge) in the BOS
    - Increase bulk oil removal efficiency by utilizing the after-cooler to assist in oil-helium separation
  - Increase environmental safety by,
    - Using a smaller BOS which allows a smaller oil charge in order to minimize the environmental impact due to oil leakage

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- <u>Summary</u>
  - The compressor system is often considered a component not worth the same level of consideration as compared to the cold box
  - Consequently, the helium compressor has not received the same attention in R&D as the cold box
  - However, *<u>it should!</u>*...since it,
    - Provides the cold box the "potential energy" (exergy) and,
    - *<u>At least half</u>* of the input power is lost in the compressor system
    - With increased awareness of energy usage and costs, *this has been an ongoing area of active research and development for the JLab cryogenics group*