Essential Physics and Engineering of Cryogenics for Accelerators

Simplified Concepts & Practical Viewpoints

By

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What is Cryogenics?

It is the production of temperature below 123K (-150 C)

Examples of Cryogenic Fluids

Cryogenic Fluid	Tsat, 1 atm [K]
Helium	4.22
Hydrogen	20.28
Neon	27.09
Nitrogen	77.31
Argon	87.28
Oxygen	90.19
Methane	111.69

History

Cryogenics was primarily used for

- Gas separation

Helium was first liquefied by Heike Kamerlingh Onnes on July 10th 2008, in Leiden (NL)

He Observed superconductivity in 1911

This lead to the application of Cryogenics to:

- Physics research
- Medical Applications (MRI Magnets)
- Instruments

The Other Applications are:

- Biological & Medical
- Space research
- Vacuum

Superconductivity

No resistance below a critical temperature

a) Low Temperature Super conductors (below 20K)
   Used for Magnets and RF cavities

a) High Temperature Super conductors (around 70K Level)
   Used for power leads

All these need Cryogenics

Particle Accelerators

Particle Accelerators use Magnets and RF cavities

At room temperature the iron core saturates at about 2T, where as the magnets built with super conductors can be designed for large magnetic fields like10T and more and are compact

Similarly the room temperature RF cavities are built for less than 500 HZ. Higher frequency designs require low temperature environment for operation

For a given energy, the accelerators designed with superconductors require:

- Lower capital cost
  - Since it requires fewer number of Magnets and/or RF cavities
  - Less length of the accelerator
- Lower Operating cost

There fore for large accelerators, Superconducting Structures at cryogenic temperatures is a proven cost effective reality

All large particle accelerators need Cryogenics

Tentative Schedule

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- Discussions ??

1. Introduction

- **Helium refrigeration and liquefaction systems are an extension of the traditional household refrigeration systems**
- **Let's begin with the question,**

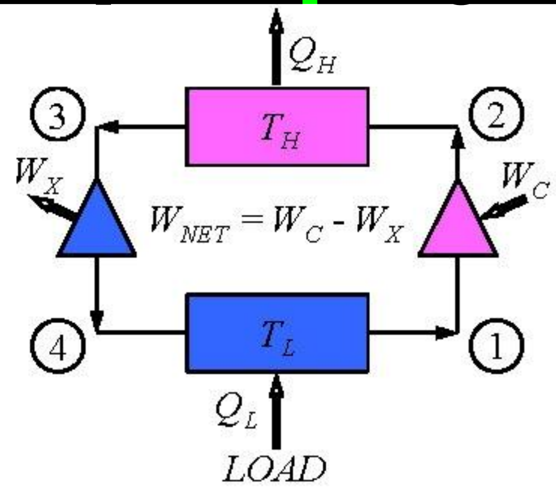
  "What is a refrigeration system?"

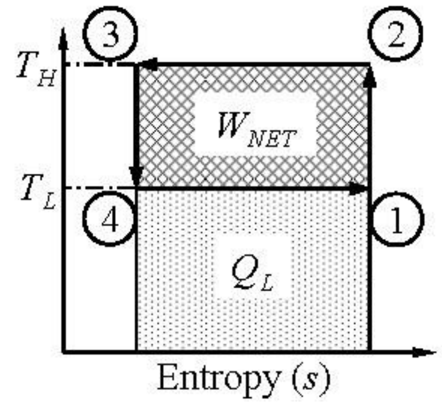
- **A refrigeration system *transfers* heat energy from *low temperature to high temperature*.**

- **Normally, the term refrigeration is used for absorbing heat energy at a constant temperature, but this does not have to be the case**

- **Let's look at an ideal vapor compression cycle, operating between two constant temperature reservoirs…**

Carnot Vapor Compression Refrigeration Cycle





#1 to #2: Compressor	• **Fluid is compressed isentropically (requiring W_C)**
#2 to #3: Condenser	• **Heat, Q_H, is rejected isothermally (at T_H)**
#3 to #4: Expander	• **Fluid is expanded isentropically (extracting W_X)**
#4 to #1: Evaporator	• **Heat, Q_L, is absorbed isothermally (at T_L)**

Net input work: $W_{CARNOT} = W_C - W_X = Q_H - Q_L = (T_H - T_L)\cdot\Delta S$

Cooling provided: $Q_L = T_L\cdot\Delta S$

Coefficient of Performance: $\mathrm{COP} = Q_L / W_{CARNOT} = (T_H/T_L - 1)^{-1}$

Inverse COP: $\mathrm{COP}_{INV} = 1 / \mathrm{COP} = T_H/T_L - 1$

Introduction (cont.)

- **The coefficient of performance (COP) is the ratio of the cooling provided to the required input power**
- **Carnot vapor compression cycle, the refrigerator operating between -10 and +50 °C**

$$COP = \{(273+50) / (273-10) - 1\}^{-1} = 4.4 \text{ W/W}$$

- **The Carnot work (W_{CARNOT}) for 4.4 kW of cooling is 1 kW**

Note: This is not a violation of the first law of thermo since a *refrigerator* is *transferring energy* from one temperature to another and *not converting it*

- **Thermodynamic efficiency is the ratio of the ideal input power to the actual required input power**

Introduction (cont.)

- **The <u>Carnot cycle</u> is an <u>'ideal cycle'</u> in the sense that it does not have any 'irreversibilities' (i.e., 'lost work')**

  for a given path from state 1 to 2, with no irreversibilities, the heat transfer is

$$Q = \int_1^2 T \cdot dS$$

<u>Note</u>: This is a statement of the 2nd law of thermodynamics

Introduction (cont.)

- The <u>Carnot cycle</u> is an <u>'ideal cycle'</u> in the sense that it does not have any 'irreversibilities' (i.e., 'lost work').

- *The term '<u>idealized cycle</u>' will be relegated to a <u>practical system</u>* that one can visualize using ***<u>ideal components</u>***

- The Carnot cycle has the maximum *COP* (or the minimum inverse *COP*) for the process of ***<u>transferring heat energy between two thermal reservoirs</u>***

- This distinction gives the *<u>'Carnot cycle'</u> the <u>recognized qualification</u>* <u>for 'efficiency' comparisons of other cycles performing the same function</u>.

Introduction (cont.)

- **For general process cycles an *exergy* (or 'reversible work') analysis is performed to determine the minimum required work input or maximum obtainable work output for an *ideal process***

Note: (Mass) specific physical exergy is defined as,

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

where, h **is the enthalpy [J/g]**

T_0 **is the reference, or 'zero' availability, temperature (say, 300 K environment)**

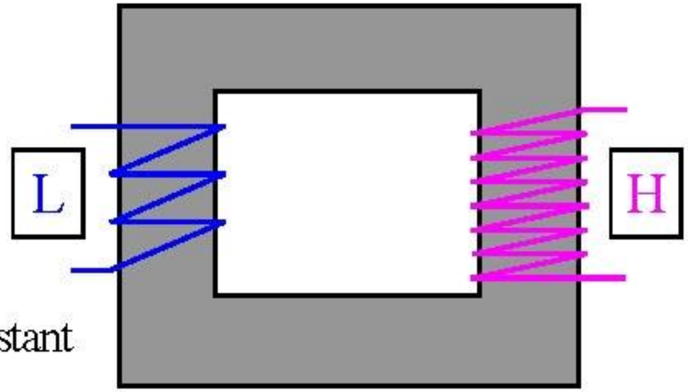
s **is the entropy [J/g-K]**

Introduction (cont.)

Ideal Electrical Transformer

$$\frac{V_H}{V_L} = \frac{I_L}{I_H}$$

$$P = V_H \cdot I_H = V_L \cdot I_L = \text{constant}$$



Ideal 'Thermal' Transformer

$$\frac{T_H}{T_L} = \frac{Q_H}{Q_L}$$

$$S = \frac{Q_H}{T_H} = \frac{Q_L}{T_L} = \text{constant}$$

- **Why is *__any__* input energy required to *__transfer__* heat energy from a cold to a hot temperature reservoir?**
- **A thermal transformer that permits the heat energy transfer from cold temperature to hot temperature, with no input work *__does not exist__*.**
- **This is quite unlike an ideal electrical transformer, which will permit the transfer between voltage and current with no additional input power.**
- **This 'transmission' (or transfer) limitation of heat energy between temperatures implies that there is a *__'quality'__* for *__heat energy__*.**
- **The *__source__* *and* *__sink__* temperatures sets this limit on the conversion *__'quality'__* *for the* *__heat energy__*.**

Introduction (cont.)

- **The minimum work required (or maximum work output), known as the *reversible work*, is *independent of the path* (from state point 1 to 2) *and the working fluid*.**
  - **This is a very important statement!**
- **In other words,**
  - **The selection of the process path (cycle) and the working fluid are based upon the desired working fluid properties (i.e., saturation temperature and pressure, latent heat, density, specific heat, viscosity, thermal conductivity, etc.) for the available practical components**
- **These selections are coupled,**
  - ***But do not determine the reversible work!***

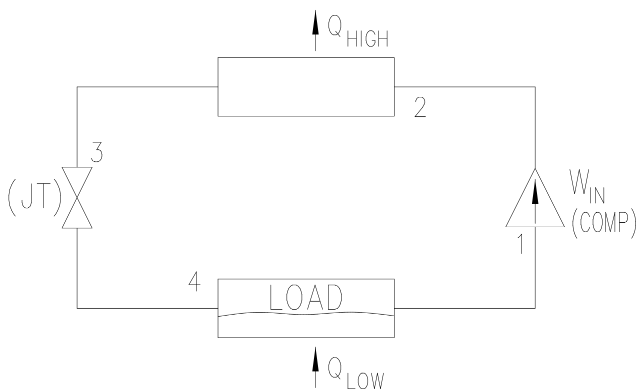
System Performance & Efficiency

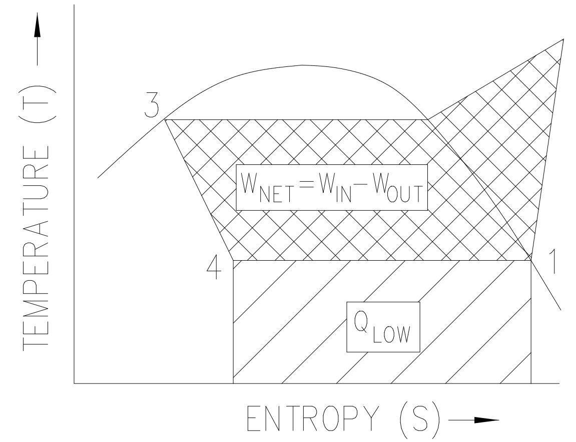
- For ideal systems the conversion from mechanical to electrical energy (or visa-versa) can be 100%.
- Approximately 3kW of thermal energy is required to produce 1kW of mechanical energy.
- This thermodynamic limitation is expressed by the 2nd Law of Thermodynamics and embodies the concept that the thermal energy has a 'quality' (or 'availability')
- For refrigeration, the input energy required is due to the loss in 'availability' (or decrease in 'quality') of the thermal energy as it is transferred from a low temperature (load) to a high temperature (environment).

Introduction (cont.)

Vapor compression process

e.g.: Typical Freon refrigerator



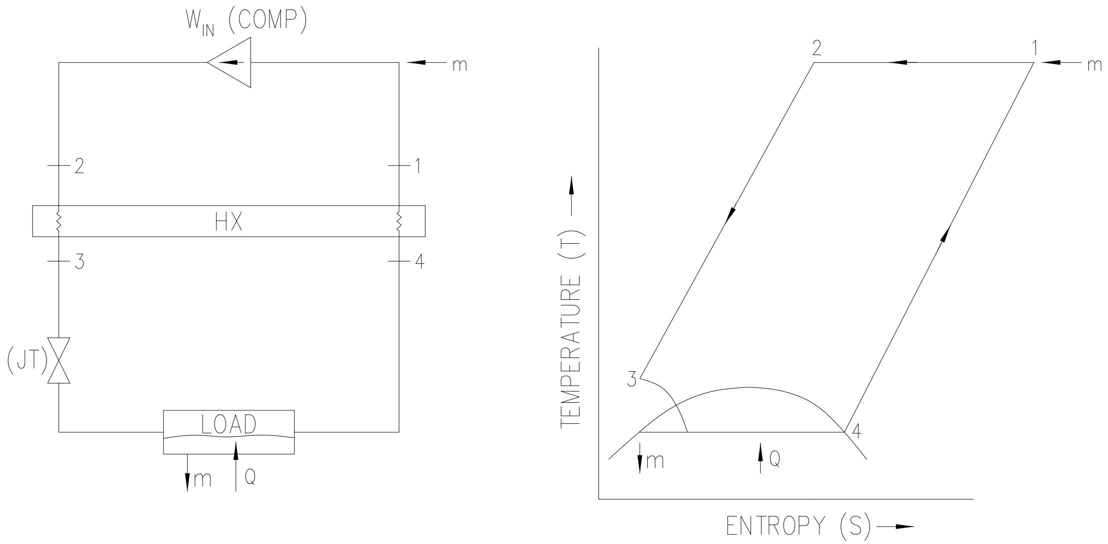


This process typically requires 1 kW of input power for ~3 kW of cooling load, so the efficiency as compared to the Carnot cycle, otherwise known as the exergetic efficiency is,

$$\text{exergetic efficiency} = \frac{W_{Carnot}}{W_{actual}} = \frac{Q_{LOW} \cdot COP_{INV}}{W_{actual}} = \frac{3 \cdot (0.23)}{1} = 0.68$$

Introduction (cont.)

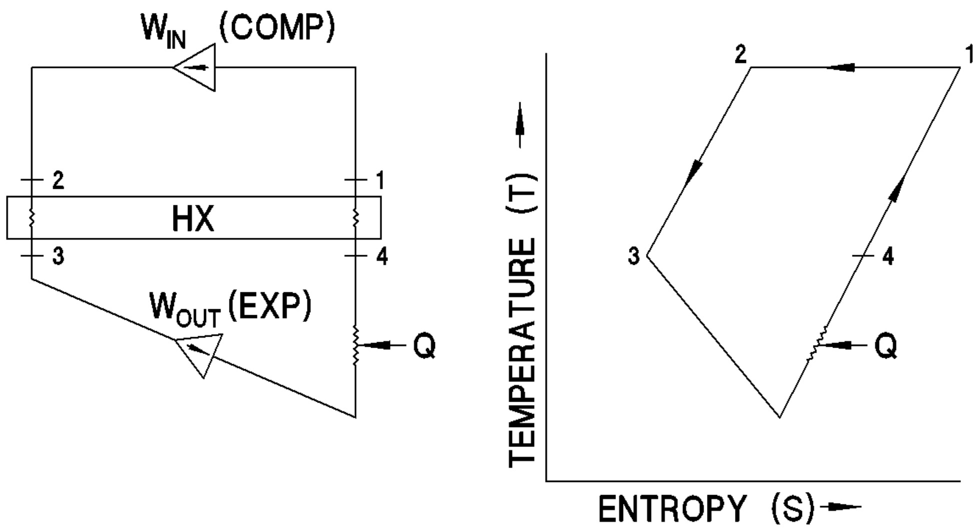
- **Hampson process**



- Uses a heat exchanger (HX) between the compressor and the load for heat energy exchange between the supply and return streams.
- Process supports lower temperature load operations more efficiently than the vapor compression process.

Introduction (cont.)

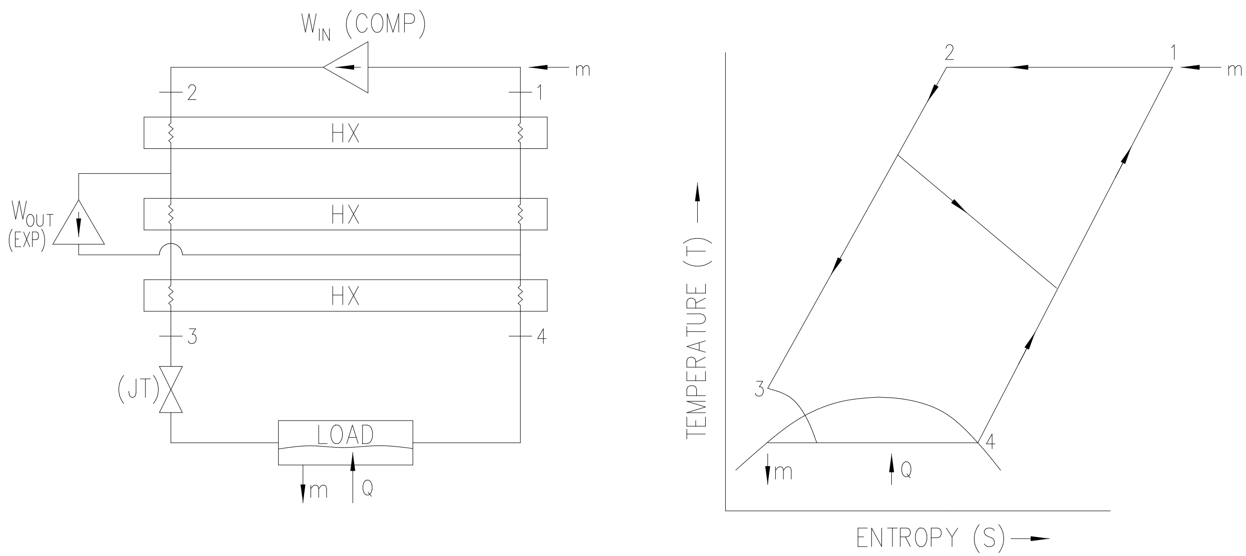
- **Modified Brayton process**



- Uses a heat exchanger (HX) and an expander between the compressor and the load
- Process supports lower temperature load operations more efficiently than the Hampson process

Introduction (cont.)

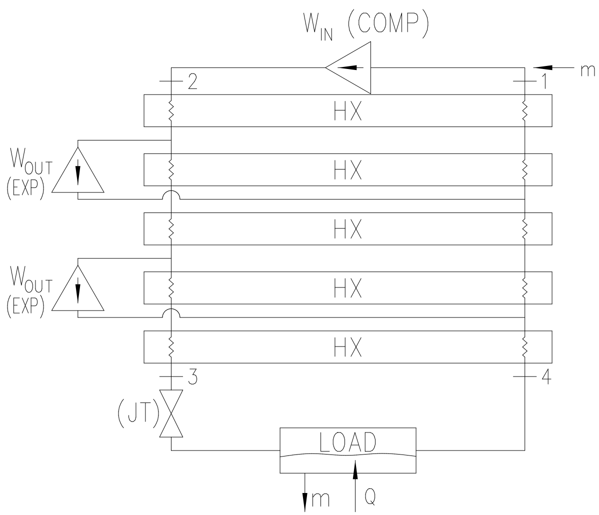
- **Claude process**

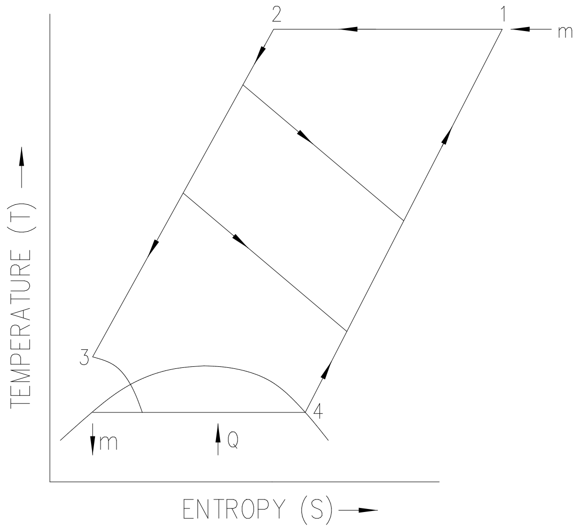


- Additional heat exchangers and an expander are used between the compressor and the load.
- Supports lower temperature load operations more efficiently than the Hampson process.

Introduction (cont.)

- **Collins helium liquefaction process**





- Process developed by Sam Collins [1] at MIT and is an extension of the Claude cycle.
- Supports lower temperature load operations more efficiently than the Claude cycle.
- The widely used helium liquefiers originally known as CTI-1400's were based on this process.

Introduction (cont.)

Summary - Key Ideas

- **Coefficient of performance and thermodynamic efficiency**
- **Carnot cycle (as a reversible cycle operating between two constant temperature reseviors).**
- **Quality of thermal energy**
- **Reversible work and fluid/process path independence**
- **Exergy analysis as a means to determine the reversible work for an arbitrary reversible process**
- **Present day cryogenic processes cycles are an extension of basic cycles modified to achieve better efficiency**

2. Carnot Helium Refrigeration and Liquefaction Systems

Before proceeding to the Carnot helium refrigerator and liquefier it is instructive to revisit the introduction to the 2nd Law of Thermodynamics.

Clausius (In)equality (the 2nd Law of Thermodynamics)

$$\frac{\Delta Q_L}{T_L} = \frac{\Delta Q_H}{T_H}$$

'quality' or 'availability'

<u>This equation is a statement of thermal energy quality equivalence</u>

<u>Eg:</u>

$$\frac{300W}{300K} = \frac{4W}{4K} = \frac{2W}{2K}$$

Carnot Helium Refrigeration and Liquefaction Systems (Cont)

Q_L = 1W at T_L = 4.22 K is equivalent in quality as

Q_H = 70 W at T_H = 300K

Ambient condition (i.e., 300K and 1 atm) is the 'zero-grade' energy state

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

$$W_{REV} = W_{ideal} = \sum_i (\dot{m}_i \cdot \varepsilon_i)_{IN} - \sum_j (\dot{m}_j \cdot \varepsilon_j)_{OUT}$$

<u>Carnot Refrigeration System:</u> **Min. input power for a given rate of *<u>thermal energy transfer between two thermal reservoirs.</u>***

The <u>work input</u> for the <u>Carnot system</u> expressed as:

$$W_{carnot} = T_0 \cdot \Delta S - \Delta H$$

<u>This is a very powerful equation</u>

- **The terms are as follows:**
- $T_0 \cdot \Delta S$ **is the heat rejected to the environment**
  or, the input power to an isothermal compressor
- ΔH **is the heat absorbed or the ideal refrigeration**
  or, the ideal work output from an ideal expander
- W_{carnot} **is the ideal net input work required**
  which is the difference between (a) and (b) above

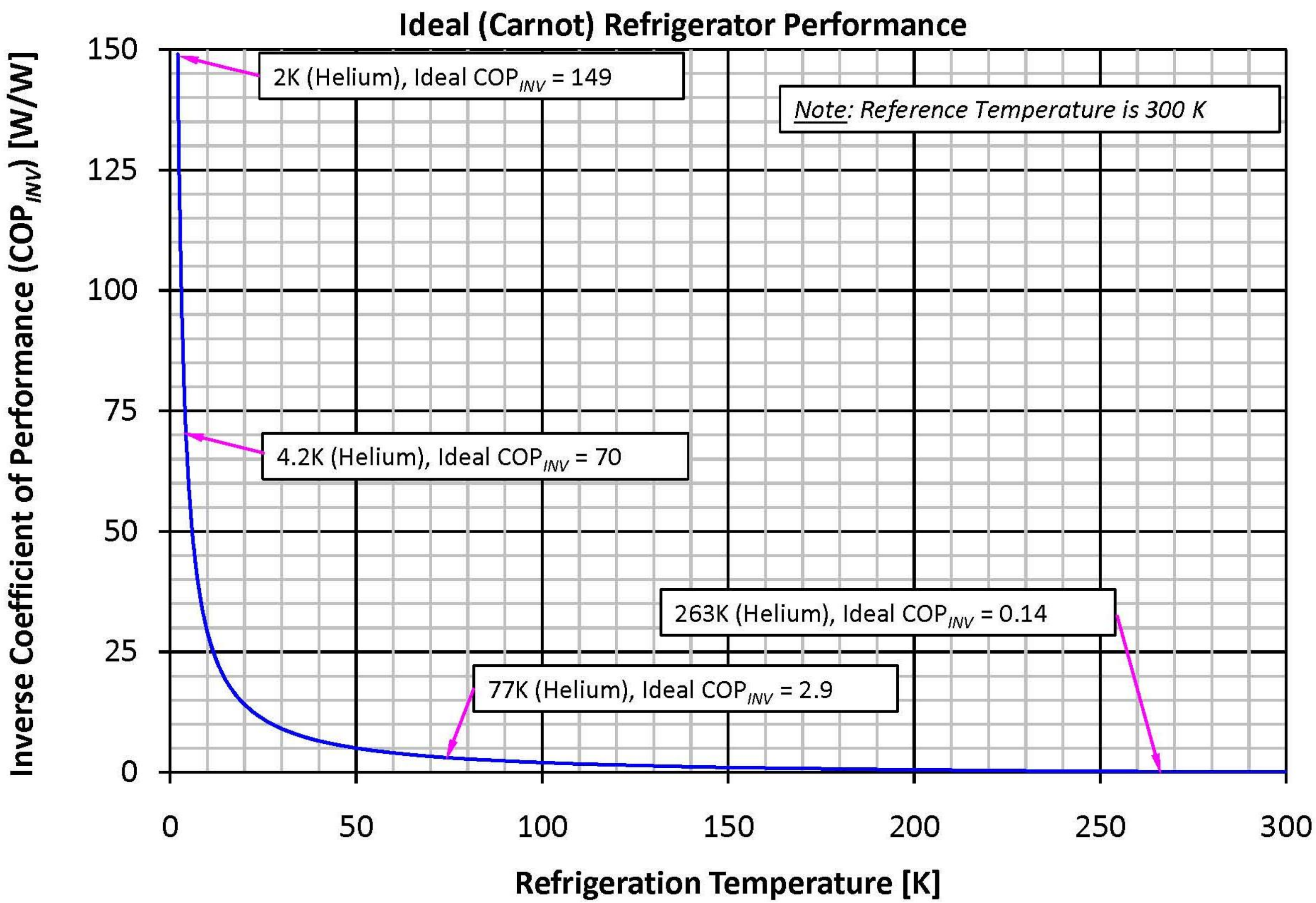
Carnot Helium Refrigeration and Liquefaction Systems (Cont)

- **A refrigerator *transfers* heat energy from a low temperature reservoir to a higher temperature reservoir.**

- **Most helium refrigerators transfer heat energy from approximately 4.22K to ambient 300K.**

- **A liquefier is different from a refrigerator since we cool high temperature fluid to a low temperature, which then leaves the cycle (at a low temperature). The heat energy removed is constantly varying (decreasing as it is being cooled), although it is rejected at the same (high or ambient) temperature reservoir.**

Carnot Helium Refrigeration and Liquefaction Systems (Cont)



	T_load (K)	Pcarnot W / W
A / C Sys.	263.0	0.14
Methane	111.7	1.69
Oxygen	90.2	2.33
Argon	87.3	2.44
Nitrogen	77.3	2.88
Neon	27.1	10.07
Hydrogen	20.3	13.79
Helium	4.2	70.09
Helium @Lambeda	2.2	137.25
Helium @ 2.0 K	2.0	149.00

Carnot Helium Refrigerator

By definition, a refrigerator *transfers* heat energy from a low temperature reservoir to a higher temperature reservoir.

- **The Carnot work for a refrigerator is as follows:**

$$w_{carnot} = T_0 \cdot \Delta s - \Delta h = \text{specific carnot work}$$

$$COP_{INV} = \frac{W_{carnot}}{Q_L} = \frac{T_0 \cdot \Delta S - \Delta H}{\Delta H} = \frac{\dot{m} \cdot (T_0 \cdot \Delta s - \Delta h)}{\dot{m} \cdot (\Delta h)} = \frac{T_0 \cdot \Delta s - \Delta h}{\Delta h}$$

with Q_L equal to the cooling provided

Carnot Helium Refrigerator (cont)

helium refrigerator operating between 300K ambient and the 4.22K

$$w_C = T_0 \cdot \Delta s = (300)\cdot(4.833) = 1449.9 \ \text{[W/(g/s)]}$$

$$w_X = \Delta h = 20.42 \ \text{[W/(g/s)] (or 1.4\% of } w_C\text{)}$$

$$w_{Carnot} = w_C - w_X = 1429.5 \ \text{[W/(g/s)] (or 98.6\% of } w_C\text{)}$$

$$COP_{INV} = \frac{W_{carnot}}{Q_L} = \frac{\dot{m}\cdot(T_0\cdot\Delta s - \Delta h)}{\dot{m}\cdot(\Delta h)} = \frac{(300)\cdot(4.833) - 20.42}{20.42} \cong 70\left[\frac{W}{W}\right]$$

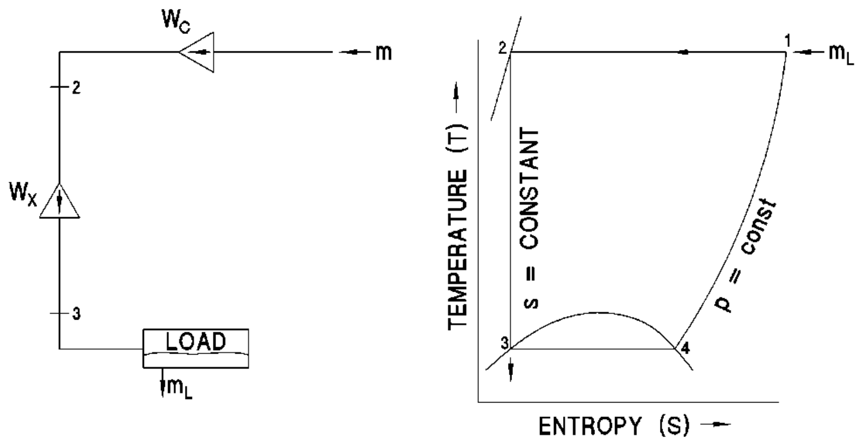
If the expander work is not recovered

$$COP_{INV} = \frac{\dot{m}\cdot(T_0\cdot\Delta s)}{\dot{m}\cdot(\Delta h)} = \frac{(300)\cdot(4.833)}{20.42} \cong 71\left[\frac{W}{W}\right]$$

<u>Note</u>: Non expander work recovered refrig. systems, start with a 1.4% efficiency penalty

Carnot Helium Refrigeration and Liquefaction Systems (Cont)

Carnot Helium Liquefier



$$w_C = T_0 \cdot \Delta s = (300) \cdot (27.96) = 8387 \text{ [W/(g/s)]}$$

$$w_X = \Delta h = 1564 \text{ [W/(g/s)] (or 18.6\% of } w_C\text{)}$$

$$w_{Carnot} = w_C - w_X = 6823 \text{ [W/(g/s)] (or 81.4\% of } w_C\text{)}$$

In *non expander work recovered* *liquefaction systems*, they start with 18.6% efficiency penalty.

Performance Comparisons of Helium *Refrigerators* and *Liquefiers*

$$\frac{\text{Carnot work required for liquefaction [W/(g/s)]}}{\text{Carnot work required for refrigeration [W/W]}} = \frac{W_{carnot}}{COP_{INV}} = \frac{6823\ \text{[W/(g/s)]}}{70\ \text{[W/W]}} \cong 100\ \text{W/(g/s)}$$

That is, the Carnot work required for approximately 100 W of refrigeration is equivalent (on an equal Carnot work basis) as the Carnot work required to liquefy 1 g/s at 1 atm saturation condition.

Performance Comparisons of Helium *Refrigerators* and *Liquefiers* (Cont.)

***If the expander output work is not recovered*,**

$$\frac{\text{Ideal Power required for liquefaction [W/(g/s)]}}{\text{Ideal Power required for refrigeration [W/W]}} = \frac{8387\ \text{[W/(g/s)]}}{71\ \text{[W/W]}} \cong 120\ \text{W/(g/s)}$$

That is, the Carnot work required for approximately 120 W of refrigeration is equivalent (on an equal Carnot work basis) as the Carnot work required to liquefy 1 g/s at 1 atm saturation condition If the expander output work is not recovered.

Performance Comparisons of Helium Refrigerators and Liquefiers (w & w/o Expander work recovery)

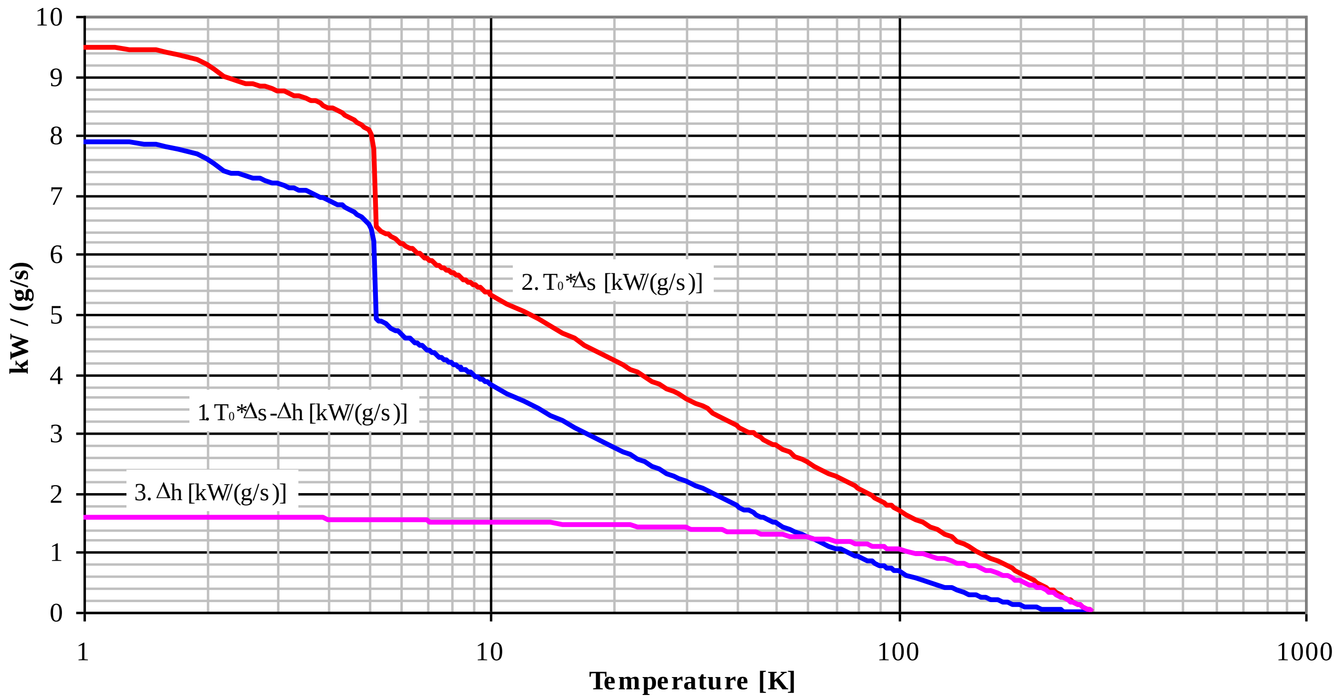
$$\frac{\text{Ideal Power required for liquefaction [W/(g/s)]}}{\text{Ideal Power required for refrigeration [W/W]}} = \frac{8387 \text{ [W/(g/s)]}}{71 \text{ [W/W]}} \cong 120 \text{ W/(g/s)}$$

$$\frac{\text{Ideal Power required for refrigeration [W/(g/s)]}}{\text{Ideal Power required for liquefaction [W/(g/s)]}} = \frac{\left(\frac{w_{Carnot}}{w_C}\right)_l}{\left(\frac{w_{Carnot}}{w_C}\right)_r} = \frac{\left(\frac{6823}{8387}\right)_l}{\left(\frac{1429.5}{1449.9}\right)_r} = \frac{81.4}{98.6} \cong 82.5\%$$

A refrigeration cycle having 30% of Carnot efficiency is expected achieve 25% in liquefaction mode

Carnot Helium Refrigeration and Liquefaction Systems (Cont)

- **Carnot work required for a given liquefaction load**

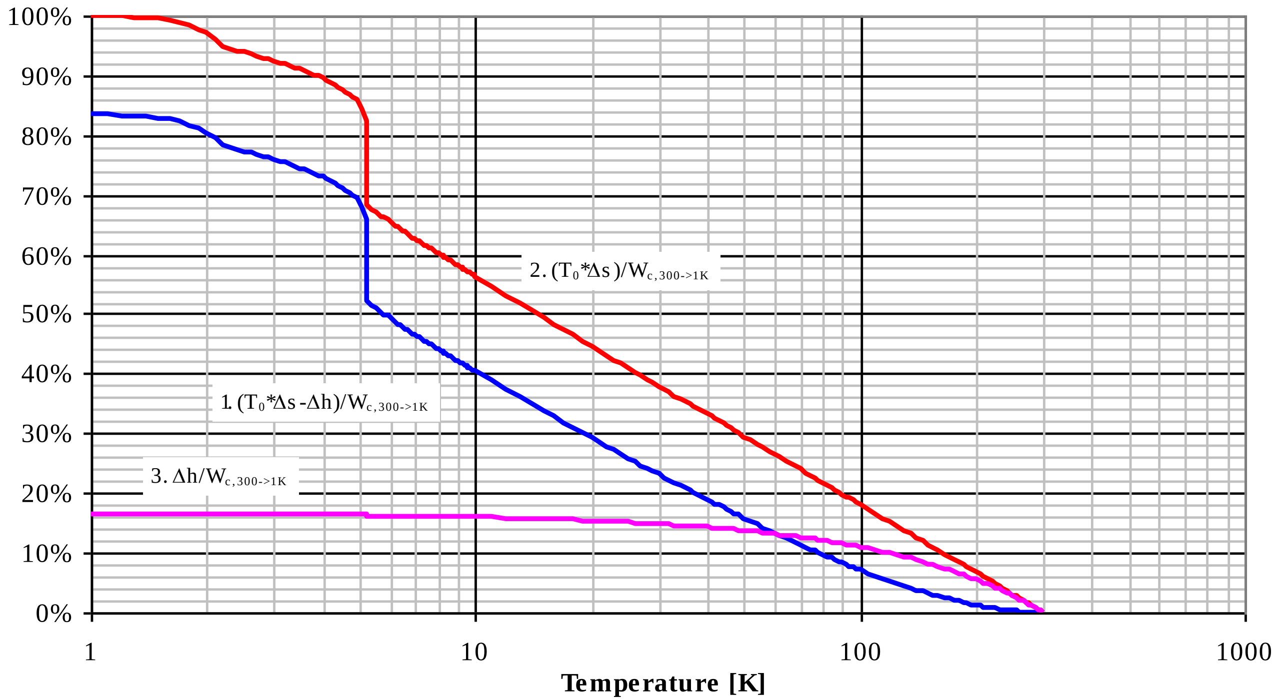


Carnot work [1. $(T_0 \cdot \Delta s - \Delta h)$] required to cool helium from 1 atm & 300K to the specified final temperature

isothermal compressor work [2. $(T_0 \cdot \Delta s)$] and the expander output work [3. (Δh)]

Carnot Helium Refrigeration and Liquefaction Systems (Cont)

Carnot work required for a given liquefaction load



Ratio (in %) of Carnot work, isothermal compressor work and expander output to a reference value for the isothermal compressor work (of 300 to 1K)

Carnot Helium Refrigeration and Liquefaction Systems (Cont)

Carnot work required for liquefaction load for a given temperature range

Temperature Range (K)		$T_0 * \Delta s$ [W/ (g/s)]	%		Δh [W/ (g/s)]	%	$T_0 * \Delta s - \Delta h$ [W/ (g/s)]	%
300 - 80		2058	**24.5%**		1143	**73.0%**	915	**13.4%**
80 - 4.22		6329	**75.5%**		421	**27.0%**	5908	**86.6%**
300 - 4.22		8387	**100.0%**		1569	**100.0%**	6823	**100.0%**

Carnot work for different fluids

Fluid	Tsat,0 [K]	Liquefaction (W/(g/s))	Refrigeration (W/W)
Helium	4.22	6823	70
Hydrogen	20.28	12573	13.8
Neon	27.09	1336	10.1
Nitrogen	77.31	770	2.9
Argon	87.28	477	2.4
Oxygen	90.19	635	2.3
Methane	111.69	1092	1.7

Summary

In this chapter the Carnot work (or the minimum input work) required for the refrigeration and liquefaction is explained.

the effects of non recovered expander work (generally the case for most of the helium systems) on the refrigeration and liquefaction processes.

In practice all the systems are compared to the ***true reversible Carnot work*** (W_{carnot}) , which includes the expander out put work

3. Ideal Helium Refrigeration Systems and Carnot Step

In this chapter we look at reversible cycles using an *ideal gas* and *perfect components*.

- **This will provide the *basis* for analyzing real systems.**

- **A system design based on an ideal system and constructed with real components should result in an efficient system.**

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

Carnot Step

- **Typically in helium (refrigerator) systems, there are multiples of certain similar non-simple process steps; e.g., warm screw compressor stages, expansion stages in a cold box, etc. to accomplish a given process.**

- **The *Carnot Step* is defined *(by the author)* as the arrangement (or "spacing") of a *given number* of the same type of process steps which yield the minimum irreversibility.**

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

Carnot Step (Cont.)

- **This optimal arrangement of process steps is applicable to ideal and real processes and will typically yield the minimum energy expenditure (for that process and selected components)**

- **It is important to note that *Carnot Step* is not necessarily a reversible 'step', since it depends on whether the process and/or components are reversible**

- **A typical helium system consists of:**

  (1) load, (2) cold box and (3) compressor

- **Clearly, an efficient system depends upon an efficient design of each part of the system**

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

The Load:

- **Every attempt should be made at the load level (temperature) to minimize the entropy increase of the helium, recovering the returned load flow exergy (refrigeration) while satisfying the load requirement.**

- **The losses introduced from a distribution system become a load as well, requiring the cold box and compressor system to be larger (i.e., greater capital cost), thereby incurring additional operational cost.**

- ***Example*: For a thermal shield between 300K and 4K with equal conductance on both sides, the idealized choice for the shield temperate to minimize the total reversible input power (i.e., the load Carnot Step) is found by equating the temperature ratios and is 35K**

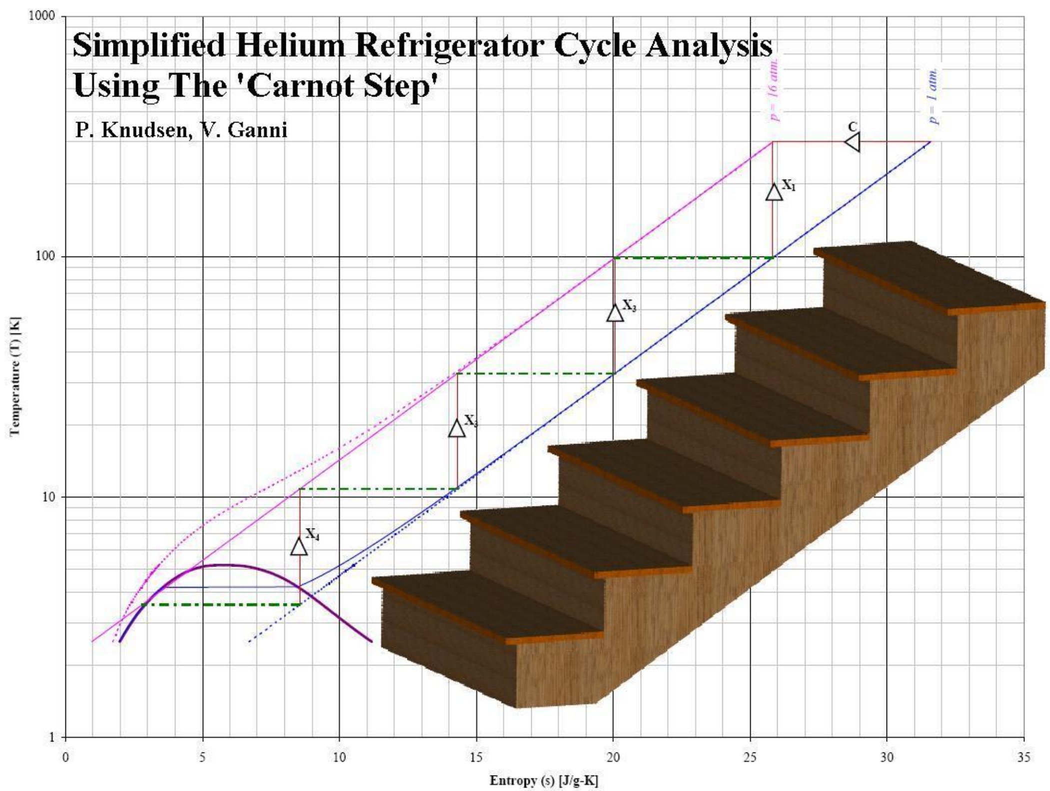
Idealized Helium Systems and Carnot Step (Cont.)

The Cold Box:

*The cold box **bridges** the temperature difference from the load to ambient conditions, transferring the entropy increase at the load to the compressors.*

The cold box has **no input power** and **can only utilize the availability (i.e., exergy) supplied to it by the compressor(s)**. Obviously, it is critically important for the cold box to utilize the supplied exergy with a minimum of 'losses'.

The cold box provides a process path analogous to transferring a load from a deep basement floor (4.2K) to the ground floor (300K) by walking up the stairs. So, given the 'height' between the 'floors' (4.2K to 300K), we would like to know the *minimum number and optimal spacing of the steps* that will yield a minimum irreversibility.



Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

The Cold Box:

- ***The expanders provide cooling (refrigeration) by extracting work. So, the <u>number of cold box (expansion or refrigeration) steps is same as the number of expanders</u> in the process.***

- **The next chapter (4) will address the optimal *Carnot Step* 'spacing' for a given number of steps that yields the minimum irreversibility (or minimum compressor input work).**

- **Therefore, the cold box Carnot Steps spacing (distribution) provides a means for evaluating a given cold box system design.**

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

The Compressor System:

- *The compressor system uses the input energy (usually electrical) to increase the availability (e.g., exergy) of the helium gas being supplied to the cold box.*

- **For a multistage polytropic compression process, an *equal pressure ratio* among each of the equal efficiency stages yields the minimum actual input work for a given mass flow rate.**

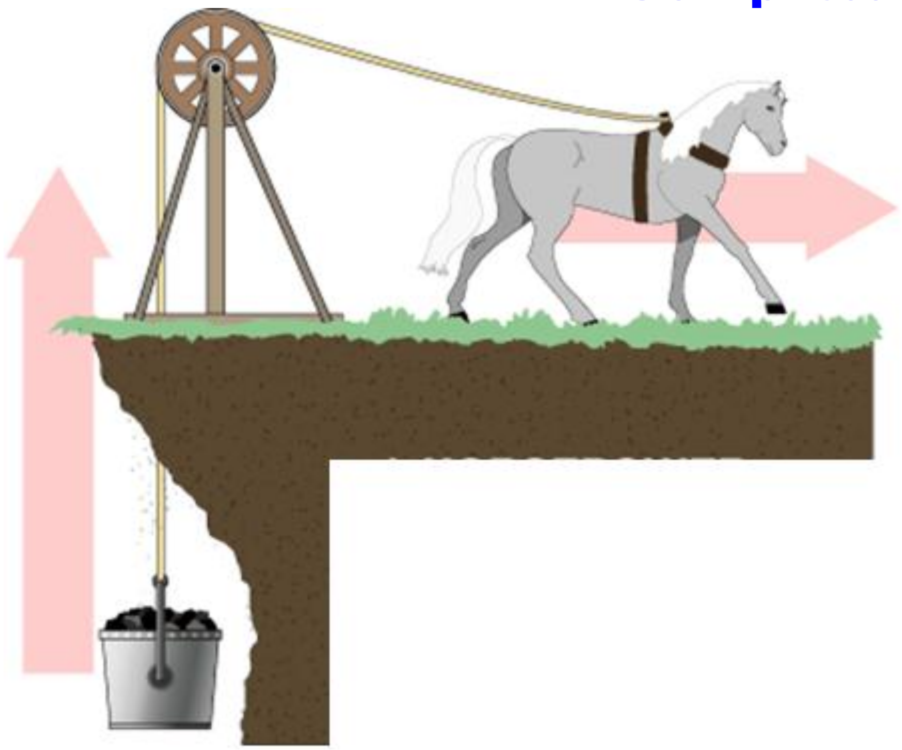
- **Since *isothermal compression* requires the minimum ideal work, it is used to determine the *compressor Carnot Step(s)***

- **Therefore, the compressor Carnot step provides a means for evaluating a given compressor system design (efficiency).**

Helium Refrigeration System Analogy

Cold Box

Compressor



Load

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

The Availability to the Cold Box:

- The availability (exergy) supplied to the cold box is the isothermal work input to the compressor system.
- It is also equal to the load Carnot work plus the cold box and load losses (i.e., irreversibility, or lost work).
- For a simple two-stream system, using the ideal gas assumption, the specific isothermal work is:

$$w_{C,\ iso} = \Delta\varepsilon_C = T_i \cdot \Delta s - \Delta h = T_i \cdot \phi \cdot C_p \cdot \ln(P_r)$$

where, for an ideal gas, $\Delta h = 0$ and, $\Delta s = \phi \cdot C_p \cdot ln(P_r)$

So, $\Delta s \propto ln(P_r)$

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

The Availability to the Cold Box:

- **From this it can be seen that the *availability (exergy) supplied to the cold box increases proportionally to the mass flow rate and the logarithm of the pressure ratio***

- **The required decrease in entropy (i.e., increase in exergy) can be achieved by increasing the mass flow rate or increasing the pressure ratio**

- **The analysis that follow are primarily centered on the cold box since the load is specific to a given application and the compressor performance is easily compared to the isothermal compression work**

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

Ideal Helium Refrigeration Systems Using the Ideal Gas:

- **The Carnot (reversible) work required for the refrigerator is,**

$$\Delta\varepsilon_{load} = w_{C,\ ideal} = (T_0 \cdot \Delta s - \Delta h) = \ [(300)\cdot(4.833) - 20.42] = 1429.5 \text{ W/(g/s)}$$

- **The inverse coefficient of performance is,**

$$COP_{INV} = \frac{w_{C,iso}}{(q_{load} / \dot{m}_{load})} = \frac{w_{C,iso}}{\Delta h} = \frac{[(300)\cdot(4.833) - 20.42]}{20.42} = 70 \text{ W/W}$$

So, then it requires 70 W of ideal (isothermal) input work for every 1 W of load

- **The pressure ratio for an isothermal compressor, *<u>operating with an ideal gas</u>* is,**

$$P_r = e^{\left[w_{C,\ ideal} / \left(\phi \cdot C_p \cdot T_0\right)\right]} = 9.91 \quad \text{with, } R = \phi \cdot C_p = 2.077 \text{ J/g-K}$$

<u>Note</u>: if real fluid properties were used, $P_r = 10.57$

<u>Ideal Helium Refrigeration Systems</u>:

- For an ideal gas, an ideal helium refrigerator should be constructed with the compressor operating between a 1 atm suction and ~10 atm discharge pressure
- In this process the gas is cooled to the load temperature at ~10 atm using the return gas and expanded <u>*isothermally*</u> (a limitation of the ideal gas assumption since an ideal gas only has a single phase) while adsorbing the refrigeration load
- So, there is only a <u>*single Carnot step*</u> (or single expansion step) for the <u>*ideal gas helium refrigerator*</u>

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

Ideal Helium Liquefier Using the Ideal Gas:

The Carnot work required for the liquefier is,

$$w_{C,iso} = T_0 \cdot \Delta s = (300)\cdot(27.96) = 8387 \text{ [W/(g/s)]}$$

$$w_{X,ideal} = \Delta h \qquad = 1564 \text{ [W/(g/s)] (or 18.6\% of } w_{C,iso}\text{)}$$

$$w_{C,ideal} = w_{C,iso} - w_{X,ideal} \qquad = 6823 \text{ [W/(g/s)] (or 81.4\% of } w_{C,iso}\text{)}$$

Note*:** ***the expander output work is used to reduce the isothermal compression work; for real systems, this is not practical

- ***So, then it requires 6.823 kW of ideal (net isothermal) input work for every 1 g/s of liquefaction load***

- **As for practical systems, the expander output work is not recovered , so that the pressure ratio for an isothermal compressor, *operating with an ideal gas* is,**

$$P_r = e^{\left[w_{C,\,iso}/\left(\phi\cdot C_p\cdot T_0\right)\right]} = 700{,}000 \quad \text{with, } R = \phi\cdot C_p = 2.077 \text{ J/g-K}$$

Idealized Helium Liquefier:

- For an ideal gas, an ideal helium liquefier should be constructed with the compressor operating between a 1 atm suction and a (approximately) 700,000 atm discharge pressure

- It is not a practical option

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

Idealized Helium Liquefier (Cont.):

- This ideal process is obviously not a very practical liquefier, but it does present the fact that heat energy is being transferred at a temperature that continuously varies

- In other words, the liquefaction load is really a refrigeration 'load' whose load temperature begins at 300K and continuously decreases as it is cooled, finally ending at the (specified) load supply temperature (typically 4.2K)

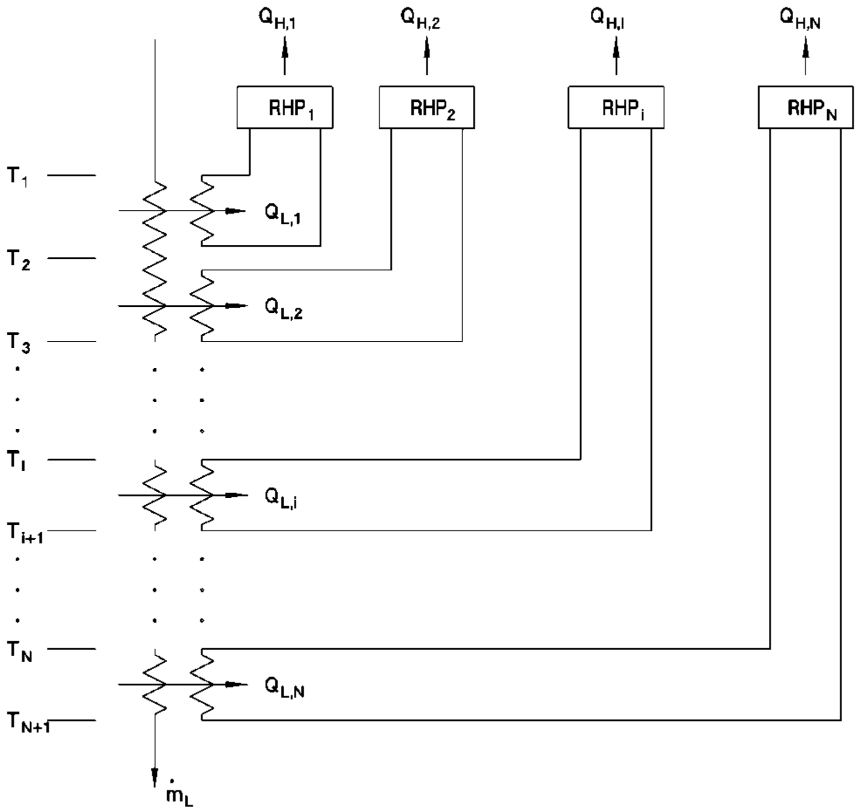


Figure 3.4.1: Ideal liquefier process

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

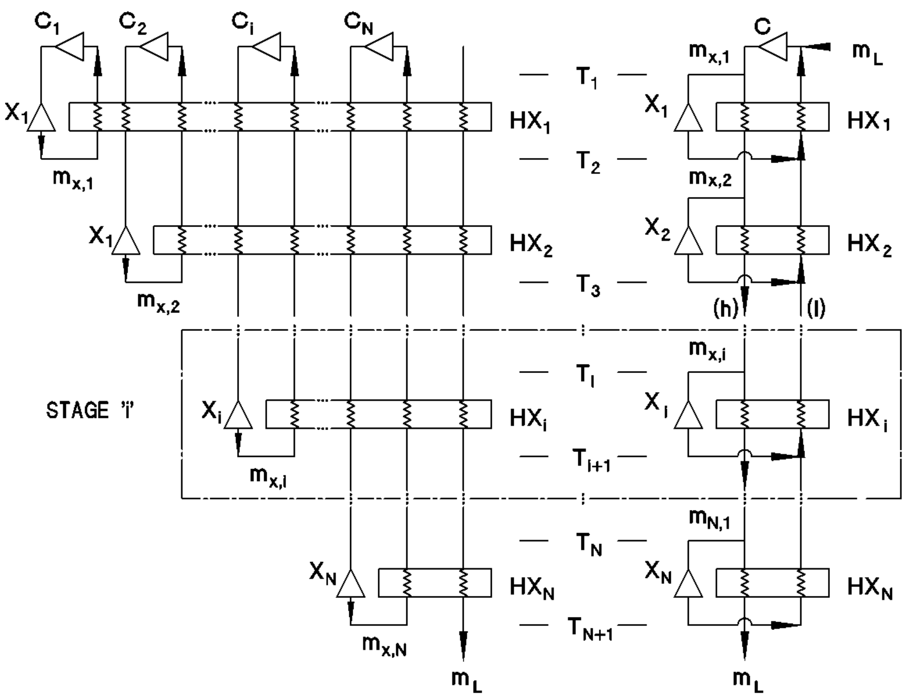


Figure 3.4.2: Ideal Claude Liquefier (ICL)

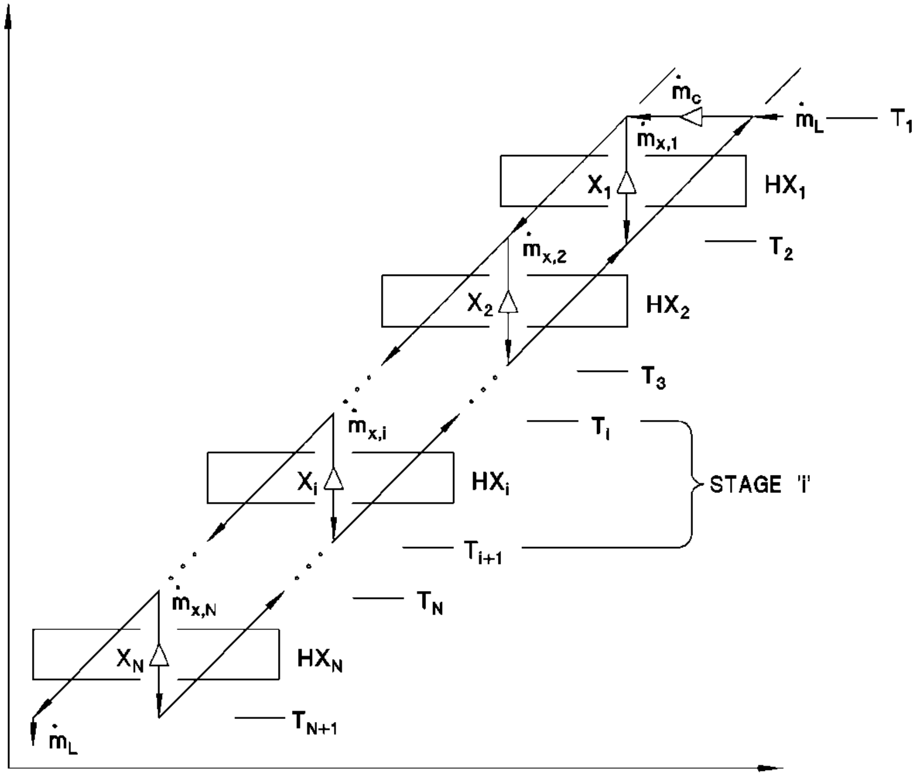


Figure 3.4.3: TS Diagram for the Ideal Claude Liquefier (ICL)

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

- **From Figure 3.4.3 and using ideal gas and isentropic process relations,**

$$T_r = P_r^{\phi} = const.$$

$$T_{r,T} = \left(\frac{T_1}{T_{N+1}} \right) = (T_r)^N$$

$$N = \frac{\ell n(T_{r,T})}{\phi \cdot \ell n(P_r)} = \frac{\ell n(T_{r,T})}{\ell n(T_r)}$$

Where, T_r – expander temperature ratio (***Carnot Step***) $= T_i / T_{i+1}$

P_r – pressure ratio across expanders $= p_h / p_l$

T_1 – ambient temperature [K]

T_{N+1} – load temperature [K]

$T_{r,T}$ – total temperature ratio (ambient to load temperature) $= T_1 / T_{N+1}$

N – total number of (ideal) expansion stages

Ideal Helium Refrigeration Systems and Carnot Step (Cont.)

- So, ***the Carnot step is the same for each expander stage*** (i.e., T_r is the same for each stage) ***and equal to the expander temperature ratio (which is set by the pressure ratio***

- As an example, for a 300K to 4.2K liquefier (e.g., T_1 = 300K, T_{N+1} = 4.2K) with an expander pressure ratio of 16 (e.g., P_r = 16), the total temperature ratio is, $T_{r,T}$ = 300 / 4.2 = 71 and the temperature ratio for each expander stage is, $T_r = (16)^{0.4} = 3.03$

- So, the (ideal) number of expander stages required for the ICL is, $N = \ln (71) / \ln (3.03) = 3.85 \approx 4$

- Referring to Figure 3.4.3, each expander flow is the same and equal to the liquefaction flow

- As we will see in Chapter 4, this is only true if there is a single perfect HX for each stage (or step); otherwise the expander flow is greater than this

Summary

- The ***'Carnot Step'*** is one of a given number of similar process steps that yield the minimum irreversibility.

- Assuming an ideal gas, the ideal ***refrigerator requires only a single Carnot step*** whereas the ***ideal Claude liquefier requires a number of Carnot steps***, each with the same temperature ratio and expander mass flow

4. The Theory Behind Cycle Design

In this chapter we look at *idealized helium system(s)* and *possible practical systems* that one can visualize.

The *idealized* system ***may or may not be a reversible system*** depending on the process and the fluid that is used.

A system design based on an idealized system and constructed with real components should result in an efficient (or even an optimum) system.

An analogy to carrying (transferring) a load up from a deep basement floor (4.22K) to the ground floor (300K), the system design must select a process path (or a cycle if performed continuously).

This path can be depicted on a TS or a Te diagram and in our analogy, can be inclined as steps or vertical as an elevator. Generally a 'straight line of approach' path is preferred, except as necessary to accommodate imposed constraints of local deviations.

Dewar Process

Before discussing the design of refrigeration systems, it is very important to understand the load(s) and their effect on the system design.

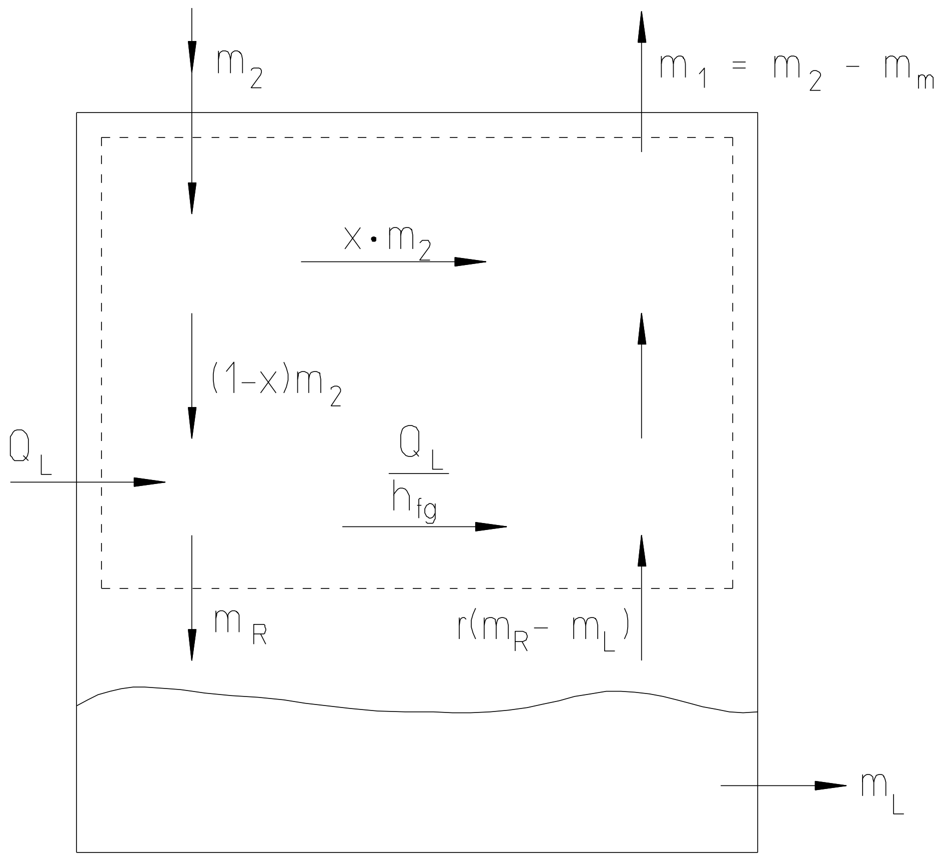
The load and the distribution systems are analyzed first (to minimize entropy generation; see sec 3.1.1) and must be understood before proceeding.

Since the loads and distribution system are project specific, only typical loads can be analyzed here.

Consider a simple system with loads interacting with a helium dewar, as shown in Figure 4.1.1.

The Theory Behind Cycle Design (Cont.)

Dewar Process



The Theory Behind Cycle Design (Cont.)

Dewar Process

$X = \text{Quality of } m_2$ $\qquad$ $r = \dfrac{\rho_g}{\rho_f}$

$m_R = (1-X)\, m_2 - \dfrac{Q_L}{h_{fg}}$ ……(1) $\qquad$ $m_m = (1-r)\, m_R + r\, m_L$ ……(2)

Case 1: $\quad m_L = 0 \quad m_R = \dfrac{m_m}{(1-r)} \quad \rightarrow \quad m_m = (1-r)\left[(1-x)\, m_2 - \dfrac{Q_L}{h_{fg}}\right]$

Case 2: $\quad m_R = m_L \quad \rightarrow \quad m_R = m_m \quad \rightarrow \quad m_m = m_R = m_L = \left[(1-x)\, m_2 - \dfrac{Q_L}{h_{fg}}\right]$

For Dewar Boil off (heat leak into Dewar) test: $m_2 = m_L = 0$

Flow leaving the Dewar $\quad = -m_m = \left[(1-r)\dfrac{Q_L}{h_{fg}}\right]$

Or Dewar Heat leak $\quad Q_L = \left[\dfrac{-m_m}{(1-r)} h_{fg}\right]$

Dewar Process

It is very important for helium systems to account for the mass of the displaced vapor from the Dewar during the filling process (i.e., the ratio of vapor to liquid density is ~ 1 / 7.4 at 1 atm.).

It is important to note that the rate of rise is greater than the makeup rate for the no withdrawal case.

As such, proper accounting is required in helium system liquefaction measurements since the rate of rise ($\dot{m}_R$) compared to the makeup helium ($\dot{m}_m$) can account for a 15% to 35% higher rate of production (depending on whether the dewar pressure is 1.0 or 1.6 atm) as given by equation (2), if the liquid withdrawal is zero.

This effect is not significant for other fluids with (typically) small vapor to liquid density ratios (e.g., 1 / 175, for 1 atm. nitrogen).

Real Gas Helium Refrigeration Systems

Case-1: Consider the *idealized refrigerator* explained in section 3.3 using a real gas, an isothermal compressor, a single ideal heat exchanger (effectiveness = 1) and a single ideal expander (isentropic efficiency = 1) at the cold end.

Real Gas Helium Refrigeration Systems

For this configuration shown in Figure 4.3.1, with a given load temperature, there is a unique solution for the high pressure supply to the cold box.

For a load at the 1 atm. saturation condition, the high pressure supply must be approximately 70 atm if the real gas is helium.

This is ***not*** a reversible process since the heat exchanger has a non-zero cold end ΔT equal to the difference between the expander inlet temperature (~7.7K) and the load temperature (~4.2K).

However, the usefulness of this cycle is in presenting the effect of the *real gas* and the minimum number, size and location of the components similar to the ideal system described in sec. 3.3.

The Theory Behind Cycle Design (Cont.)

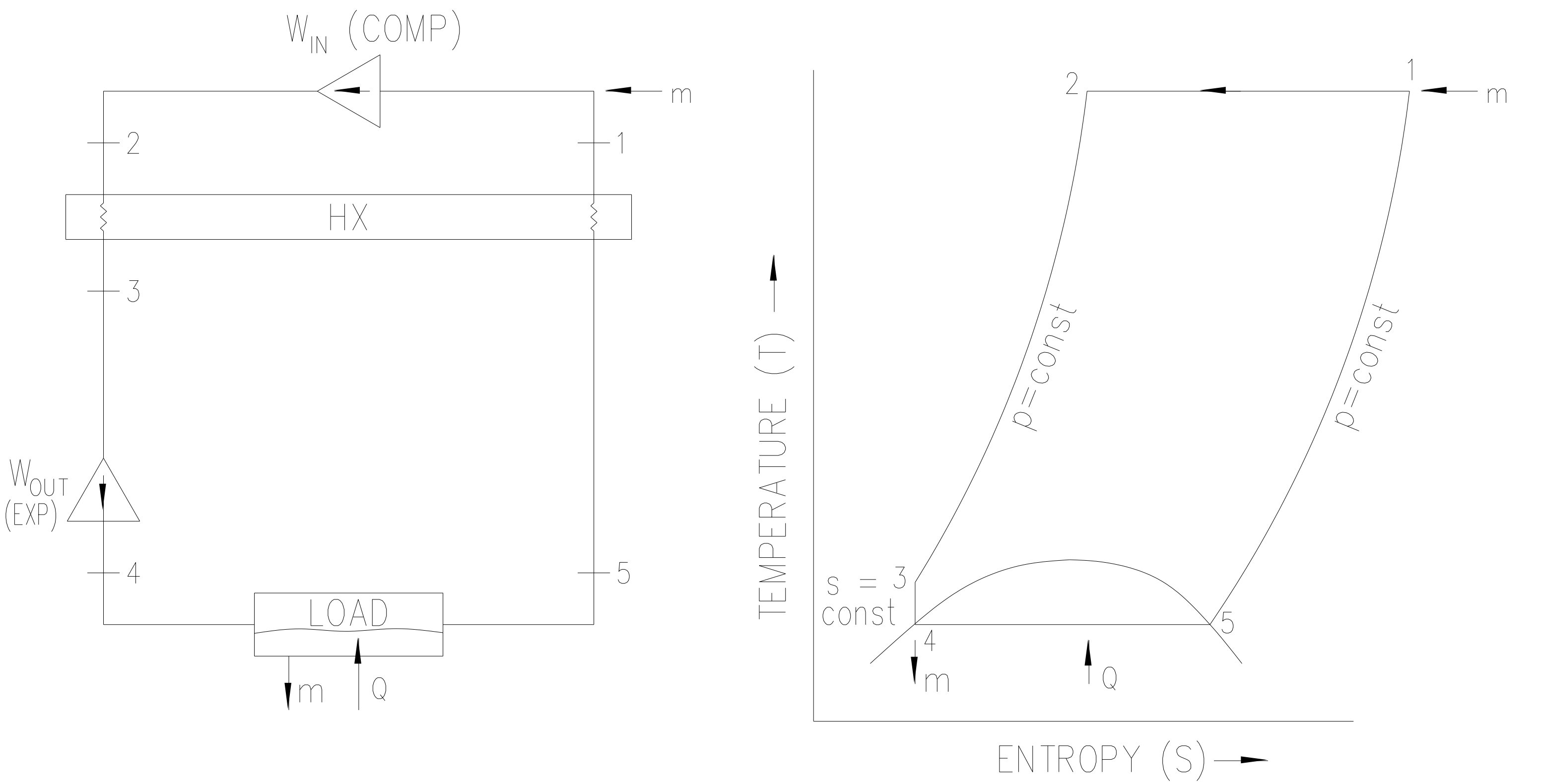


Figure 4.3.1: Idealized helium refrigeration system operating with real gas

The Theory Behind Cycle Design (Cont.)

For helium, the isothermal compressor work for this cycle is,

$$w_{C,\,iso} \cong R \cdot T_0 \cdot \ln(P_r) = (2.077)\cdot(300)\cdot\ln(70) = 2647 \text{ W/(g/s)}$$

$$w_{C,\,iso} \cong T_0 \cdot \Delta S \qquad = 300\cdot(31.41 - 22.59) \quad = 2646 \text{ W/(g/s)}$$

$$COP_{INV} = w_{C,\,iso} / \Delta h = (2646 - 20.42)/20.42 = 128.6 \text{ W/W}$$

As shown above, this cycle requires 1.8 times more input power (a Carnot efficiency of 55%) than the idealized cycle described in section 3.3. The efficiency loss in the above cycle is due to the non-ideal, real fluid properties.

Real Gas Helium Refrigeration Systems (Cont.)

Case-2: The above Case presents the influences and the importance of the real fluid properties in refrigeration process.

This is mainly due the real fluid transition from gas to liquid. Consider the following *idealized refrigeration process* to understand the real fluid property's influence.

In the system studied in Figure 4.3.2., the process assumes that at any given temperature there is a

constant entropy difference

between the supply and the return flows.

The Theory Behind Cycle Design (Cont.)

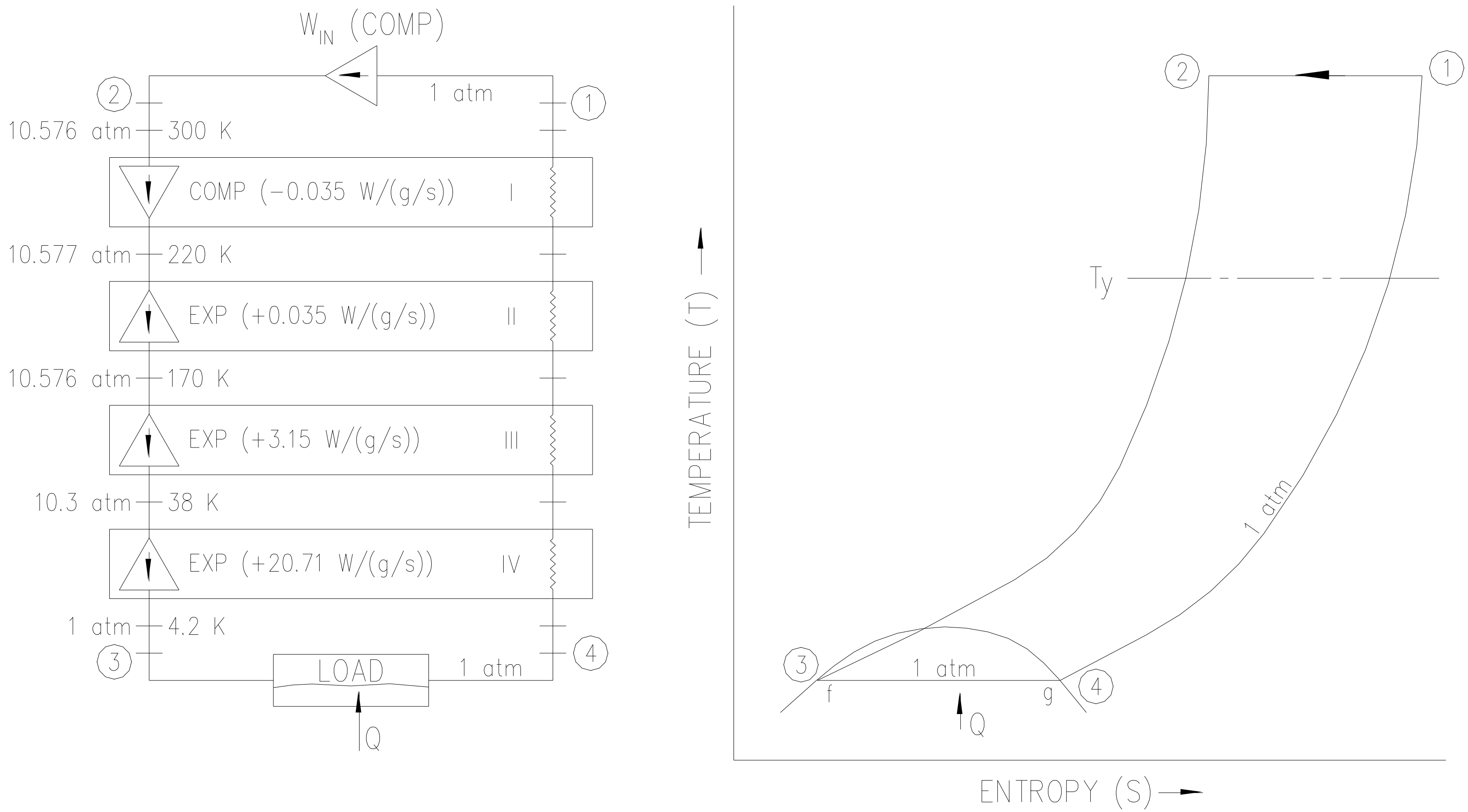
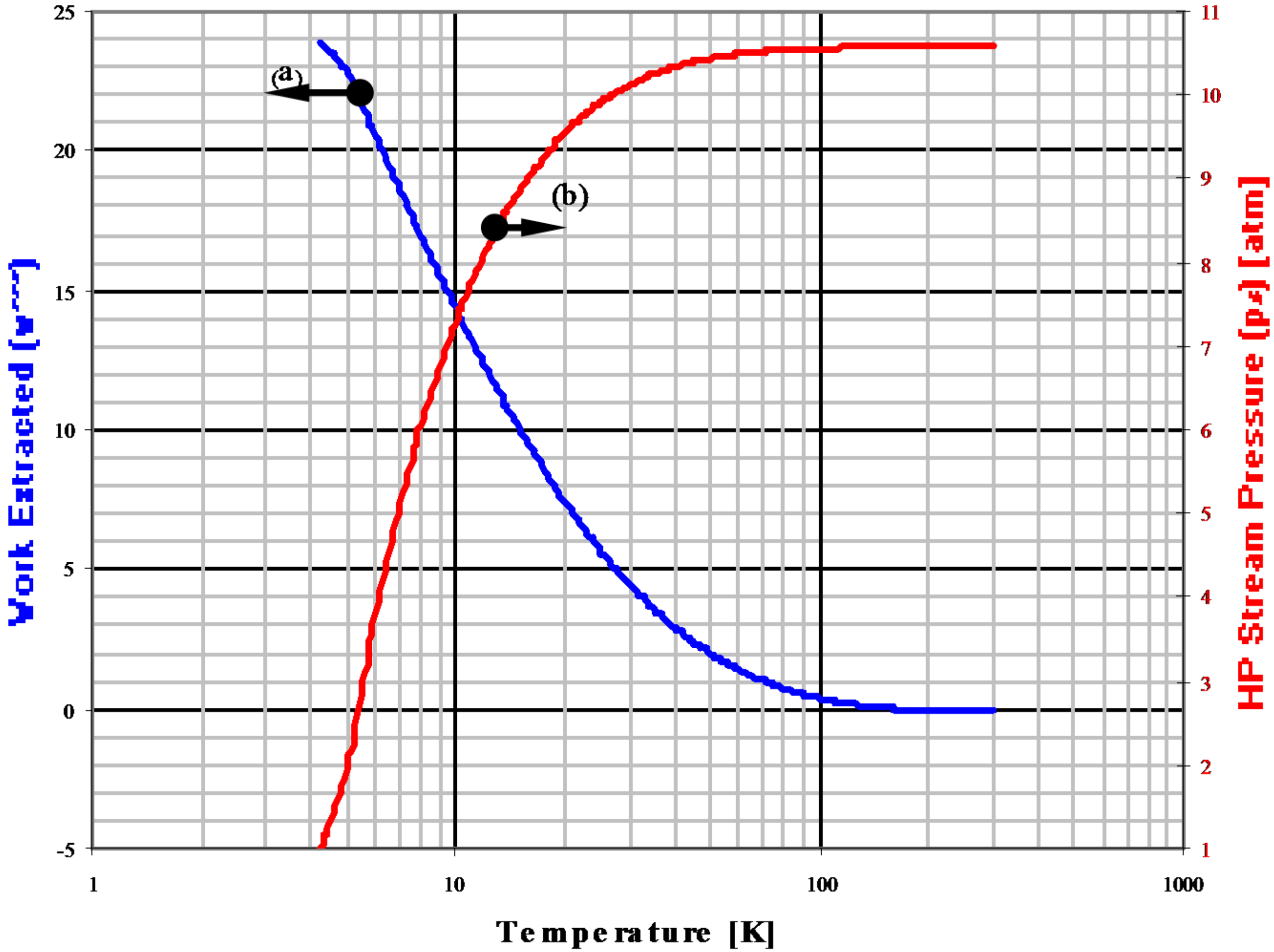


Figure 4.3.2: Idealized helium refrigeration system operating with real gas and ***constant entropy difference***

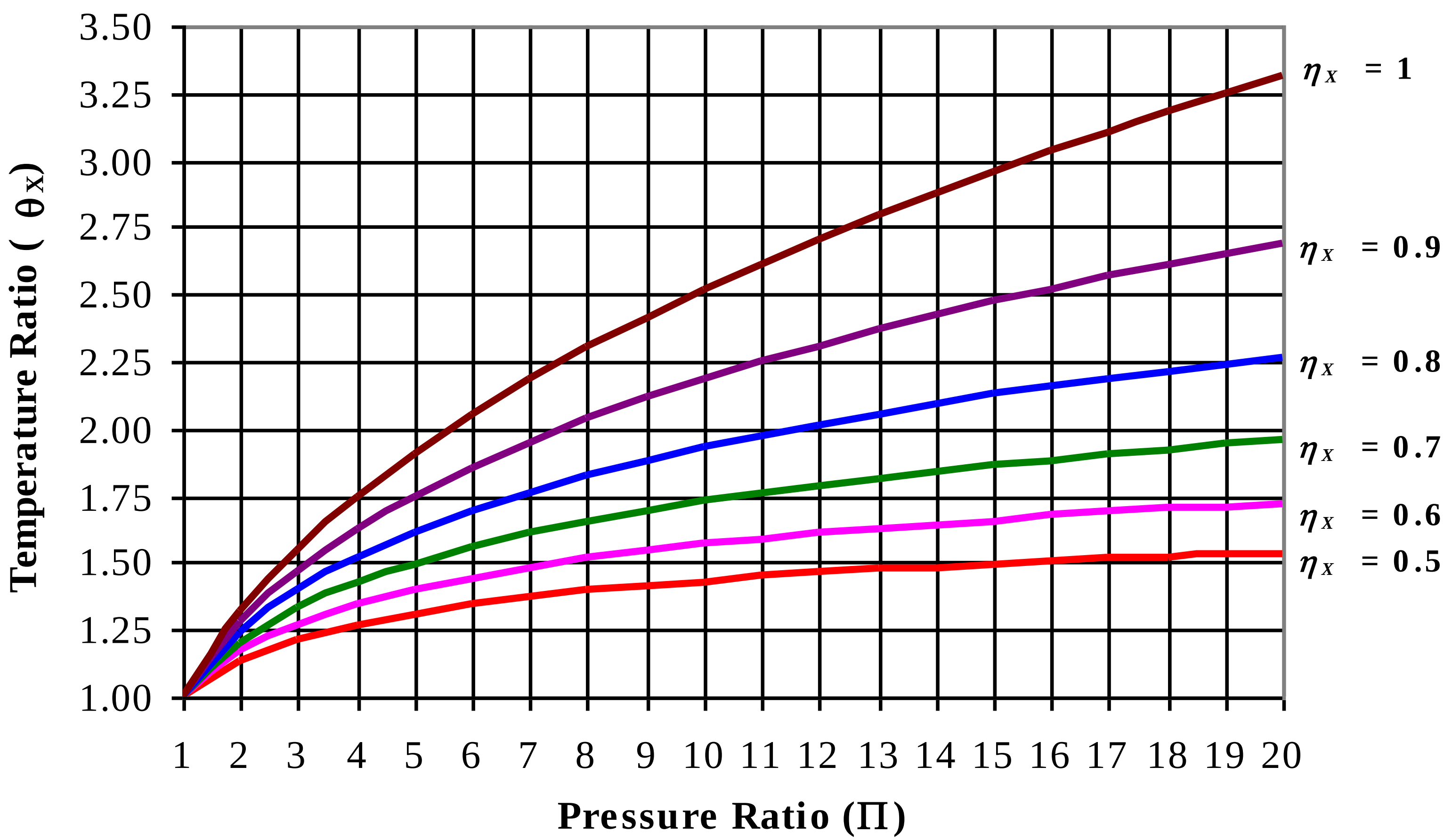
The Theory Behind Cycle Design (Cont.)

Idealized helium refrigeration system (cont.)



The Theory Behind Cycle Design (Cont.)

Expander pressure ratio vs. temperature ratio for selected efficiencies



The Theory Behind Cycle Design (Cont.)

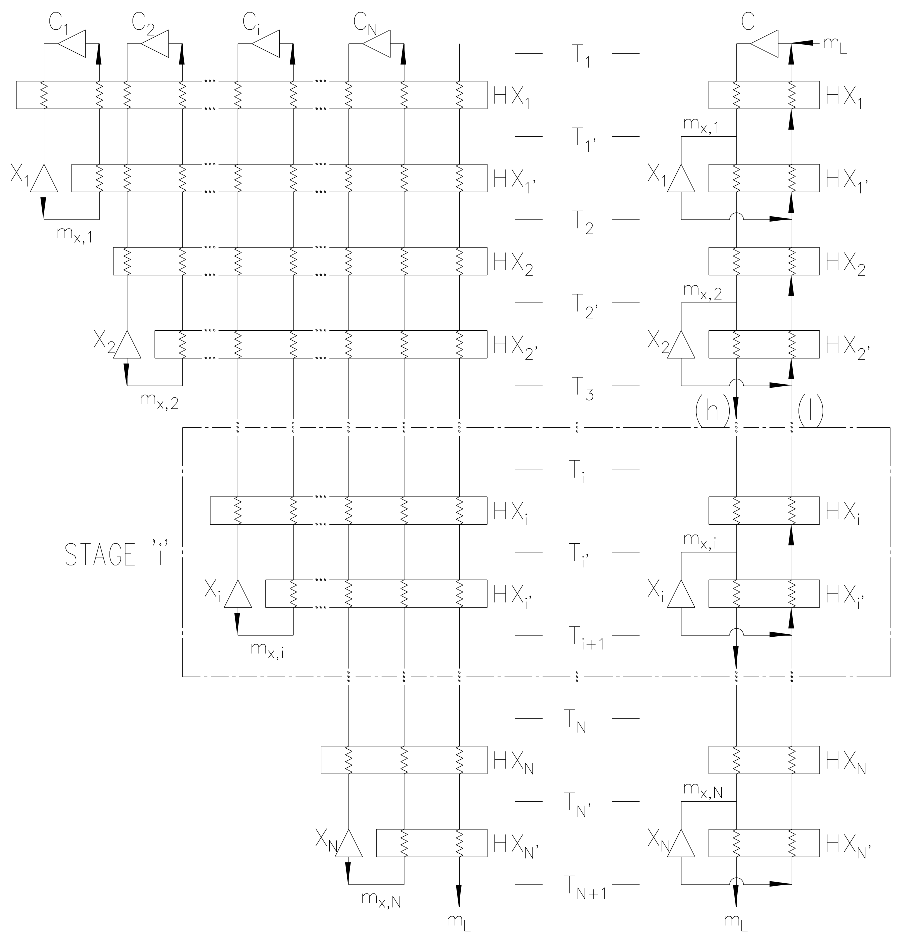


Figure 4.4.2: Claude liquefier with additional HX per stage

The Theory Behind Cycle Design (Cont.)

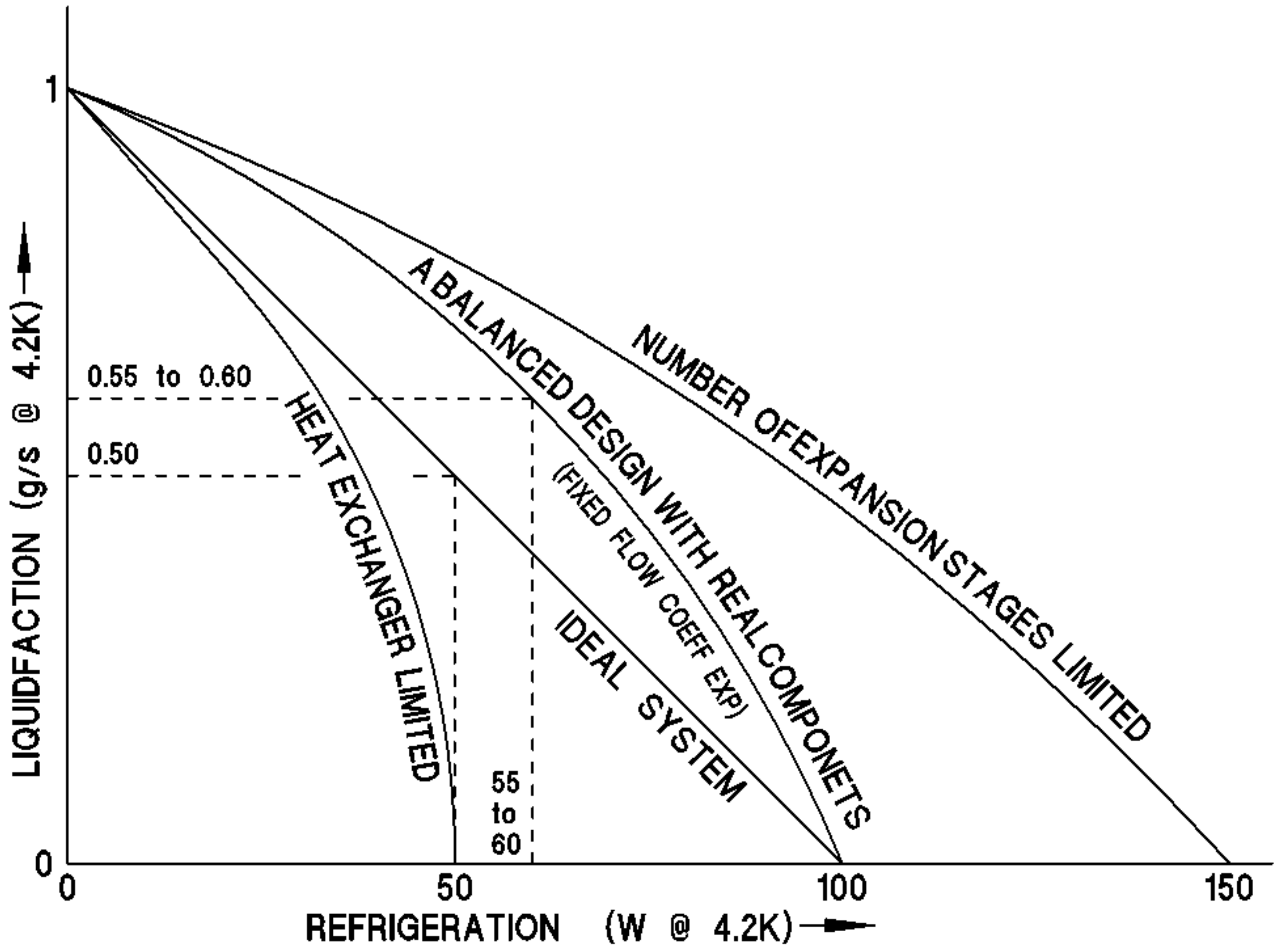


Figure 4.8.1: **The Effect of Components on System Load Capacity**

Summary

The use of the *Carnot step* for cold box design.

For a given number of expansion stages (with equally efficient expanders), these Carnot steps (the stage temperature ratios) are *theoretically* the same for both refrigerator and liquefier and result in minimizing the compressor flow and therefore, the input power.

This is indirectly saying that the ideal placement of the expanders with respect to temperature for both refrigerator and liquefier are approximately (disregarding real gas effects) the same, but the flow requirement through the expanders may not be the same.

***Note:* The importance and ramifications of the Carnot step was recognized by the author in the mid 80's. Since then it has been taught to colleagues and utilized in new system designs, as well as improving the operation of existing systems.**

5. System Optimization

Experienced designer follows and understands the developments of the helium Refrigeration systems over the years.

Here is an attempt to present some of the advances in the filed and their practical basis.

It is *easy to ask* to provide an *Optimum System* to support a given load

Requires serious thought to answer

System Optimization (Cont.)

What is an optimum system?

Does it result in a:

- **Minimum operating cost**
- **Minimum capital cost**
- **Minimum maintenance cost**
- **Maximum system capacity**
- **Maximum availability of the system**

Traditionally a design for maximum efficiency at one operating point is referred as the optimum system design.

System Optimization (Cont.)

- **The above five factors (or perhaps more) are rarely looked at as the optimization goals.**

- **The demand on equipment varies substantially between operating as a refrigerator (i.e., Hx dominance) and liquefier (i.e., expander dominance).**

- ***The challenge is to envision a cycle considering these optimization goals, using <u>real components,</u> capable of operating close to <u>maximum efficiency</u> for a load varying from a maximum to minimum capacity and from full refrigeration to full liquefaction mode or in any partial load combinations.***

System Optimization (Cont.)

- The majority of the above goals can be accomplished with a system design based on a process naturally responding to (track) the loads.
- Considerable interdependency exists between the above five factors.
- A well-designed system is a result of optimizing the specified main factors (prioritized project requirements) and an overall optimization of the remaining factors.
- If an analysis for all the possible operating modes is completed at the design stage, it will identify the factors compromised and the type and magnitude of the effects.

System Optimization (Cont.)

- The trade-off relationship between the first two factors, the minimum capital cost and minimum operating cost can be quantified to some extent by the following guidelines.
- The first step is to establish a cycle that suits the expected loads using the guidelines described in earlier chapters.
- The exergy analysis shows (Appen-G) how much of the actual input energy each component uses in performing its duty.
- The effect of these losses can be studied by modifying the independent input parameters.
- As an example, if the warm end temperature difference for HX-1A is reduced, LN2 usage is reduced. It is a balance of the cost of an increased HX size vs. that of a reduced utility cost.

System Optimization (Cont.)

- **In the process industry, typically \$1000 of capital investment is worthwhile if it reduces the electrical input power by 1 kW (@~\$0.04/kWh)**
- **1 kW depending on the local cost of electrical power:**
  - **\$1000 (for \$0.04/kWh) to \$2500 (for \$0.10/kWh).**
  - **It assumes a 3-year pay back for an 8500-hr. operation per year.**

$$PV = (25000) \cdot f \cdot C_E$$

where, PV - Equivalent capital investment per 1 kW saved,

f - fraction of the year the plant is operated,

C_E - local cost of electricity [\$ per kWh]

This is a very simplified view.

Pressure ratio constraints

A minimum mass flow rate will provide a minimum of heat exchanger losses, smaller cold box, smaller compressor size and higher efficiency for a given load.

- This requires the maximization of the pressure ratio.
- The final compressor discharge pressure (in atm) is almost the same as the total pressure ratio.
- Many of the critical components used are rated for e.g., 25 atm for turbo expanders, 18 atm for reciprocating expanders.
- The pressure ratios selected for the cold box need to match the types of compressor to maximize efficiency
- 150# components are rated for ~20 atm at 100ºF and below
- 300# components are rated for ~50 atm at 100ºF and below.

Pressure ratio constraints (Cont.)

- **Care should be exercised before crossing the pressure rating boundaries**
- **A higher pressure ratio has a negative effect on their reliability**
- **Oil flooded screw compressors peak efficiency between 2.5 and 4.0 per stage.**
- **More than half the total exergy is lost (nominally ~50% isothermal efficiency) in providing the pressure ratio.**
- **Most turbo expanders pressure ratio between 2 and 5.**
- **Reciprocating expanders have their high efficiencies at higher pressure ratios.**
- **Cold Box pressure ratings are normally 20 atm to permit the use of 150# components in the system design.**

Temperature (or Temperature ratio) Constraints

- **Higher pressure ratio systems require fewer Carnot steps.**
- **Carnot step establishes the characteristic temperatures required in the cycle for the efficient cold box design.**
- **Efficient system design requires the maximization of the number of Carnot steps.**

Number of Carnot steps depend on:

- **For smaller systems, the efficiency of the expanders and the increase in investment (cost) of each additional Carnot step plays a significant role in choosing the number of Carnot steps.**
- **For larger systems, the analysis made with real fluids and various arrangements will lead to the optimum number of Carnot steps.**

Mass Flow Constraints

The compromises made in choosing the pressure ratio and the number of Carnot steps (or non Carnot step selection for the design) can result in higher mass flow through the cold box and resulting in:

- **Increase the size of the heat exchangers (cold box).**
- **Increase the heat exchanger thermal losses.**
- **Increase the pressure drop.**
- **Increase the capital cost of the system.**

System Optimization (Cont.)

Expander Flow Coefficient Considerations

For efficient cold box design, the Carnot step sets the expander flow

- The Carnot step imposes a temperature ratio for each step
- For the liquefaction load the mass flow is approximately constant
- For the refrigeration load the flow demand is on the cold expander(s)

- In practice two types of expanders are used in the helium systems:
  (a) reciprocating and
  (b) turbo expanders.
  - Most turbo expanders have fixed nozzles,
  - but some large systems have variable nozzle turbo expanders.

System Optimization (Cont.)

Expander Flow Coefficient Considerations (Cont.)

- **Easy to efficiently change the flow capacity of a reciprocating expander**
- **To change the flow for turbo expanders, the inlet pressure or temperature must be changed.**
- **The Carnot step sets the inlet temperature to the expander in an optimal design**
- **The large flow capacity variation for refrigeration and liquefaction modes can only be obtained by varying expander inlet pressures.**
- **This can be done by allowing the entire system pressure to increase or decrease to match the loads (variable pressure system)**
- **The process cycle for balanced system design provides the means to address these issues.**

System Optimization (Cont.)

Heat Exchanger (HX) Considerations

HX's should be selected after analyzing both the liquefaction and refrigeration modes, and preferably after examining all off-design modes.

- **For HX's with effectiveness greater than 95%, special design care is required for the flow distribution in the HX core.**
- **Some practical guidelines for cycle designs are to limit the effectiveness not to exceed 98.5% and any single HX core not to exceed 50 NTU's.**
- **The choice of *<u>horizontal orientation of HX's</u> should be the <u>last resort</u> due to inherent flow distribution problems (especially at turn down conditions).***

The Tradeoff Relationships

- The cycle analysis should include an exergy analysis (Appendix-G).
- 300 to 80K pre-cooling choice in deign is explained later.
- Sometimes load(s) exceeds its ideal (design) operating point
- Requires a new (or the maximum possible) capacity of the existing equipment or with limited modifications
- the system is optimized for maximum capacity.

System Optimization (Cont.)

A) **The optimization is now centered on minimizing any new investment**

- **In this regard, the efficiency (operating cost) has been declared less important (than maximizing the capacity)**
- **consequence, compromises have to be made regarding the maintainability, reliability and availability of the system.**

B) High peak and low average load.

- It is neither cost effective nor efficient for continuous operation to size the equipment to handle the peak load.
- an example of this is a quench from a large magnet string system.
- Dewars have been designed to absorb this large quench energy
- Appendix-B provides an analysis for sizing the dewar size

Appendix-B

System Optimization (Cont.)

C) A system designed with minimal moving parts for maximum reliability

- By properly conceiving this requirement in the beginning.
- This is accomplished by choosing highly reliable components
- and providing the redundant components (e.g. spare compressor skid)
- This approach can prove the maximum system availability.

System Optimization (Cont.)

D) The trade-off relationship between the maintenance cost, maximum system capacity and maximum reliability of the system depends upon

- **how close to and how long the system is operated at the maximum pressures (i.e: system capacity).**
- **how the system operating at a reduced capacity when the maximum capacity is not required.**

E) In practice, a helium system with a high efficiency (low operating cost) also has a low capital cost.

- **high efficiency systems require less flow and therefore**
- **fewer or smaller compressors and**
- **smaller heat exchangers and cold box.**
- **It may require more expander stages, the number of expansion stages must be balanced**

This is contrary to the intuition of many people.

System Optimization (Cont.)

Historically, helium cryogenic systems borrowed the main sub-systems from other applications, refrigeration systems and from the air separation industry

- This is an opportunity to develop and/or improve these components and operating practices (refer to Chapter 14).
- An example is
  - — operating screw compressors with a built in variable volume ratio (presently available) to match the varying system pressures
  - — and to operate close to the maximum efficiency or the minimum input power.
- All too often and unfortunately the combination of the loads and the available systems to process them are already in place and the operator has very little influence in changing this situation.

Floating Pressure Process - System Optimization (Cont.)

The Basic Floating Pressure System Design

Also referred to as the "Ganni Cycle" or "Floating Pressure Ganni Cycles" or "Constant Pressure Ratio Cycle".

The new process variation has been developed to maintain high plant operational efficiencies at full and reduced plant capacities for the helium cryogenic refrigeration and liquefaction cycle.

Traditional cycles are designed at specified maximum capacity operating point(s). In actual systems the loads often vary. Also the components used in the system do not always perform exactly as envisioned in the design, which are traditionally represented by the TS design diagrams.

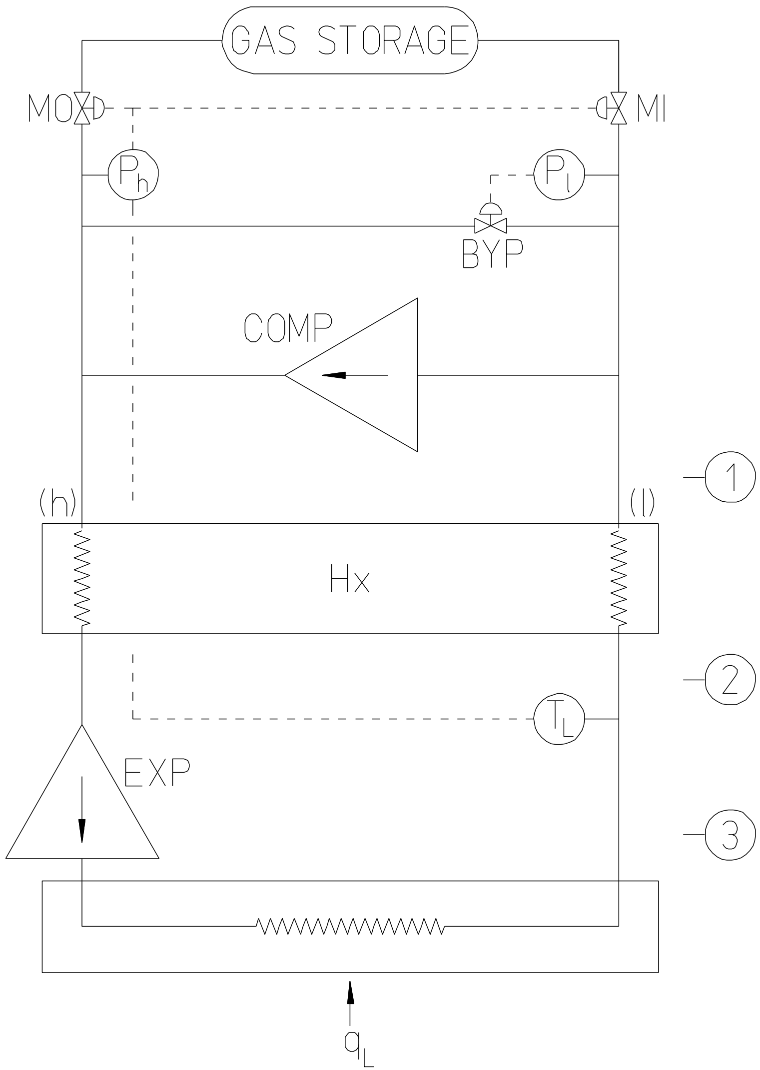
Floating Pressure Process - System Optimization (Cont.)

As such, for design and off-design modes, it has been traditionally the practice to force the plant to operate at the design pressure and temperature levels established in the cycle design (referred to as the TS design conditions) by regulating the turbo expander inlet valves, thereby (presumably) keeping the sub-components close to their peak (design) efficiencies

The Floating Pressure Process – Ganni cycle has no such bias and instead adopts a non-interference control philosophy using only a few key process parameters.

***The Floating Pressure Process invalidates the traditional philosophy that the TS design condition is the optimal operating condition for as-built hardware and actual loads*.**

Floating Pressure Process - System Optimization (Cont.)



Floating Pressure Process - System Optimization (Cont.)

Both the expander and compressor are essentially *constant volume flow devices*, so for a given mass charge they *set their own inlet pressures*, thus,

- **Compressor establishes the *suction pressure***
- **Expander establishes the *discharge pressure***

With these,
the gas charge establishes the system mass flow rate

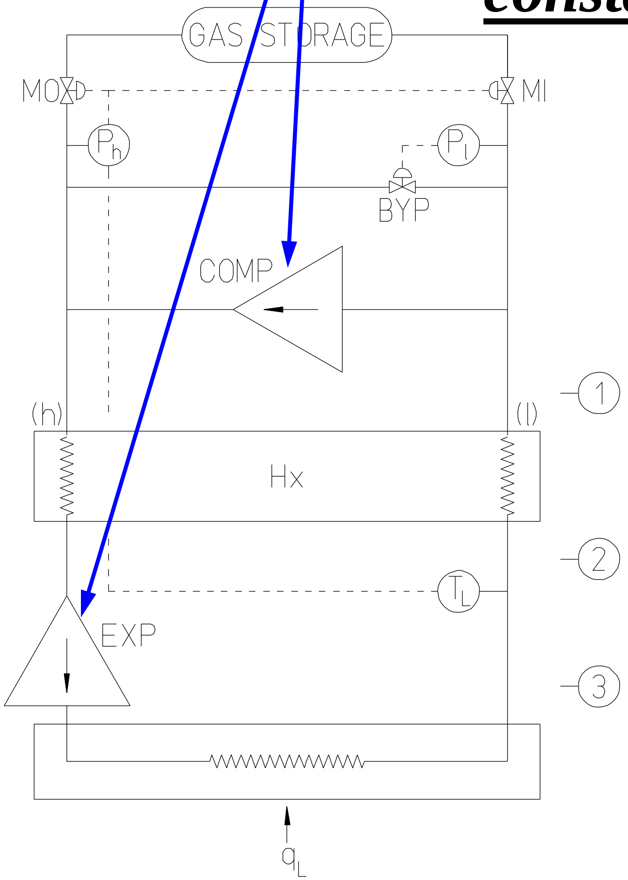
If left unconstrained, these two devices establish

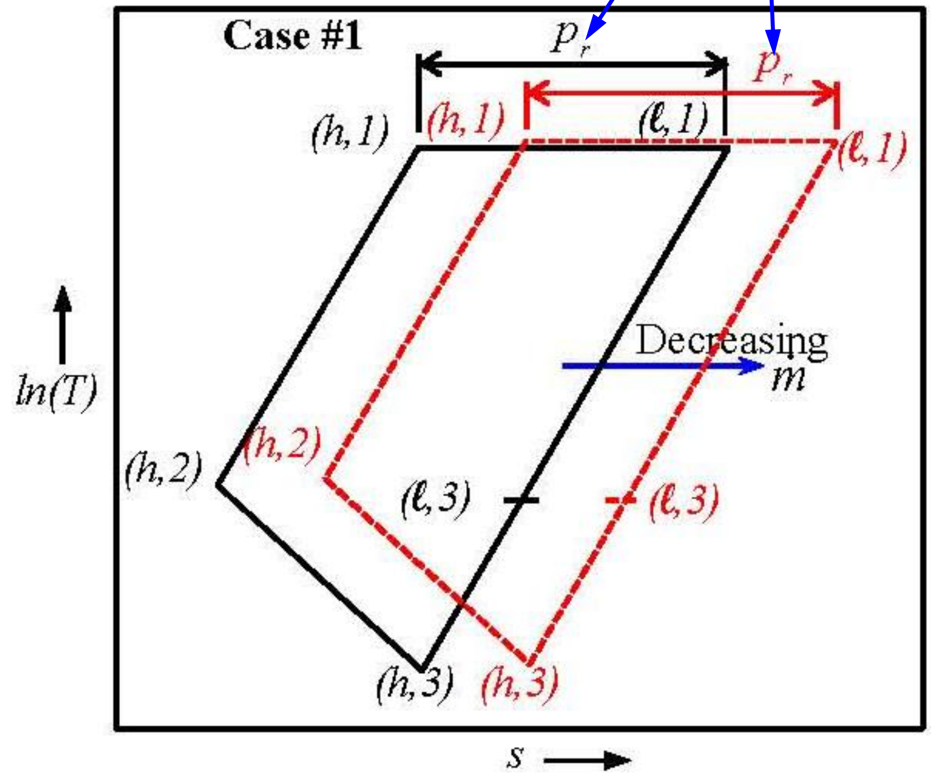
- ***Essentially constant pressure ratio* and,**
- ***Essentially constant Carnot efficiency***

For a given gas charge

Floating Pressure Process - System Optimization (Cont.)

The compressor and expander establish an *essentially constant pressure ratio and constant system Carnot efficiency*

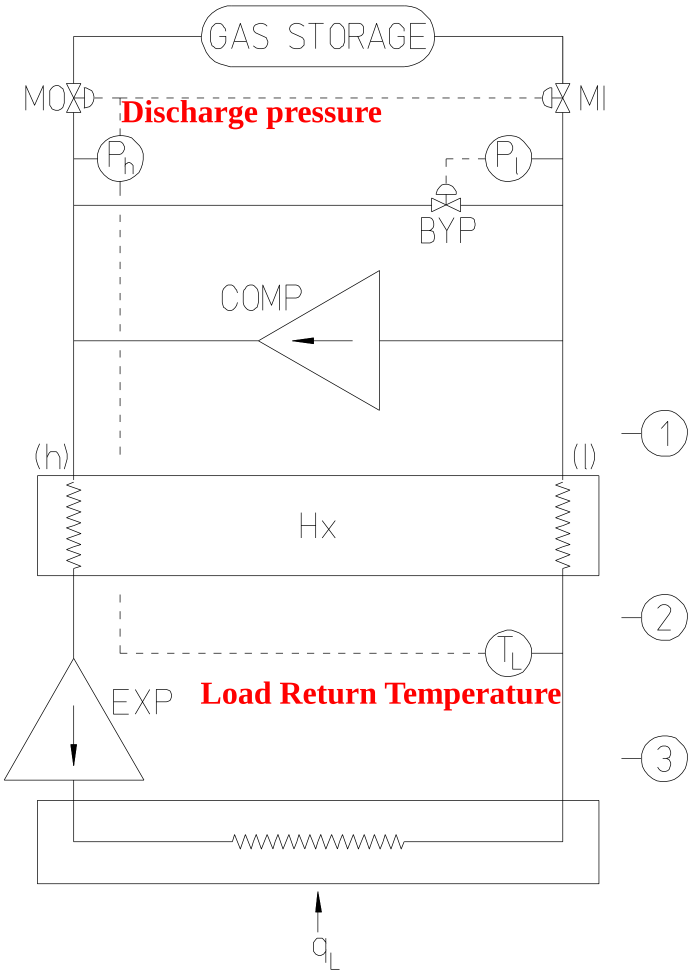




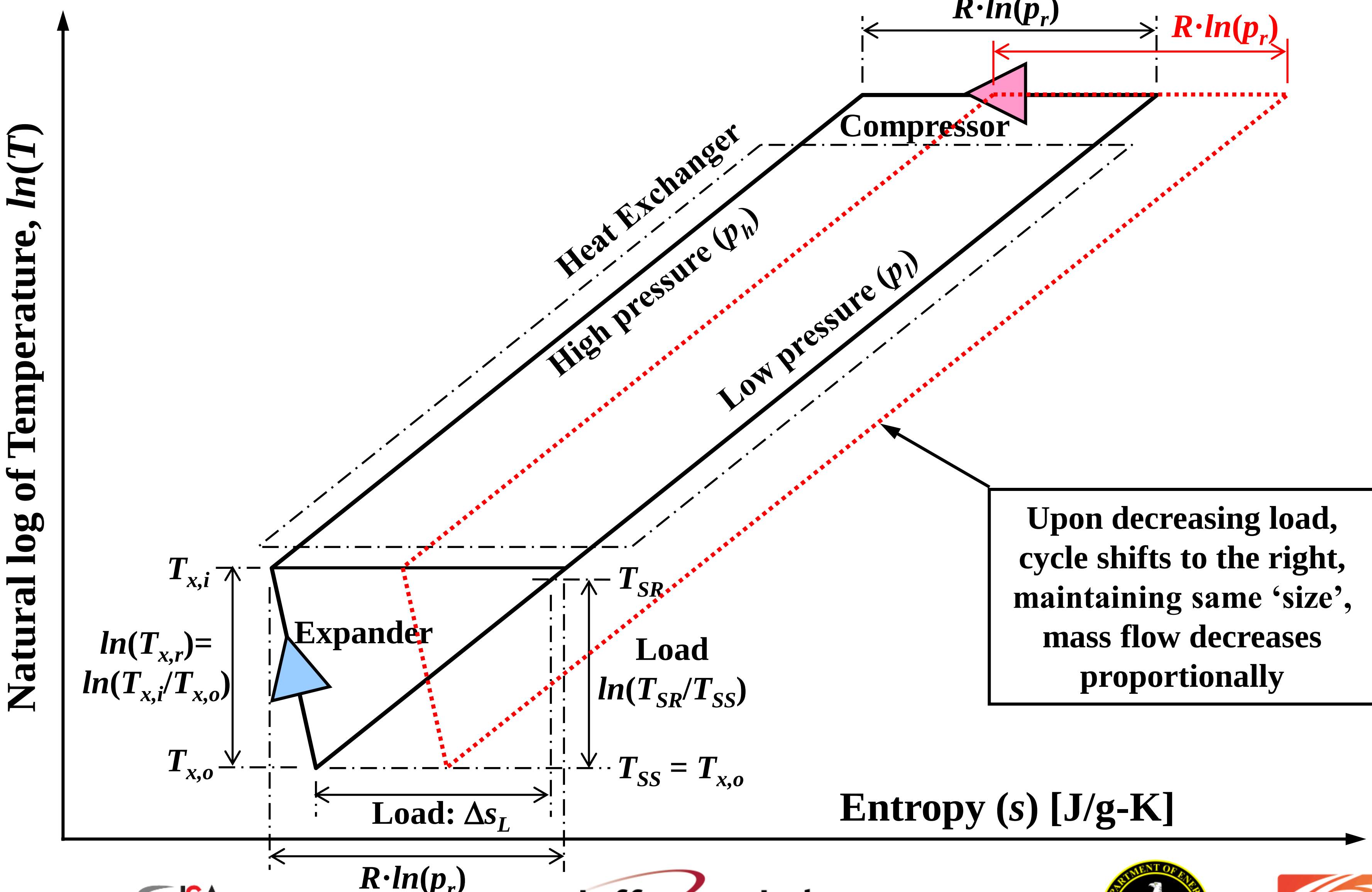
General Arrangement for Floating Pressure Process Cycle (*patent pending*)

Floating Pressure Process - System Optimization (Cont.)

- ***<u>Gas management valves establish how to respond to a given load</u>*, i.e.,**
  - **Compressor bypass (BYP)**
    - ***<u>Does not open</u>* except to prevent compressor suction from going below some minimum (usually ~1 atm)**
  - **Mass-Out (MO)**
    - **Discharges mass from compressor discharge to gas storage, decreasing p_h**
  - **Mass-In (MI)**
    - **Brings mass from gas storage to compressor suction, increasing p_h**
  - **Off-set between MO & MI (to prevent competition)**
  - **Discharge pressure (p_h) is linearly related to difference between actual (T_L) and desired load return temperature.**
    - **i.e., if T_L increases, then p_h increases**



Floating Pressure Process - System Optimization (Cont.)



Floating Pressure Process - System Optimization (Cont.)

Observations (TS diagram):

- **Y-axis is the natural logarithm of temperature**
- **Between any two arbitrary points '1' and '2',**

$$\Delta s = (s_2 - s_1) = C_p \cdot \{\ell n(T_2 / T_1) - \phi \cdot \ell n(p_2 / p_1)\}$$

$$\Delta s = C_p \cdot \{\ell n(T_r) - \phi \cdot \ell n(p_r)\}$$

- **So, at constant temperature (isotherms)**

$$\Delta s = -\phi \cdot C_p \cdot \ell n(p_r)$$

- **At constant pressure (isobars),**

$$\Delta s = C_p \cdot \ell n(T_r)$$

- **Slope of isobars is equal the specific heat at constant pressure (C_p)**

Floating Pressure Process - System Optimization (Cont.)

As the "Claude Cycle" is essentially a *constant pressure process*

and, the "Sterling Cycle" is a *constant volume process*

the "Floating Pressure Cycle" is a *constant pressure ratio* process

$$p_r \equiv \frac{p_{h,2}}{p_{l,1}} = \left(\frac{\eta_v \cdot Q_C}{\kappa_x}\right) \cdot \left(\frac{1}{\phi \cdot C_p}\right) \cdot \frac{\sqrt{T_{h,2}}}{T_{l,1}} \cong \text{Constant}$$

$$\eta_{carnot} = \frac{\mathrm{E}_L}{\dot{W}_C} = \frac{\Delta\varepsilon_L}{w_C} \cong \text{Constant}$$

That maintains *essentially constant Carnot efficiency*
over a very wide operating range

(100% to ~ 40% of maximum capacity in practical systems)

Capacity Modulation

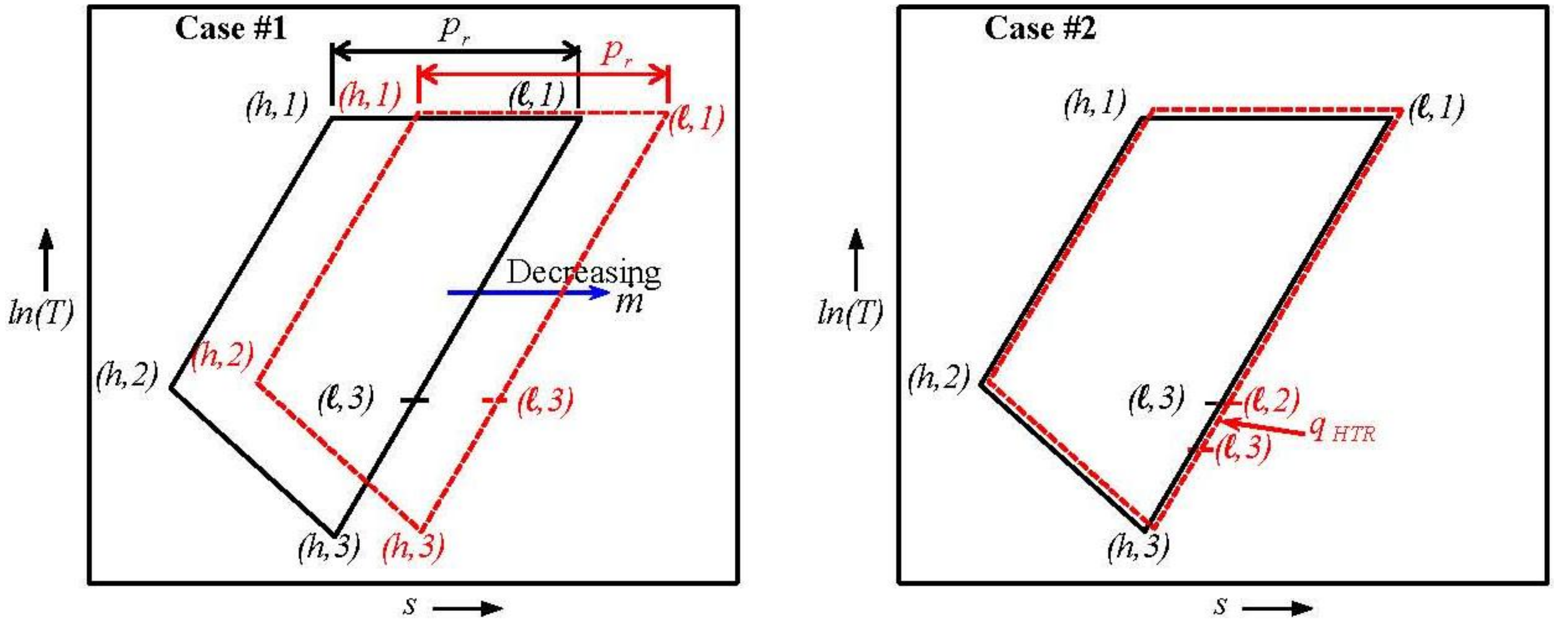
Methods to Control Shield Refrigerator Capacity

Case #	Load Adjustment Mechanism	Constraint
1	Compressor Discharge Pressure (p_h)	Zero Compressor Bypass ($\dot{m}_{BYP}$); i.e., p_r = constant
2	Load Heater (q_{HTR})	Compressor Suction Pressure (p_l)
3	Expander Inlet Valve ($\Delta p_{x,i}$)	Compressor Suction Pressure (p_l)
4	Compressor Discharge Pressure (p_h)	Compressor Suction Pressure (p_l)
5	Expander Inlet Valve ($\Delta p_{x,i}$)	Zero Compressor Bypass ($\dot{m}_{BYP}$)
6	Expander Bypass ($\dot{m}_{x,BYP}$)	Compressor Suction Pressure (p_l)

Note: Case #1 is the Floating Pressure Process. The others are traditional methods.

Capacity Modulation(Cont.)

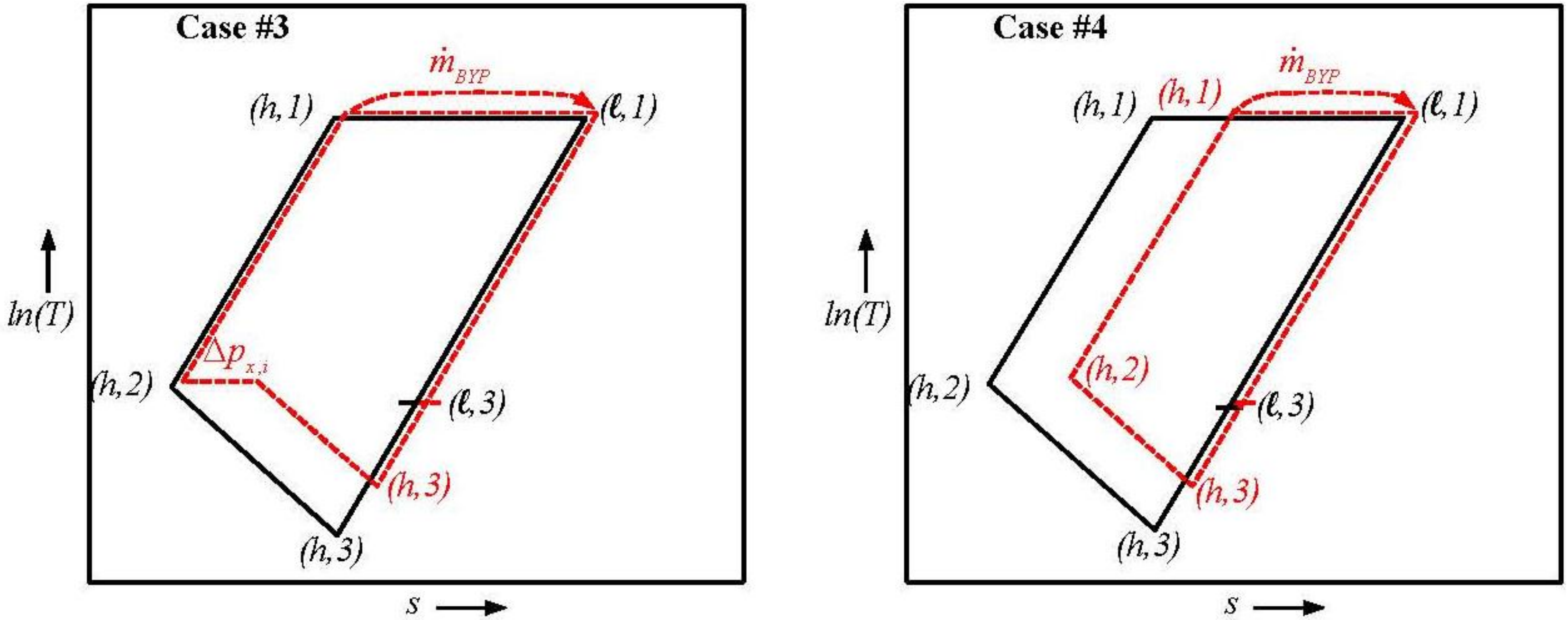
TS Diagram of Case #1 & #2



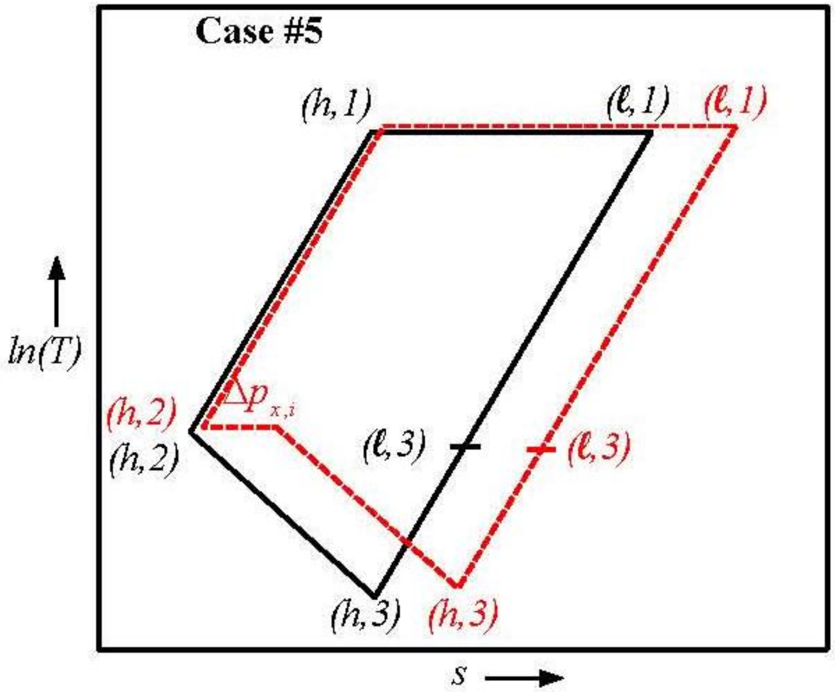
Note: Case #1 is the Floating Pressure Process

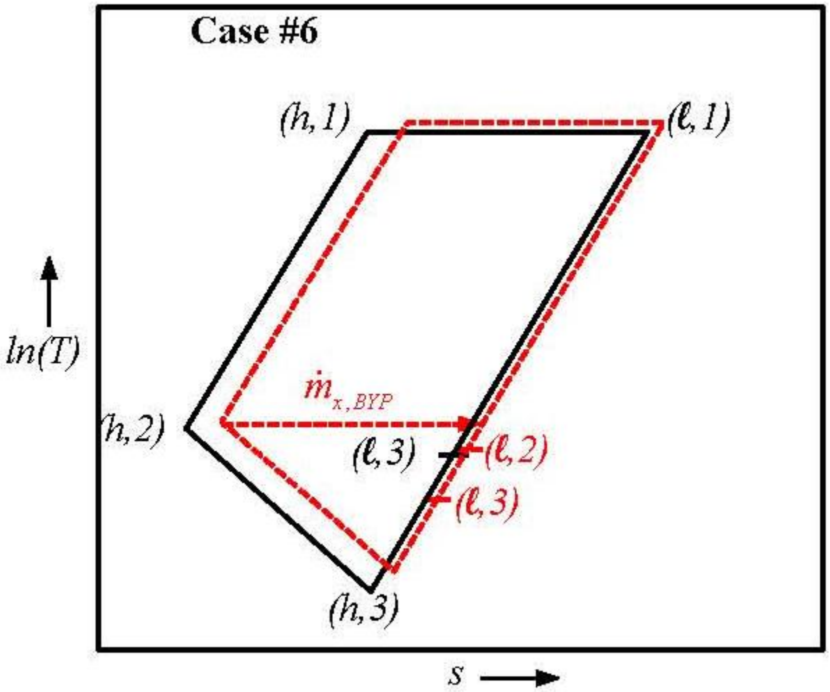
Capacity Modulation(Cont.)

TS Diagram of Cases #3 & #4



TS Diagram of Cases #5 & #6



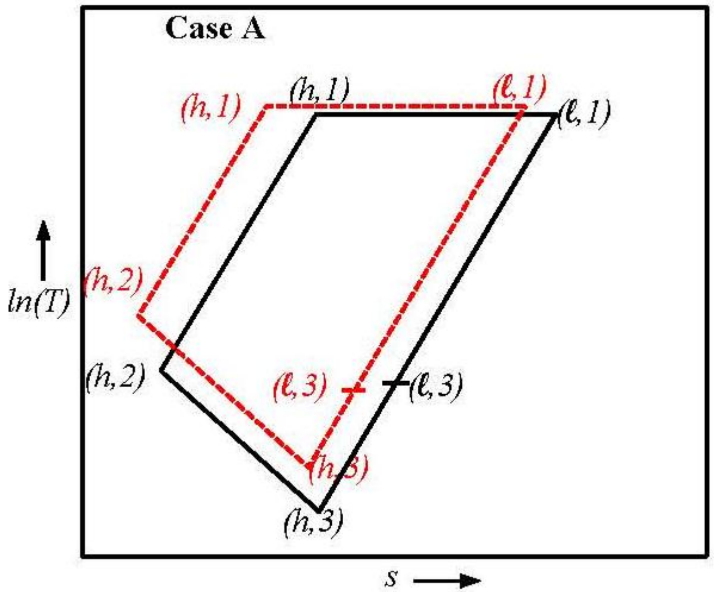


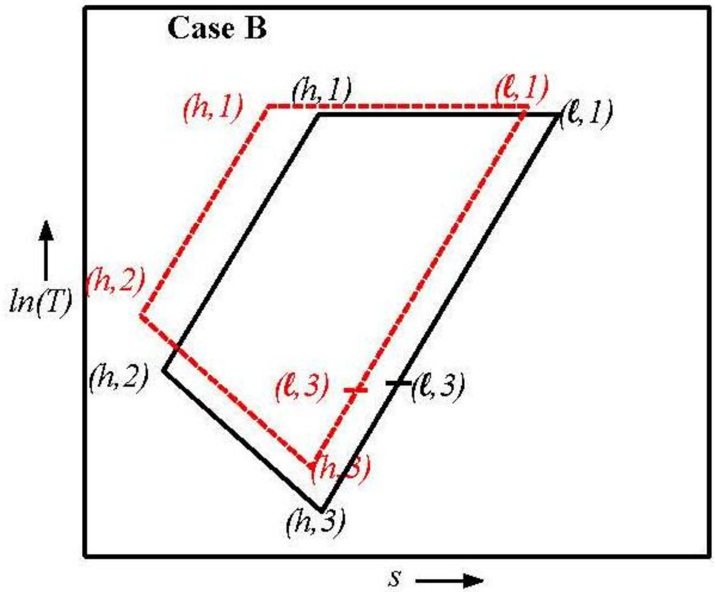
Variations in Equipment Parameters

Using the Floating Pressure Process, for selected equipment parameters that are *less than their design value*, how does the cycle move from the design condition?

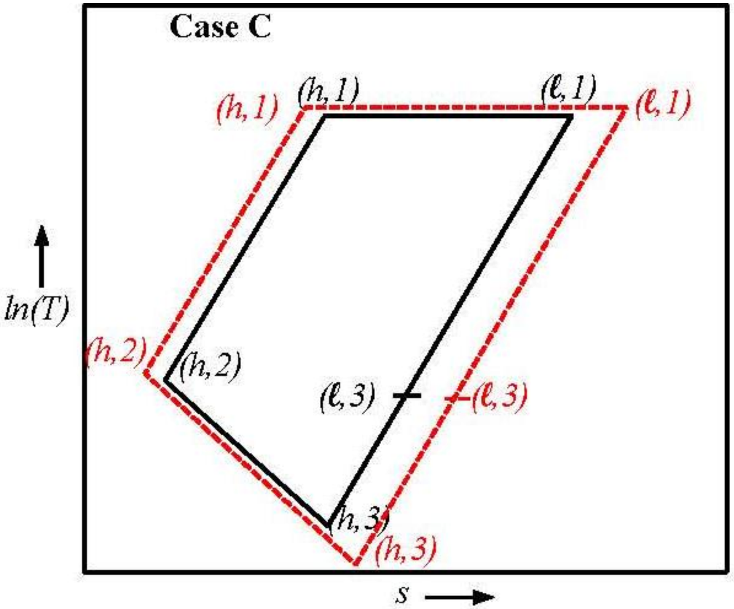
Case #	Selected Equipment Parameter Less Than Design Value	Pressure Ratio	Mass Flow
A	HX Size	Increase	Increase
B	Expander Efficiency	Increase	Increase
C	Expander Flow Coefficient	Increase	Decrease
D	Compressor Volumetric Efficiency	Decrease	Increase

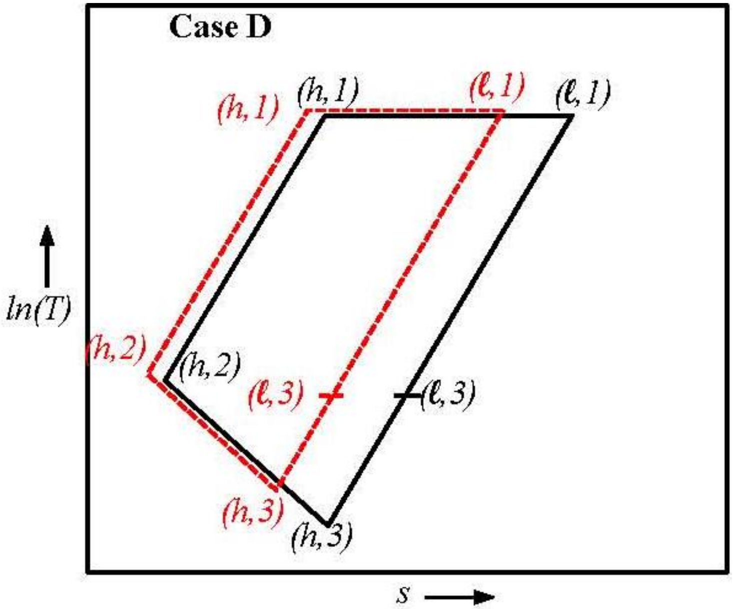
TS Diagram of Cases A & B





TS Diagram of Cases C & D





Floating Pressure Process - System Optimization (Cont.)

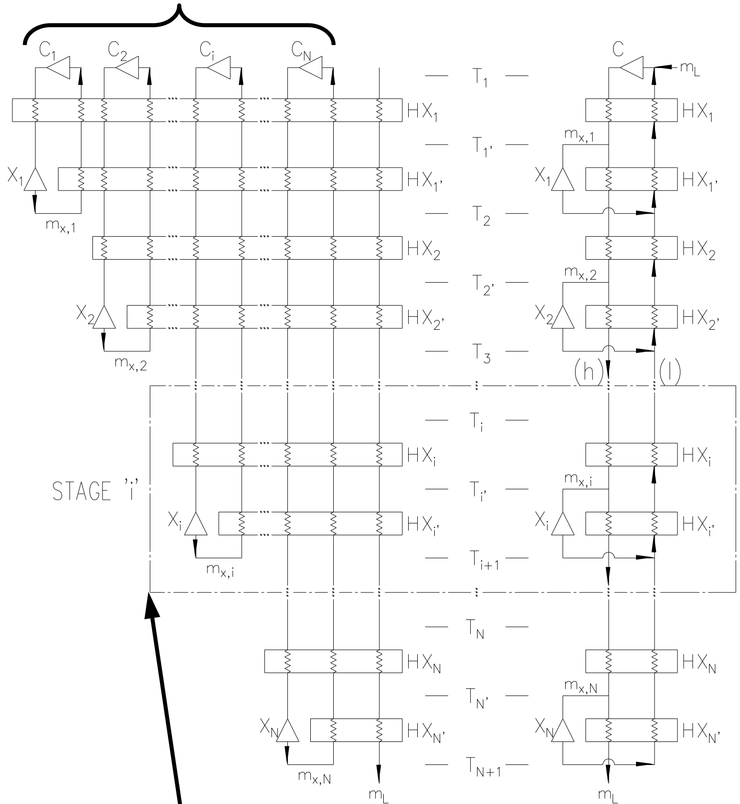
If, instead of using the Floating Pressure Process (as discussed in Case #1), one of the load adjustment mechanisms in Cases #2 to #6 were implemented in attempting to bring the off-design condition back to the TS design condition one of two results would occur:

- **For the selected equipment parameter which is *less than the design value*, the shield load *cannot be met* and system Carnot efficiency is reduced.**
- **For the selected equipment parameter which is *greater than the design*, the shield load *can be met* (matched) but at a system Carnot *efficiency less than is possible***

Floating Pressure Process - System Optimization (Cont.)

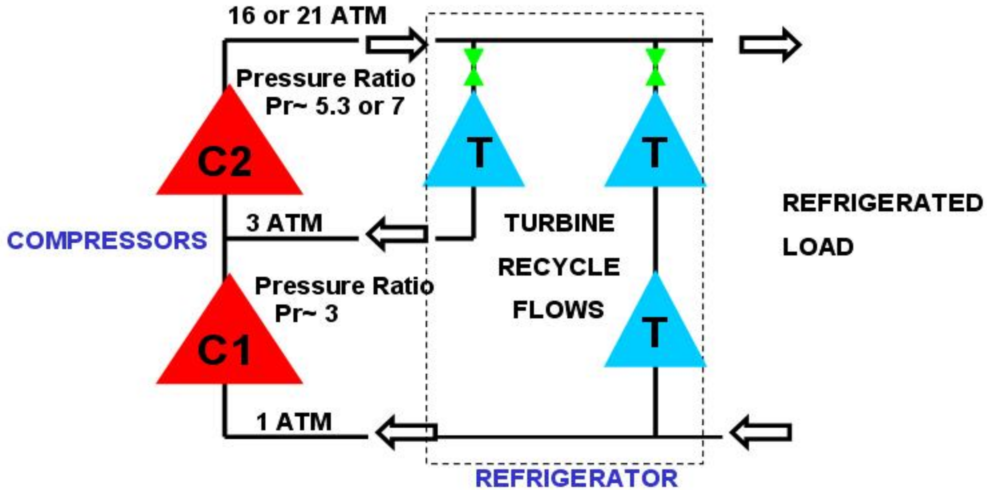
- **So, how does this apply to helium liquefiers and refrigerators?**
- **Recall that *each expansion stage is basically the cycle described in the Floating Pressure Process***
- **For liquefiers and mix-mode systems, 60 to 90% of the total system flow is through the turbines (providing the cooling)**
- **Also, recall that ~2/3rd of the total system losses are in the compressor system; so we must consider what is means to properly match the compressor and cold box system**

60 to 90% of the total system flow



Each expansion stage is like the cycle in the Floating Pressure Process

Ganni Cycle - System Optimization (Cont.)



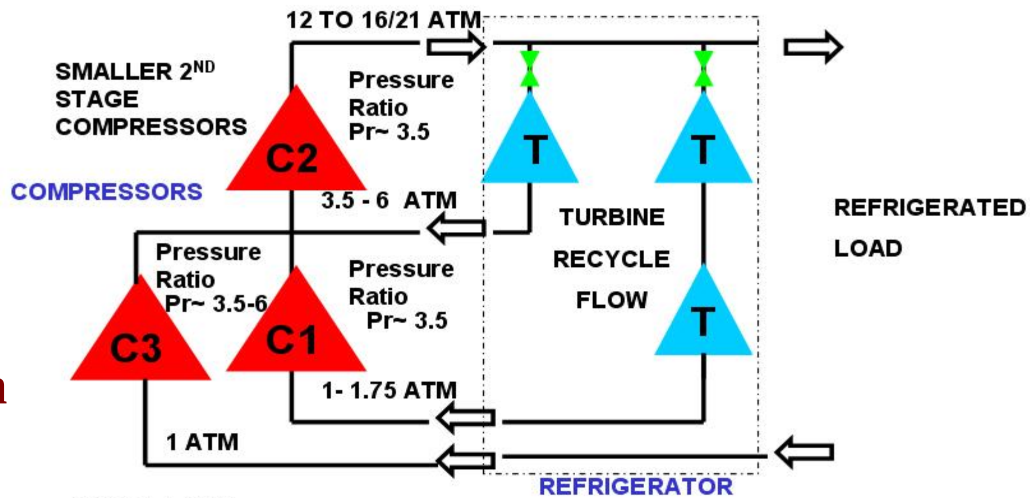
STANDARD INDUSTRIAL
HELIUM REFRIGERATOR SYSTEM

Traditional Helium Cycles

Poor pressure ratio matching.
Resulting in large losses in 2nd stage compressors (which require the largest fraction of the electrical input power).

Ganni Cycle

Good (optimum) pressure ratio matching.
Resulting in low losses for both stages.
Flow from load is separated from turbine flow (since it is a smaller fraction of the total flow).



GANNI CYCLE
HELIUM REFRIGERATOR SYSTEM

Floating Pressure Process - System Optimization (Cont.)

<u>Summary</u> **The Ganni cycle – Floating Pressure Process:**

1. Provides a basis for an optimal design at maximum load, turn-down cases and mixed modes, addressing the compressor system as the major input power loss contributor

2. Provides a solution to implement on as-built systems (existing or new) to improve system efficiency, reliability, availability and load stability under actual loads and help to improve the experimental envelop

3. Invalidates the philosophy that operating as-built systems at the TS design conditions is optimal by properly identifying the fundamental process system parameters for control

4. Is a constant pressure ratio process cycle (as the Sterling Cycle is a constant volume process and the Claude Cycle is a constant pressure level process) and maintains the compressor efficiency for varying loads

Floating Pressure Process - System Optimization (Cont.)

5. Is a variable gas charge system, whose gas charge is automatically adjusted and thus the compressor input power, to satisfy the given load

6. Not contingent on precise instrumentation for successful operation. This is due to decoupling specific values of process variables from presumed system load capacities

7. Maintains a constant volume flow (and thus the velocity) at any point in the system and preserves the expander efficiency and the oil removal effectiveness during the turn-down cases

8. ***Has been <u>licensed</u> by <u>JLab</u> to <u>Linde Cryogenics</u>, Division of Linde Process Plants, Inc. and Linde Kryotechnik AG for world wide commercialization***

Some Historical Reasons given to stay status quo:

- **"We have done this before"** (if so…good!...we share a common desire to utilize natural resources wisely!)
- **Industry,**

  An increase in system efficiency comes with,
  - **"Increase in capital cost"**
  - **"Reduced availability"**
  - **"High risk to the basic program"**
- **Users,**
  - **"T-S design is the optimum, force the system close to it"**
  - **"You should not change system operation from the basic design and/or the operation method"**
  - **"Cryogenics is not the experiment"**
  - **"The cryo system is running fine. Don't change it"**
  - **"Scale the new system from an existing one"**
  - **"Requires re-training of the operators"**

And many, many more !!!

Licensing Agreement

Jlab has licensed the Ganni Floating Pressure Helium Process Cycle technology to Linde Cryogenics,
Division of Linde Process Plants, Inc. and Linde Kryotechnik AG
for world wide commercialization.

6. Liquid Nitrogen Pre-cooling

Liquid nitrogen (LN_2) pre-cooling is widely used in helium refrigeration systems. We would like to…

- **Discuss the advantages and drawbacks of using LN_2 pre-cooling**
- **Discuss and explain various LN_2 pre-cooling schemes used**
- **Provide some simplified analyses**

Objective: provide a rational viewpoint in the evaluation of using LN_2 pre-cooling and the methods to both minimize LN_2 consumption and required equipment capital cost while reducing the dangers of LN_2 freezing and achieving an optimized design.

Liquid Nitrogen Pre-cooling (Cont.)

What Does LN_2 Pre-Cooling Do?

In steady state operation of helium refrigeration systems, LN_2 pre-cooling provides :

- **The liquefaction load flow unbalance, i.e., to cool the net helium make-up gas from 300 to 80K, ~3 g/s of LN_2 per 1 g/s of makeup helium is required**
- **The 300-80K heat exchanger (HX-1) cooling curve losses due to stream temperature difference (ΔT's) associated with the recycled compressor flow**

Liquid Nitrogen Pre-cooling (Cont.)

What Does LN_2 Pre-Cooling Do? (Continued)

- Provides the refrigeration for these needs at an operating efficiency close to that of an LN_2 plant

- Used to increase system capacity by providing refrigeration that would otherwise be required from the existing turbines, allowing them to operate at a lower temperature, so that the refrigeration they are providing is at a higher Carnot (exergetic) value.

Liquid Nitrogen Pre-cooling (Cont.)

Advantages of using LN_2:

- A lower capital investment for a given refrigeration capacity
  — increases the LHe production or the refrigeration capacity by 1.5 times or more
- A smaller cold box and compressor size for a given capacity requires a smaller building. Space is also required for a LN2 dewar - located outside
- Provides a thermal anchor point for the 80K adsorber beds and the refrigeration capacity to re-cool the beds after regeneration
- Stable operation over a larger operating range and a larger refrigeration turndown capability
- Has fewer rotating parts and lower maintenance costs for a given capacity
- Able to keep the load temperature at 80K during partial maintenance of the cold box sub- systems (i.e., turbine replacement, etc.)
- Impurities in the helium stream are frozen in the ‘warm’ HX and thereby protecting the lower temperature turbines from contamination and erosion damage.
- Extremely useful to handle the cool down of especially large loads. In general, approximately ~80% of the LHe temperature cool down load is from 300K to 80K

Liquid Nitrogen Pre-cooling (Cont.)

Disadvantages of using LN_2:

- Operating costs are typically greater (depends upon local electrical power cost vs. LN_2 cost)
- Presence of different fluids in the system, thereby presenting the potential for cross fluid leaks than can result in plugging and capacity loss (due to the increased pressure drop).
- Requires the coordination of LN_2 deliveries (although, this can be automated by the supplier)
- Weather constrains (winter road conditions and the summer power restrictions) may affect the LN_2 deliveries
- LN_2 operations in enclosures are generally more hazardous than LHe operations in enclosures and requires additional oxygen deficiency hazard (ODH) monitoring.

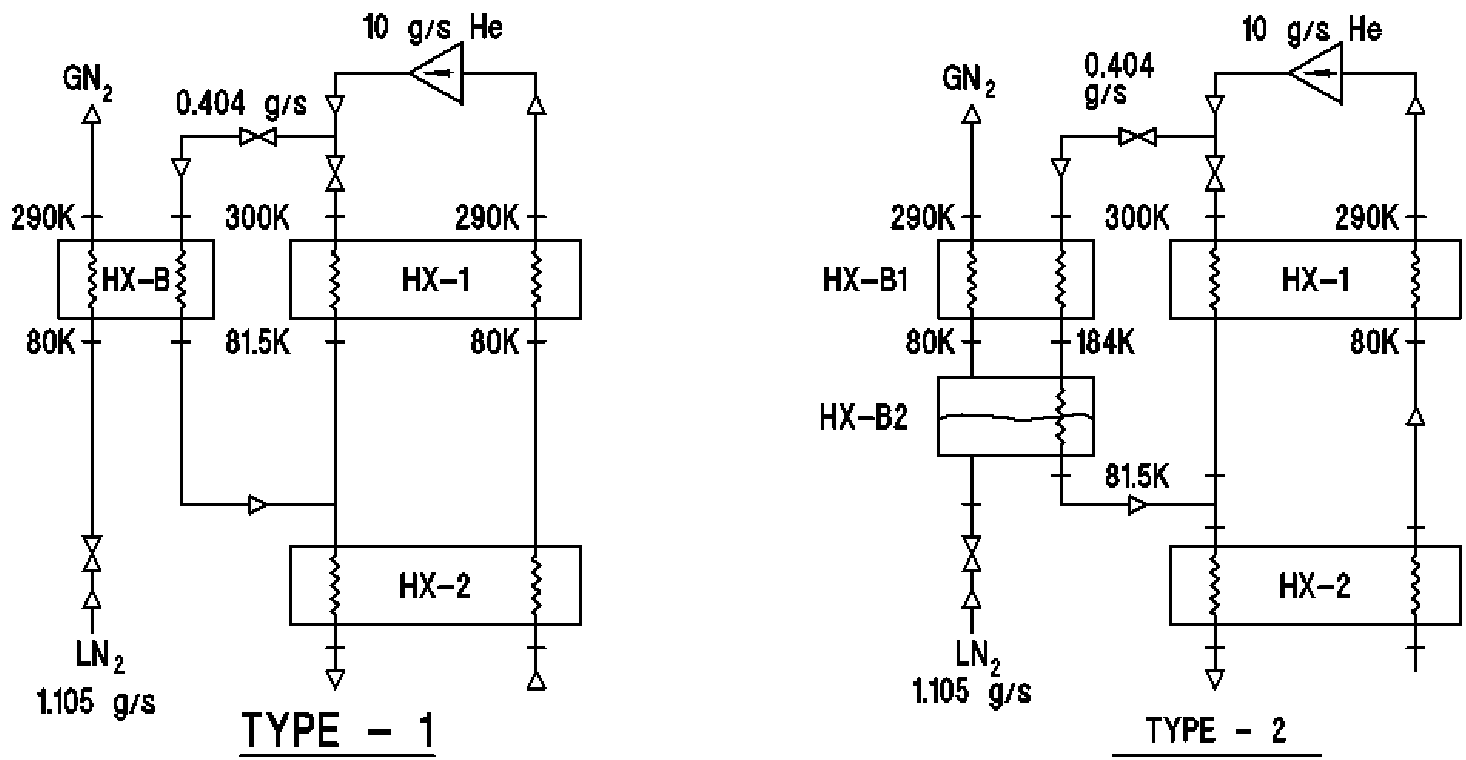
Liquid Nitrogen Pre-cooling (Cont.)

LN_2 Pre-cooling Types

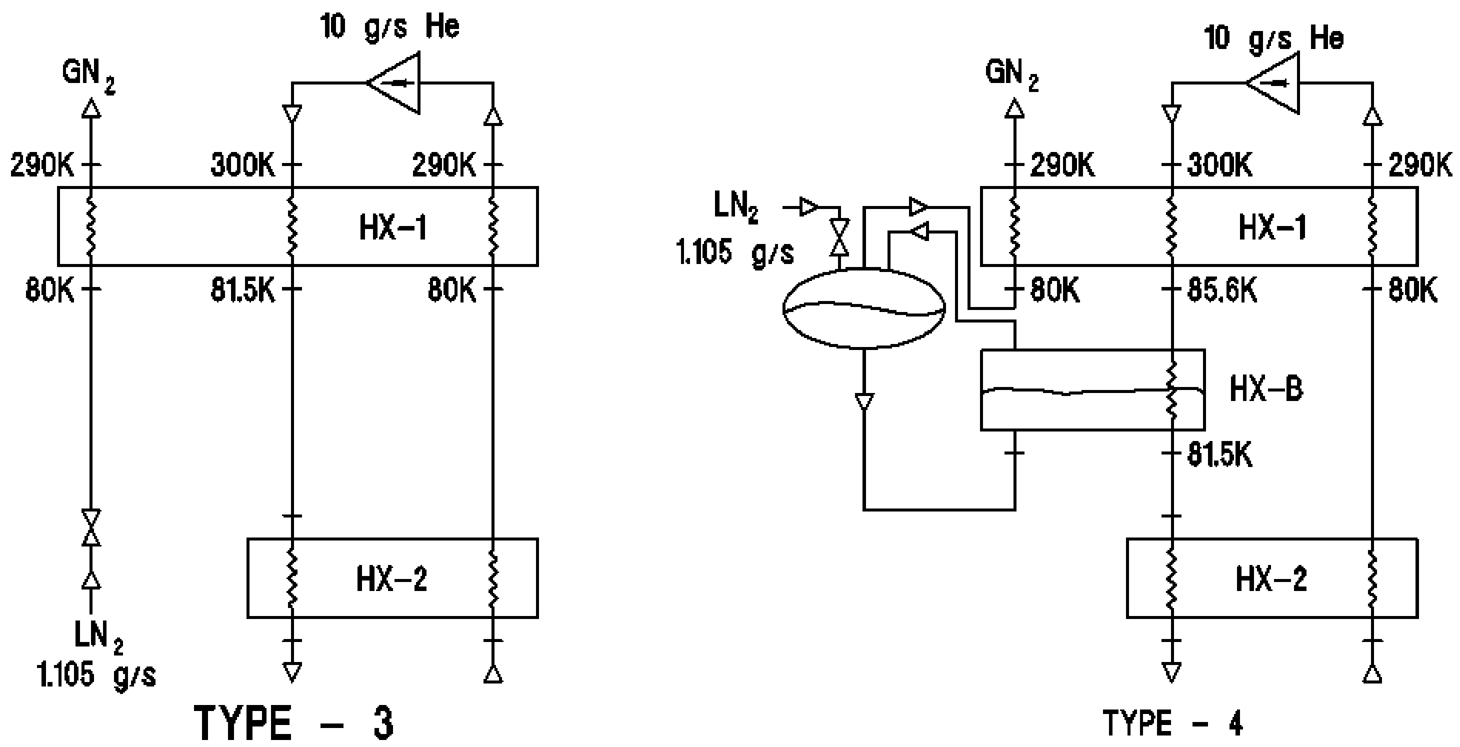
- **Different types of HX arrangements used for LN_2 pre-cooling have a major influence on the size of the HX's and the required LN_2 consumption.**

- **Six commonly used types (shown as Type-1 through Type-6 in following figures) are examined for comparison.**

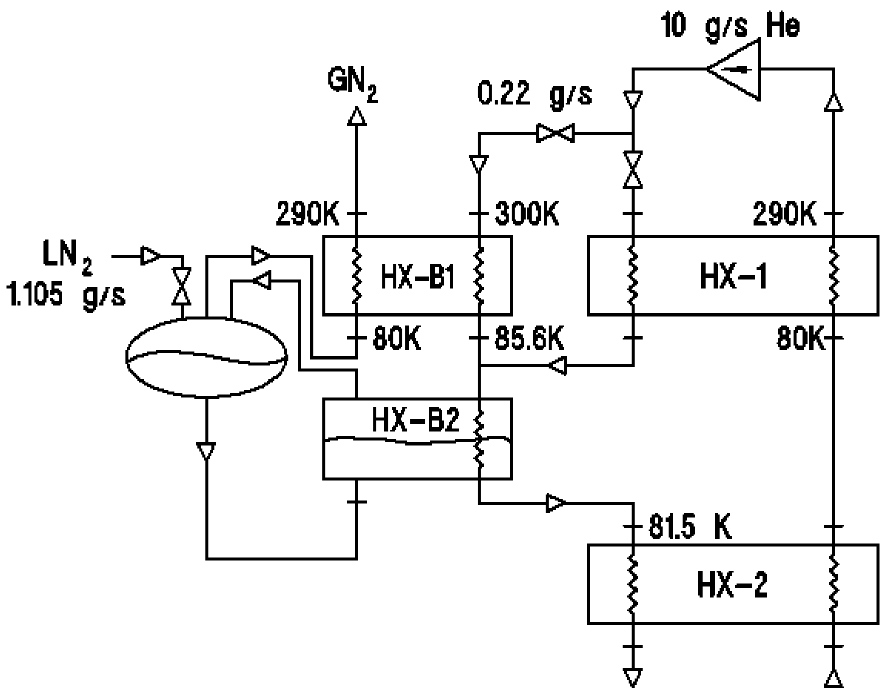
Liquid Nitrogen Pre-cooling (Cont.)



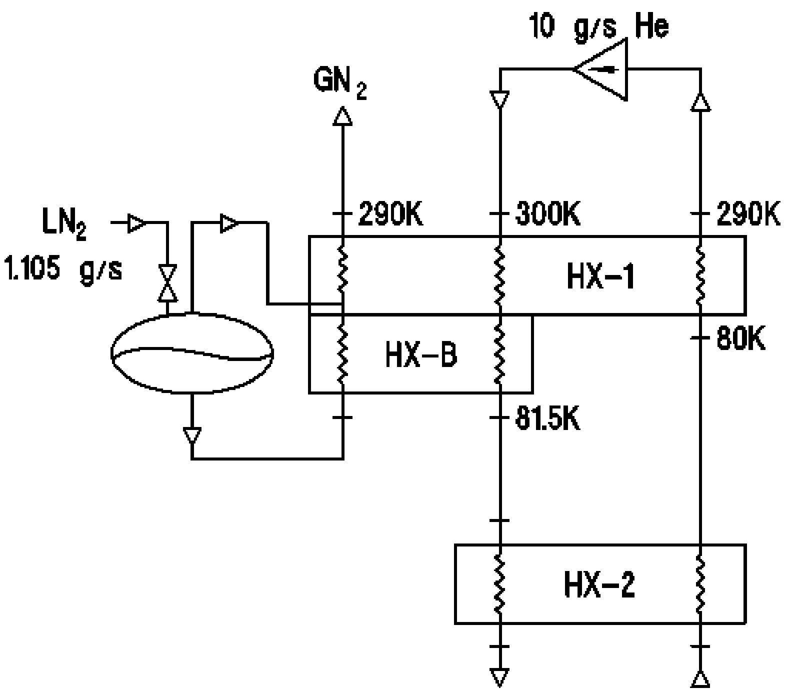
Liquid Nitrogen Pre-cooling (Cont.)



Liquid Nitrogen Pre-cooling (Cont.)



TYPE – 5



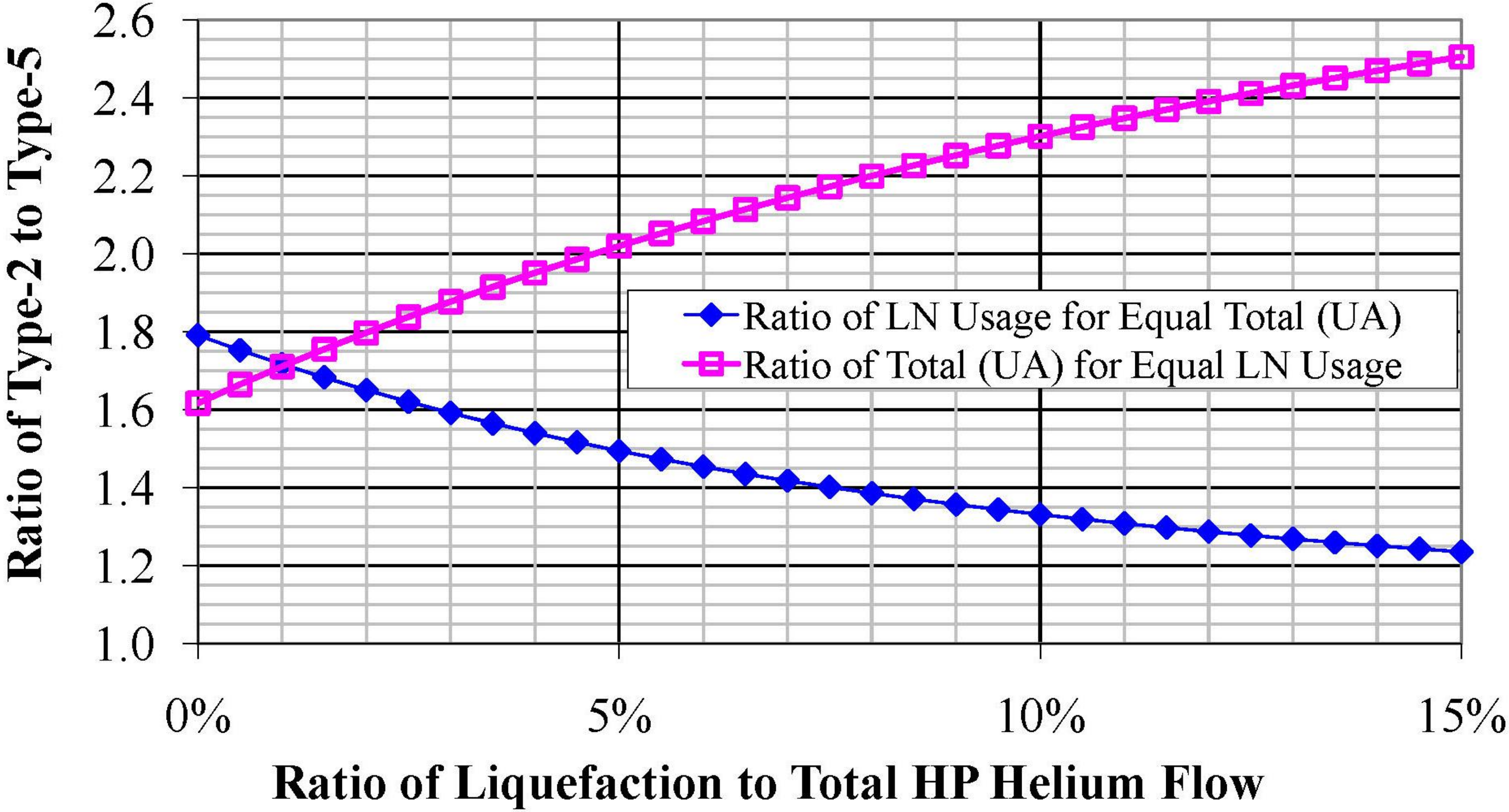
TYPE – 6

Liquid Nitrogen Pre-cooling (Cont.)

Comparison of HX Types:

Type	HX #	Duty [W]	ΔT_{LM} [K]	C_{min} [W/K]	C_{max} [W/K]	C_R [none]	(UA) [W/K]	NTU [none]	ε [%]
Type-1	HX-1	10888	4.48	49.83	51.85	0.961	2430	48.8	99.3
	HX-B	458	4.48	2.10	2.18	0.961	102	48.8	99.3
Type-2	HX-1	10888	4.48	49.83	51.85	0.961	2430	48.8	99.3
	HX-B1	243	40.14	1.16	2.10	0.552	6	5.2	95.5
	HX-B2	215	24.18	2.10	∞	0.000	9	4.2	98.6
Type-3	HX-1	11347	4.48	51.93	54.03	0.961	2532	48.8	99.3
Type-4	HX-1	11134	7.59	51.93	53.02	0.979	1467	28.3	97.5
	HX-B	213	3.11	51.93	∞	0.000	68	1.3	73.2
Type-5	HX-1	10889	7.59	50.79	51.85	0.979	1435	28.3	97.5
	HX-B1	245	7.59	1.14	1.17	0.979	32	28.3	97.5
	HX-B2	213	3.11	51.93	∞	0.000	68	1.3	73.2
Type-6 (Same as Type-4)	HX-1	11134	7.59	51.93	53.02	0.979	1467	28.3	97.5
	HX-B	213	3.11	51.93	∞	0.000	68	1.3	73.2

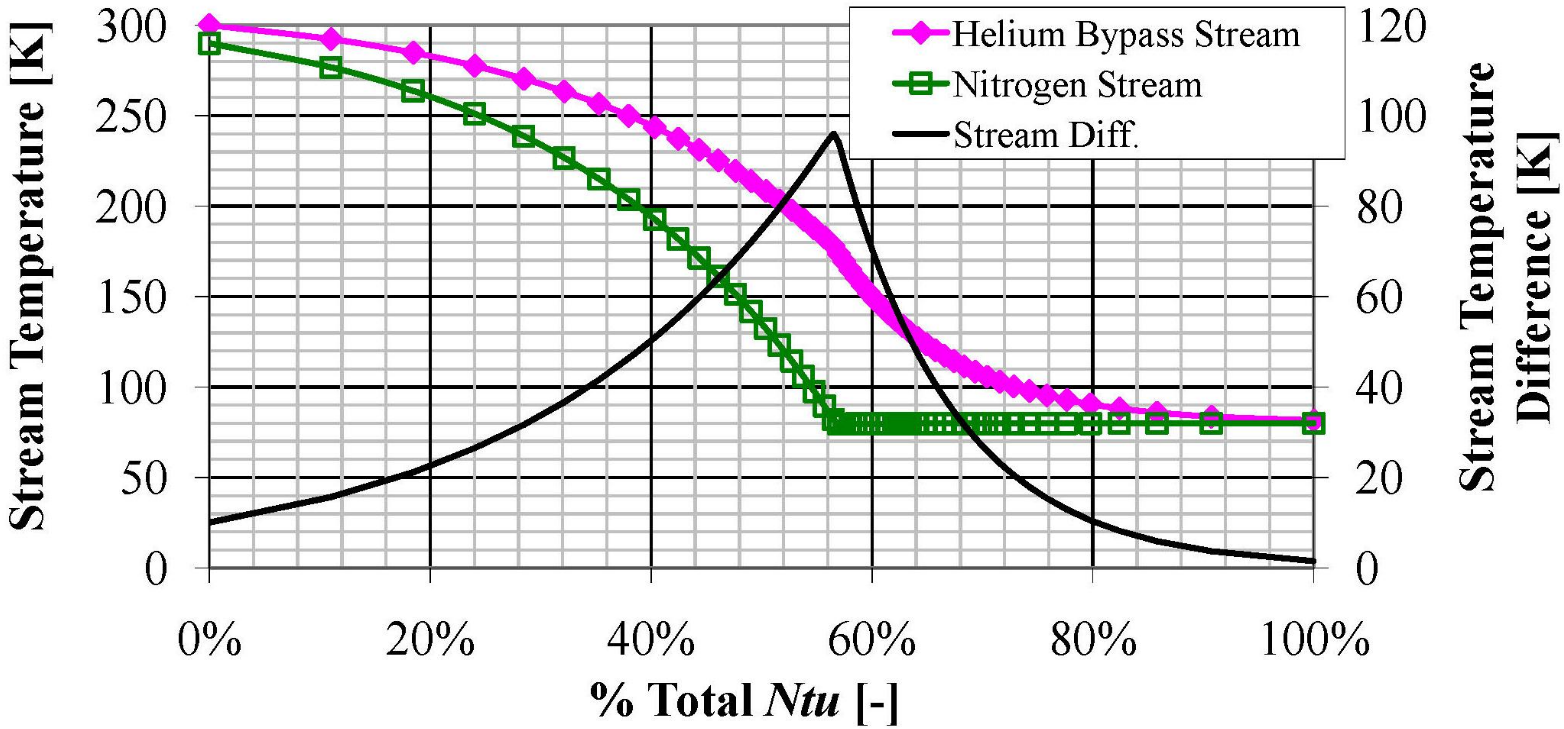
Liquid Nitrogen Pre-cooling (Cont.)



Liquid Nitrogen Pre-cooling (Cont.)

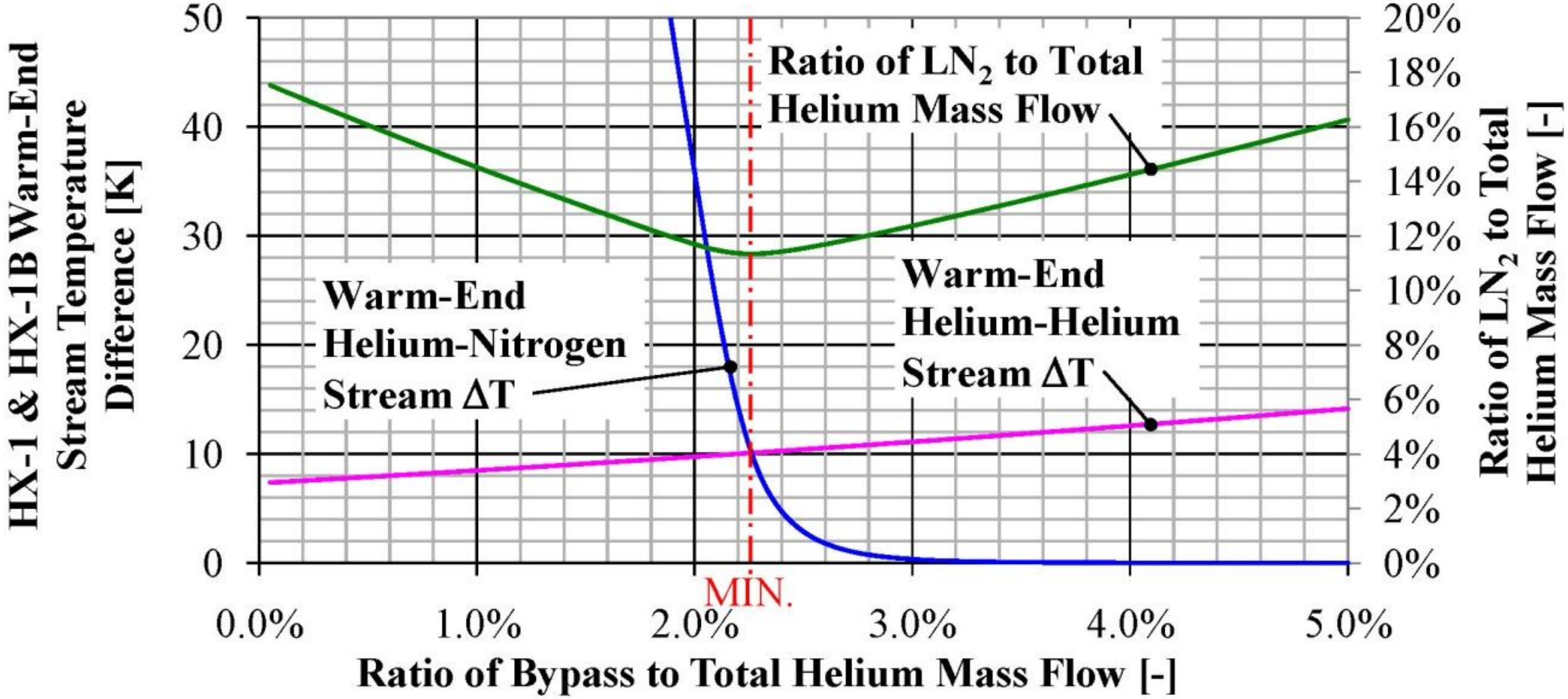
- **The cooling curve for the sensible and latent sections for Types 1-3 is very poor!**

  ***Note*: this will have a very detrimental effect on the temperature distribution in the helium-helium layers, even if there is proper adiabatic layering!**



Liquid Nitrogen Pre-cooling (Cont.)

The Type 5 arrangement offers a distinct advantage over Type 4 that is not necessarily apparent at first glance

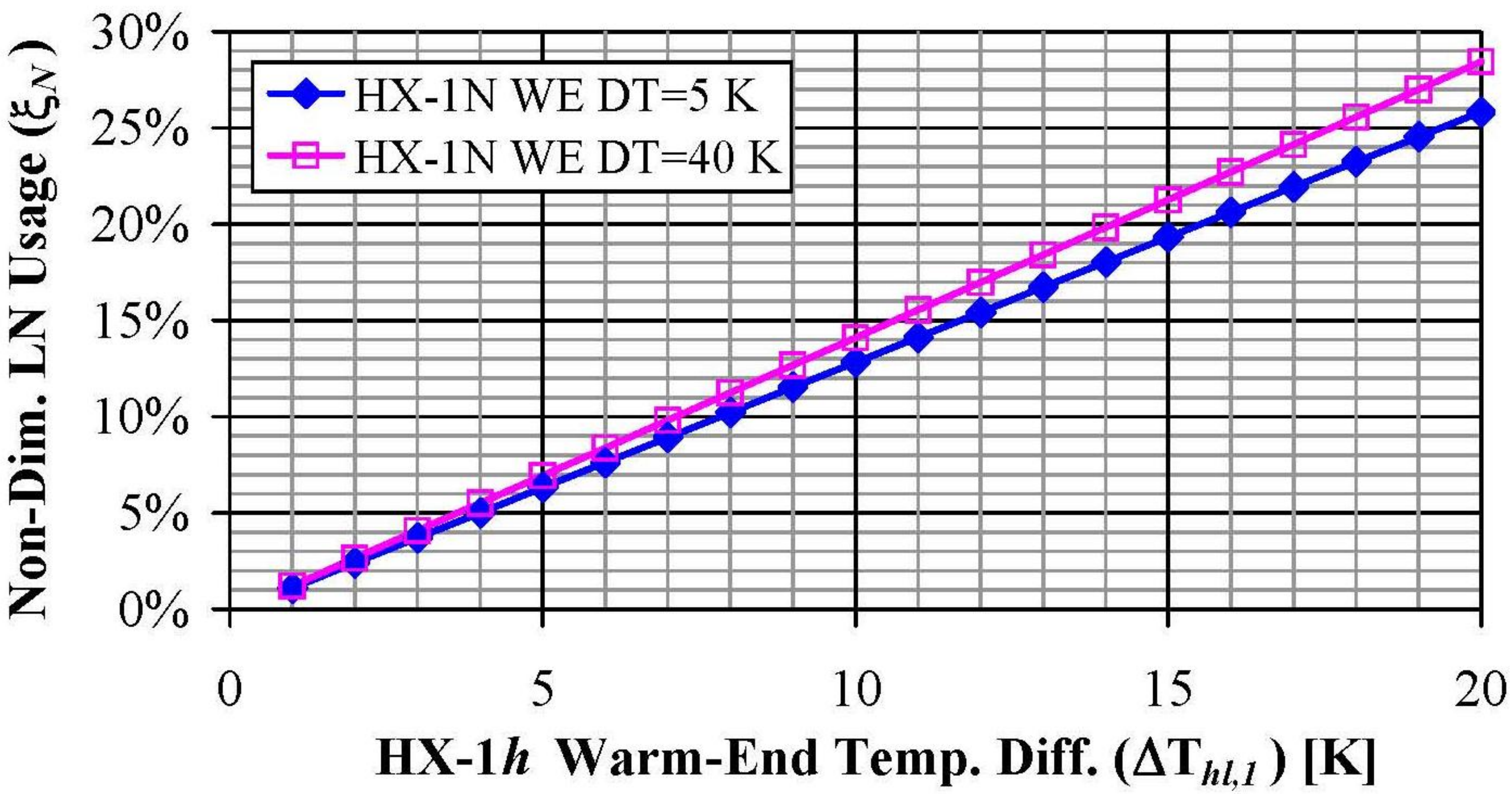


Although for a particular case, the above depicts the behavior of a Type 5 arrangement as the high pressure bypass valve position is varied (affecting the helium mass flow through HX-1 and HX-1N)

Liquid Nitrogen Pre-cooling (Cont.)

HX-1 vs. LN_2 Consumption

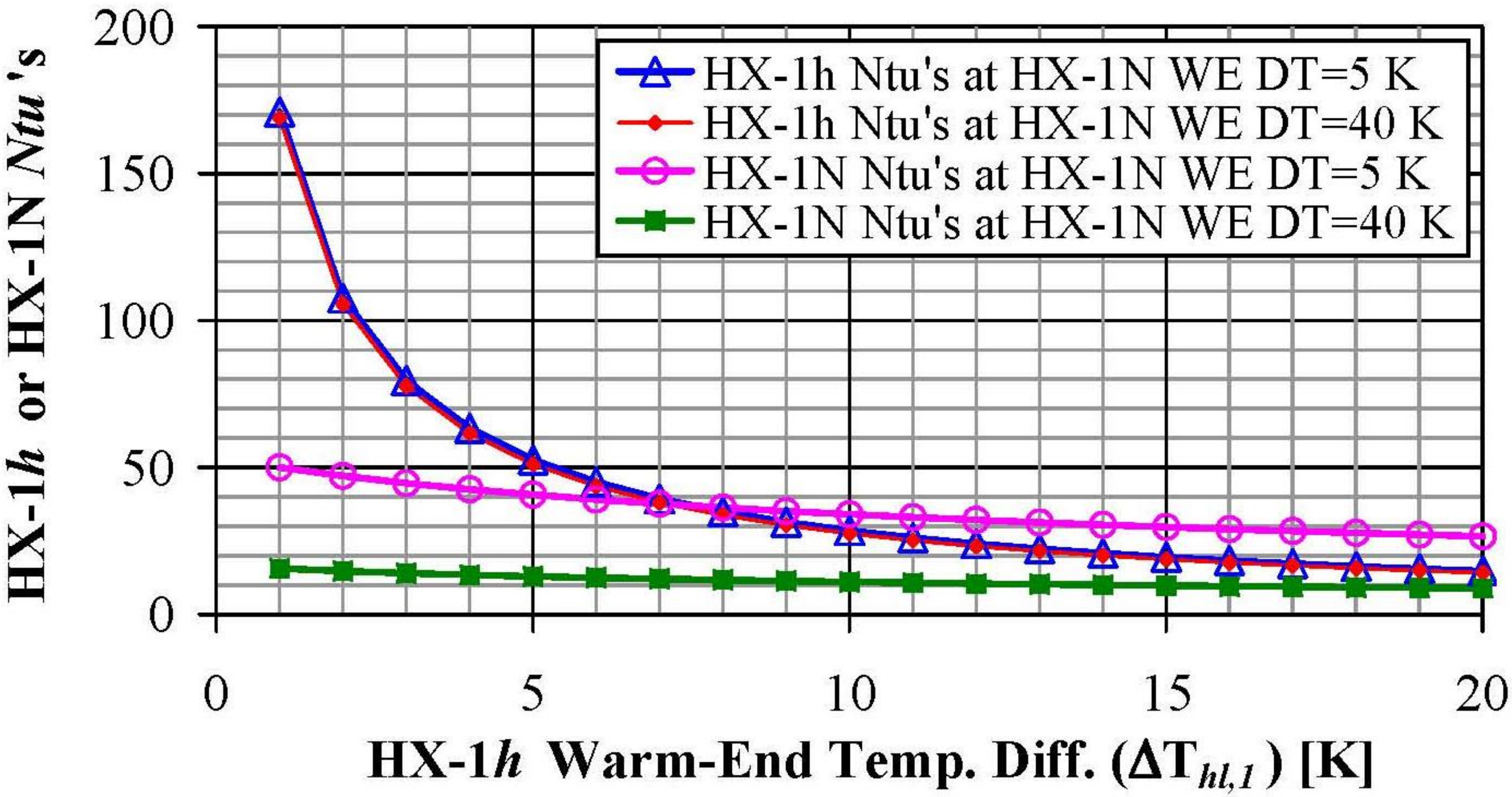
LN_2 consumption is insensitive to extreme warm-end helium-nitrogen stream temperature differences



Liquid Nitrogen Pre-cooling (Cont.)

HX-1 vs. LN_2 Consumption

LN_2 consumption is insensitive to the size of the sensible section of the helium-nitrogen HX

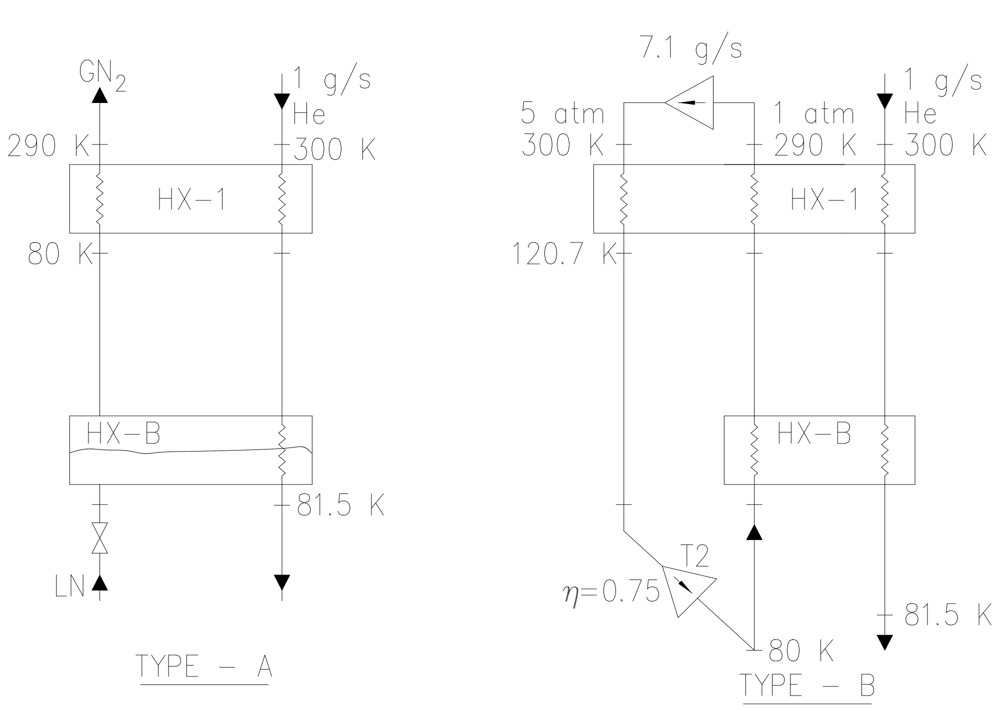


Liquid Nitrogen Pre-cooling (Cont.)

Simplified Operating Cost Comparison

- To compare LN_2 pre-cooling to turbine cooling based upon an equal operating cost basis, refer to the three methods depicted in the figures Type-A, Type-B and Type-C.
- Each accomplishes the cooling of 1 g/s of helium from 300-80K.
- The 1 g/s represents the additional total cooling flow load for a combination of liquefaction load and HX losses (due to finite stream temperature differences and heat leaks)

Liquid Nitrogen Pre-cooling (Cont.)



Liquid Nitrogen Pre-cooling (Cont.)

	LN_2 Cost Break-Even Point Analysis			
	Units	**TYPE-A**	**TYPE-B**	**TYPE-C**
Helium Liquefaction Flow	g/s	1.0	1.0	1.0
LN_2 Flow	g/s	2.7	—	—
	l/hr	12	—	—
Expander Efficiency(s)		—	0.7	0.7
Compressor Recycle Flow	g/s	—	6.5	4.5
Comp. Isothermal Eff.		—	0.5	0.5
Comp Power Input	kW	—	13	9
Given:				
LN_2 Cost	\$/liter	0.06	—	—
Electric Power	\$/kW-h	0.04	—	—
LCF		—	**1.38**	**2.00**

Liquid Nitrogen Pre-cooling (Cont.)

Simplified Operating Cost Comparison

- Based on the above example analysis of operating cost for assumed LN_2 and electric power costs, LN_2 cooling is 1.16 times more expensive compared to the single expander system and 1.72 times more expensive for the two expander system.

- So, for a system that uses 1000 liters/hr (i.e., 264 gal/hr or 225 g/s or $482,000/yr) of LN_2 over one year period, a savings of $66,000 would be made using Type-B and $200,000 for a Type-C turbine pre-cooling system rather than LN_2.

7. Helium Refrigeration Systems for Below 4.2 K Operations

Used for refrigeration at temperatures below the atmospheric pressure saturation temperature (4.22K)

These systems inherently appear as liquefaction loads to the main (4K) refrigeration system, which is *providing the refrigeration*.

The nominal 2K systems (below 2.17K lambda point) have become the norm for especially the superconducting radio frequency (SRF) technologies or the multi SRF niobium cavity cryomodules.

The performance comparisons are made for 2K operation for illustration purposes.

The values used in the illustrations can be calculated from the helium properties or obtained from the Temperature-Availability (T-e) diagram presented in Appendix-G.

The following four refrigeration process types describe some of the system design options available for sub-atmospheric load operation.

- **Type-1: Vacuum pumping on a helium bath**
- **Type-2: Sub-atmospheric refrigeration system**
- **Type-3: Cold compressor System**
- **Type-4: Hybrid systems**

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

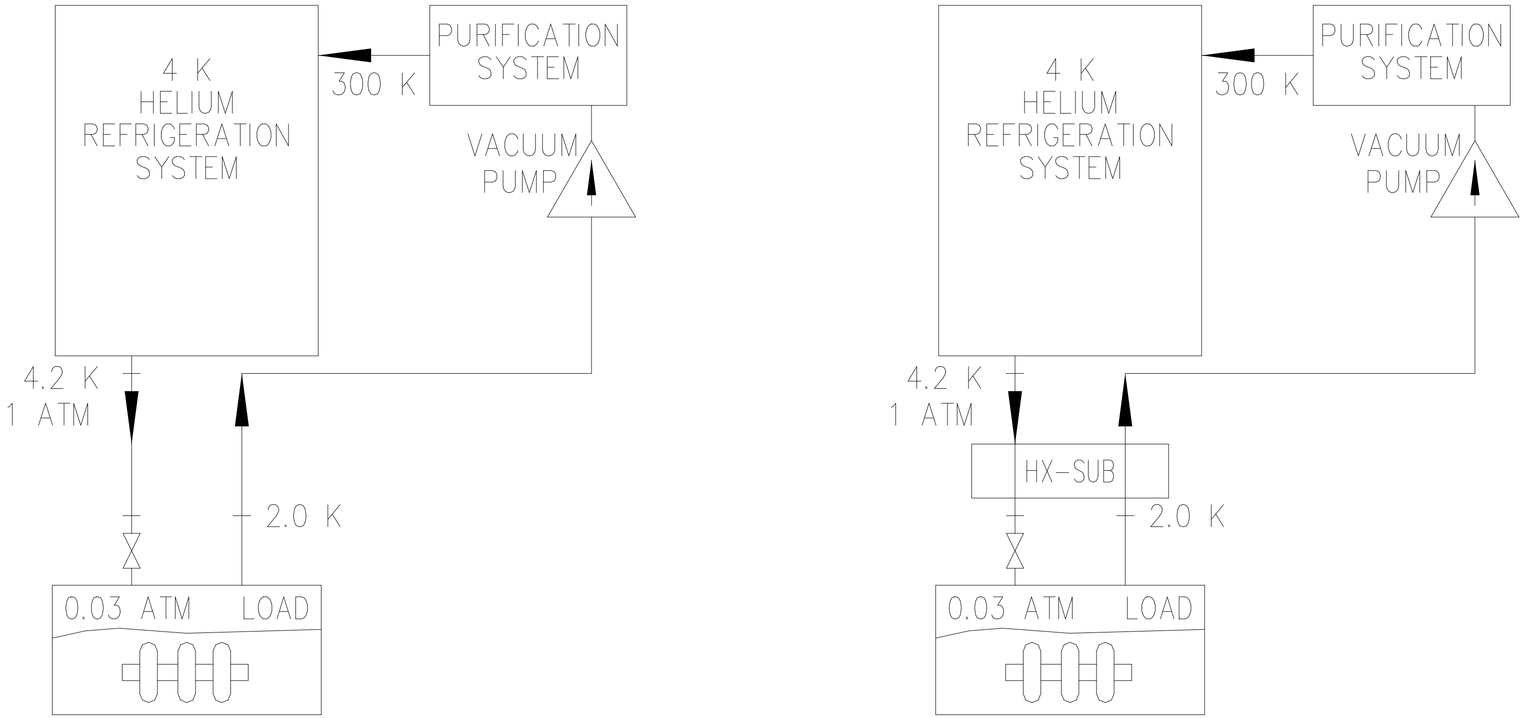


Figure 7.1.1: Vacuum pumping on the helium bath

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

1 g/s of 1 atm. saturated liquid helium will provide ~ 15 j/g or 15W of refrigeration capacity at 2.0K.

The Carnot work required is that for 1 g/s of 4.2K liquefaction Carnot work (~100 W of 4.2K refrigeration Carnot work) plus the vacuum pump isothermal compression work.

The Carnot specific power for this (Type-1-a) process is

$$= \quad [(T_0 \cdot \Delta S - \Delta H) + RT_i ln(Pr)]$$

$$= \quad \left[(6840) + \left(2.077 * 300 * \ln \{1/0.03\}\right) \right] \text{ w/(g/s) / 15 w/(g/s)}$$

$$= \quad [6840 + 2185] / 15 \qquad = \sim 600 \text{ w/w}$$

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

In an improved system including the sub-cooling heat exchanger (HX-Sub) shown in Figure 7.1.1 (b), 1 g/s of 1 atm. saturated liquid helium will provide ~ 20 j/g or 20W of refrigeration capacity at 2.0K.

The Carnot specific power for this (Type-1-b) process is

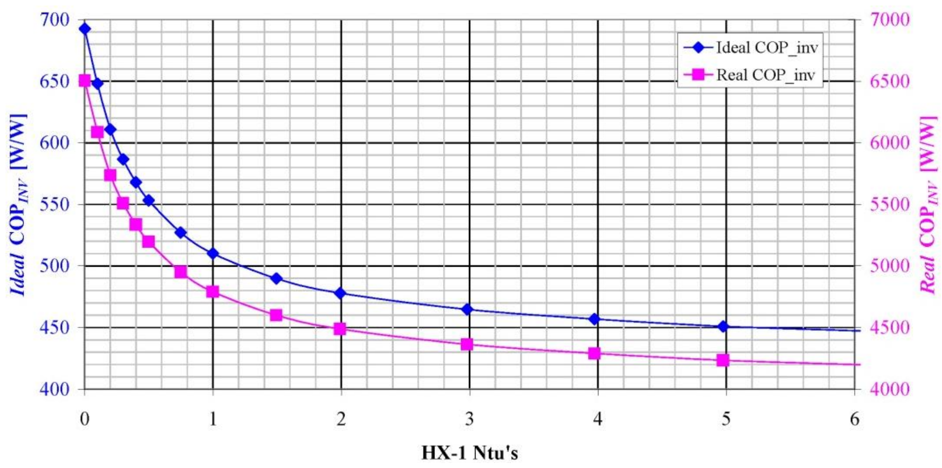
$$= [(T_0 \cdot \Delta S - \Delta H) + RT_i ln(Pr)]$$

$$= \left[(6840) + \left(2.077*300*\ln\{1/0.03\}\right)\right] \text{ w/(g/s) / 20 w/(g/s)}$$

$$= [6840 + 2185] / 20 \quad = \sim 450 \text{ w/w}$$

Or, the option (b) is~25% more efficient than option (a)

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)



Advantages:

- Simple system
- Smallest capital cost

Disadvantages:

- High operating cost

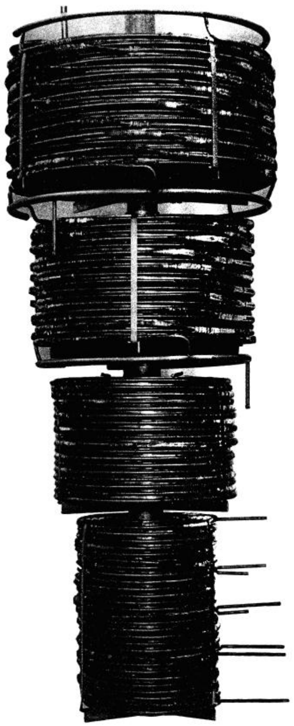
Type-2: Sub-atmospheric refrigeration system

This is an extension of the usual 4K helium refrigeration system design with the low pressure stream operating at sub-atmospheric conditions.

In this design type the practical constraints, such as the low-pressure stream pressure drop ($\Delta p / p \propto$ to the exergy loss) need to be addressed very carefully.

Type-2 process can be approximated with a standard 4K system by adding sub-atmospheric components as shown in Figure 7.2.2. Although the process is not as efficient as the integrated design, it is widely practiced with minor variations due to its ease of addition to an existing 4K system.

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)


LOW PRESSURE HEAT EXCHANGER

10 Torr pressure on the low pressure side of the heat exchangers

Figure 7.2.1: Stanford University 300W, 1.8 K Helium Liquefier (1969)

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

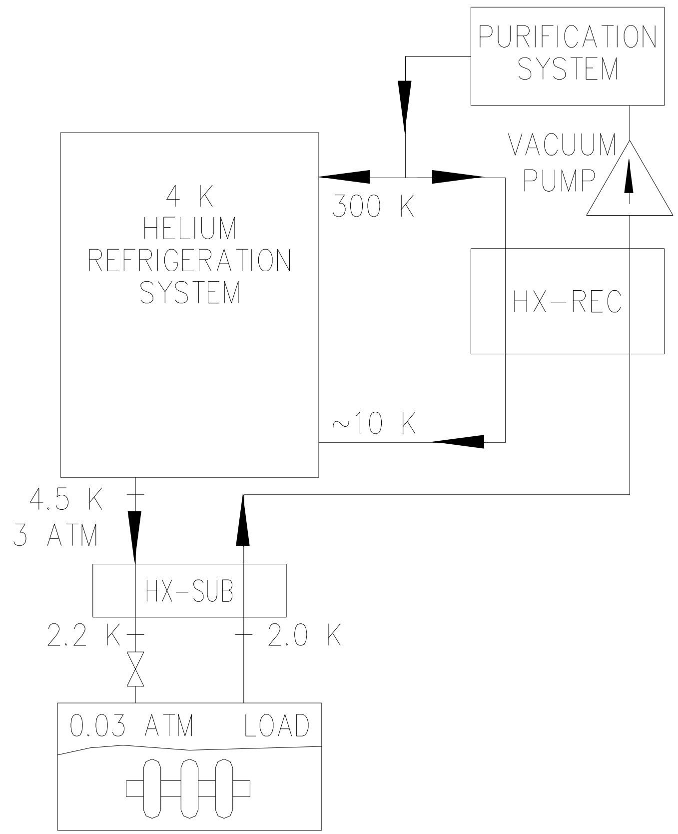


Figure 7.2.2: Vacuum pumping on the helium bath with HX

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

In this design with a load heat exchanger (HX-Sub), 1 g/s of helium flow to the load will provide ~20 J/g or 20W of refrigeration capacity at the required operating condition (e.g., 2K).

This in turn requires the input power required for 0.1 g/s liquefaction capacity from 300K to 4.5K and the 0.9 g/s liquefaction capacity from 10K to 4.5K plus the vacuum pump input power.

In both Types 1&2 the vacuum pump power required remains the same.

It is easier to transition to the load conditions (pump down)

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

The Carnot specific power for this (Type-2) process is

$$= \left[(0.1*6830)+(0.9*3003)+\left(2.077*300*\ln\{1/0.03\}\right)\right] \text{ w/(g/s) / 20 w/(g/s)}$$

= [683 + 2703+ 2185] / 20 = ~280 w/w

pressure drop (assumed to be 0.005 atm) and its sensitivity is as given in Type-2c calculation. The Carnot specific power for Type-2c is

$$= \left[(0.1*6830)+(0.9*3003)+\left(2.077*300*\ln\{1/0.025\}\right)\right] \text{ w/(g/s) / 20 w/(g/s)}$$

= [683 + 2703 + 2300] / 20 = ~284 w/w

Although the pressure drop effect on Carnot specific power seams rather insignificant, in actual systems this has very *strong influence* on the vacuum pump *pumping capacity* and its efficiency.

In the extreme limit, the vacuum pump ultimate pressure may limit the final temperature achievable for a given load.

APPENDIX – D

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

Advantages:

- Proven components for the system design
- Can be added to an existing 4K system with the addition of a refrigeration recovery heat exchanger
- Easy to reach load operating conditions
- Easy to efficiently turn down the system capacity to meet the reduced load

Disadvantages:

- Any leaks to the sub-atmospheric portion will contaminate the system
- Sub-atmospheric vacuum pumps and compressors are less efficient
- Pressure drop on the low pressure side of the refrigeration recovery HX and the system economics normally limit the system design to small to medium size loads (less than ~1KW at 2K) at low pressure (0.03 atm or 2K) operations

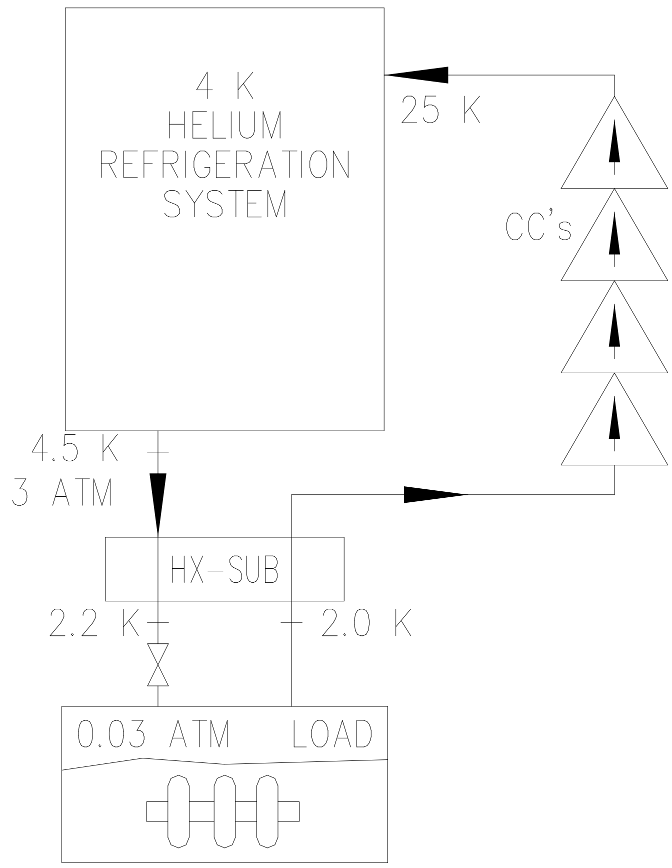
Type-3: Cold Compressor System

As explained above, Types 1&2 are not economical solutions for large capacity (typically > 1kW) nominal 2K refrigeration systems.

The volume flow and pressure ratios are too large for efficient compression of helium gas at room temperature with currently available sub-atmospheric equipment.

In a Type-3 system utilizing cold compressors (CC's), the process can be designed with minimum (to none) warm end sub-atmospheric components in the system. Since the helium is compressed at the cold end, the energy input to compress the helium including the compression inefficiencies is transferred as a refrigeration load to the 4K cold box and ultimately rejected to the environment through the 4K system compressors.

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)



Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

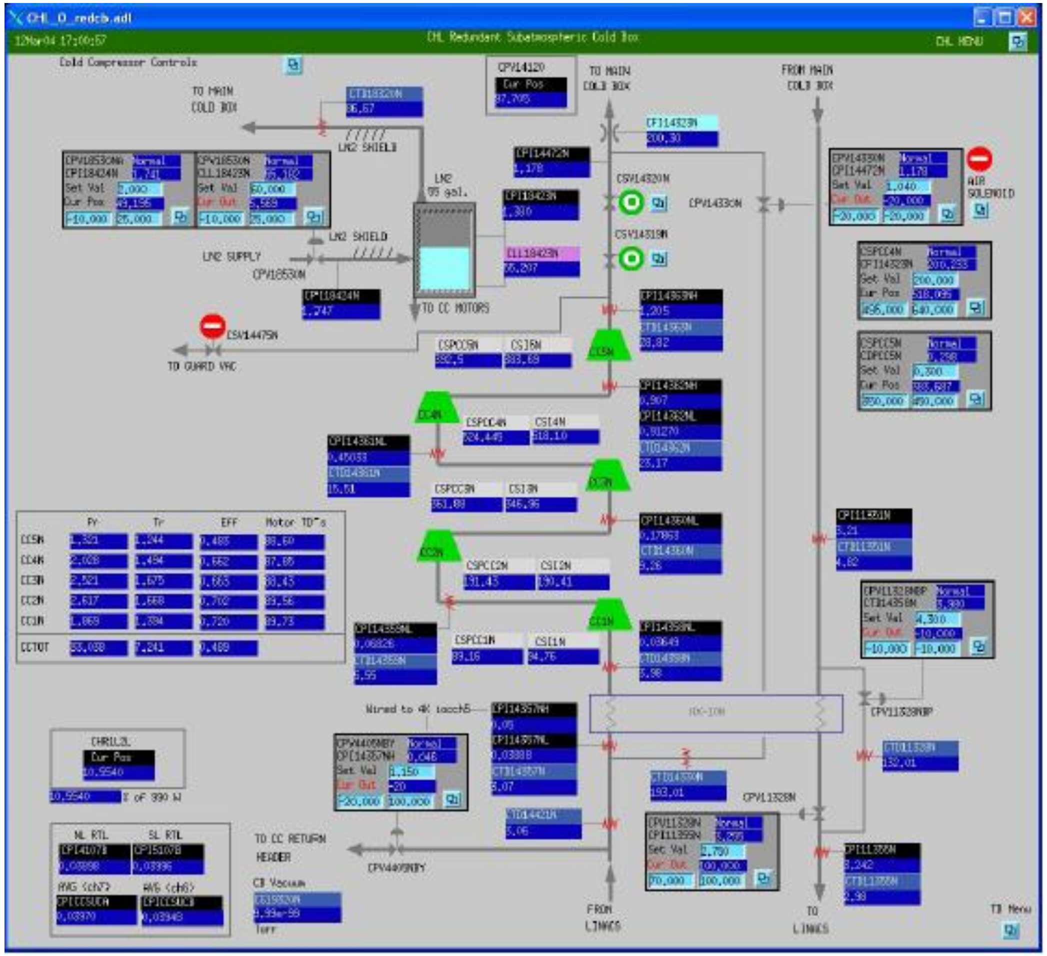


Figure 7.3.1: JLab Cold Compressor System

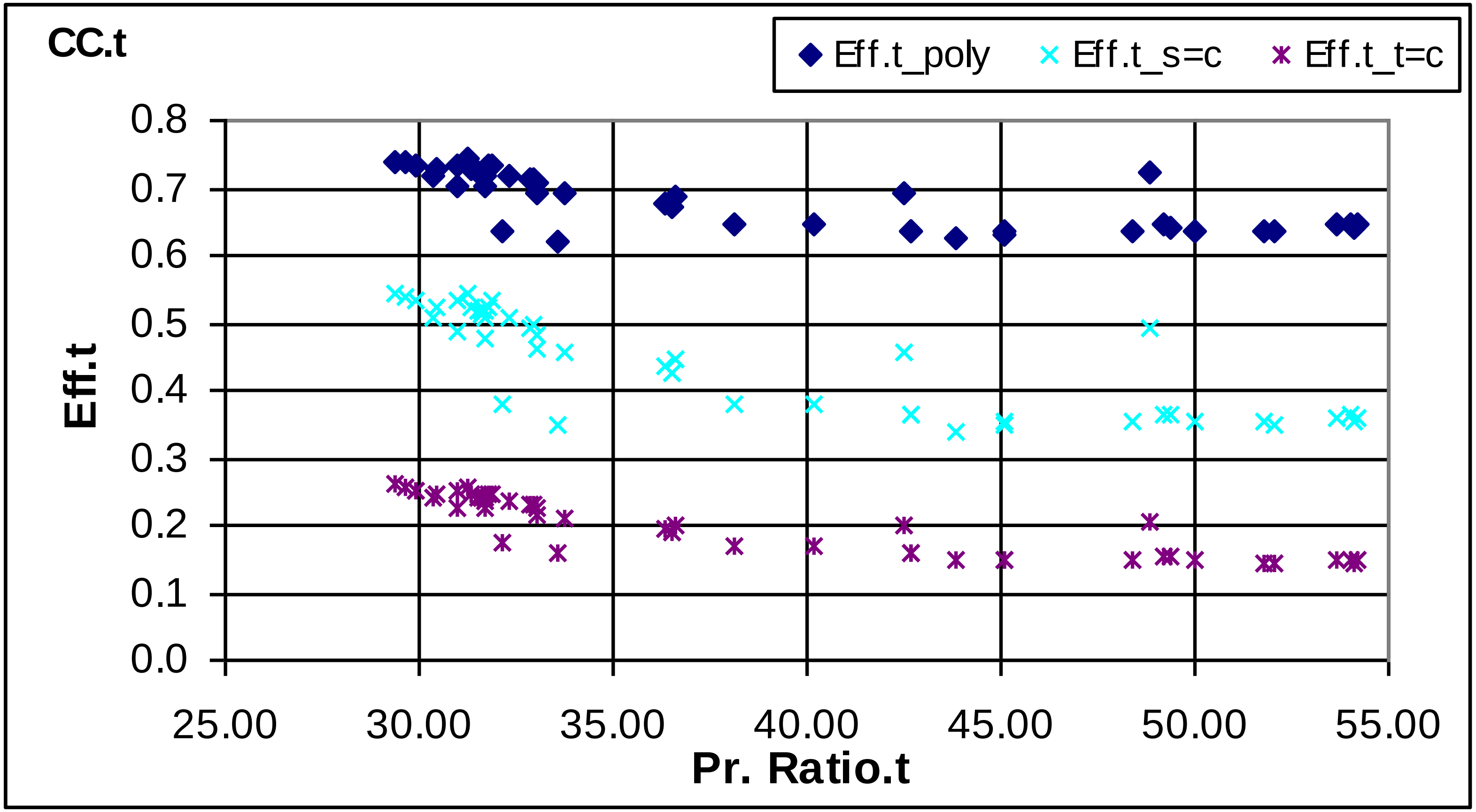
Type-3: Cold Compressor System (cont.)

The Type-3 process has many advantages and is used in many large systems. Figure 7.3.1 illustrates the cold compressor system used by JLab and a similar four-stage system designed by JLab presently in use at SNS.

The final HX-SUB should be located close to the load (e.g., SNS system) so that the distribution system heat leak is a 4K (low Carnot value) instead of 2K load, resulting in reduced CC flow and improved process efficiency.

The process of transforming the load condition from positive pressure (e.g., ~ 1atm) to the load operating conditions (e.g., ~0.03 atm) is called "pump down". The challenge is to find a satisfactory thermodynamic path (pump down path) to the final condition without violating any equipment operational limits or causing an emergency shutdown (a "trip") of the cold compressor system.

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)



Type-3: Cold Compressor System (cont.)

During the initial commissioning of the first Jlab 2K system in 1994, the causes for the instabilities (large flow variations) in the system during pump down were not understood.

At that time, the instabilities were believed to be caused by the compressor surge and stall characteristics.

By trial and error a successful and repeatable pump down path was found and JLAB continued to operate on that path.

During the commissioning of the second Jlab 2K system in 1999, it was apparent that the limitation of available torque from the cold compressor motors was one of the main reasons for the instabilities in the system during pump down. It was also identified that the power factor (PF) of the motor was one of the factors limiting the available cold compressor torque during pump down.

Type-3: Cold Compressor System (cont.)

This was the initial recognition that the flow instabilities were more probably the result of the motor characteristics than the cold compressor wheel characteristics, just the opposite of what was thought in 1994.

Since the motor and the compressor are on the same shaft, it was difficult to separate the cause and effect phenomena.

The recent testing at SNS provided the following additional insight.

The PF for all the cold compressor motors exhibited the following characteristic as shown in Figure 7.3.3 with respect to the speed.

Before this test at SNS in March of 2005, the periodic 60HZ influence on the PF was neither known nor anticipated.

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

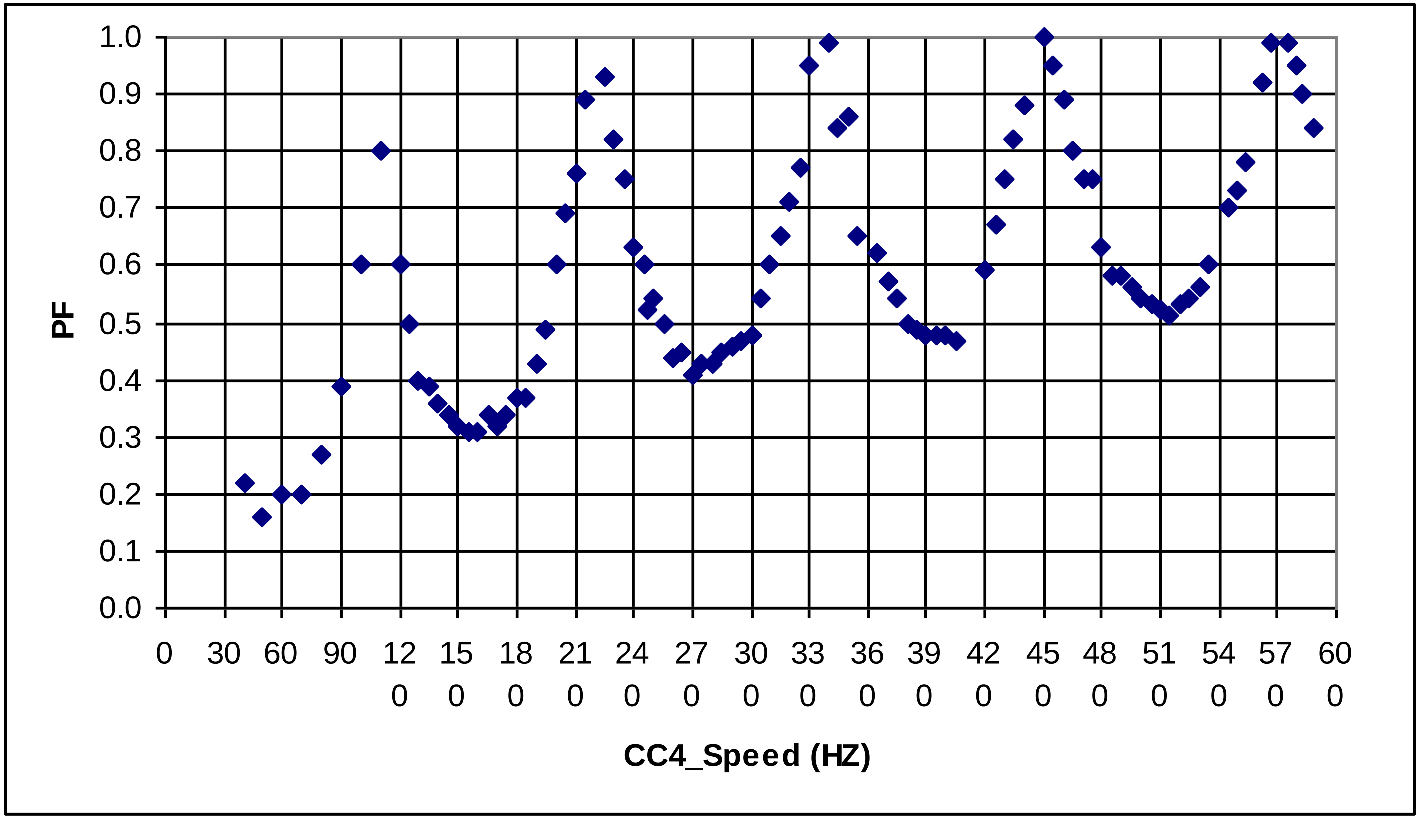


Figure 7.3.3: Variable frequency drive power factor (PF) with respect to speed

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

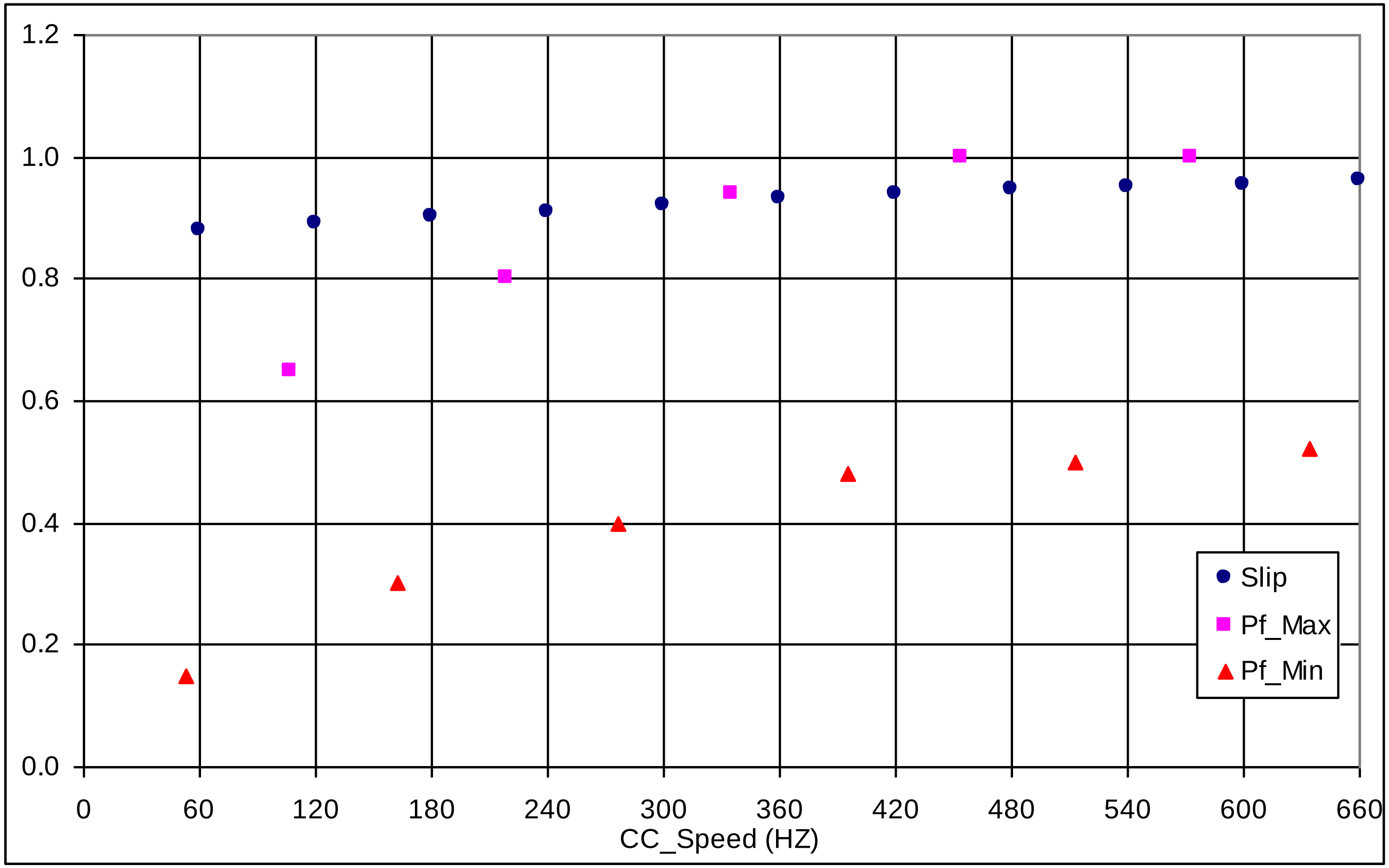


Figure 7.3.4: The minimum and maximum PF with respect to speed

Type-3: Cold Compressor System (cont.)

The minimum and maximum PF with respect to the motor (cold compressor) speed are presented in Figure 7.3.4. Both the min and max PF's are periodic at approximately 120HZ intervals and at 60HZ apart from each other. In the above graph, another curve showing the slip is plotted and is defined as the relative speed at which the lagging PF peak (min or max) occurred with respect to the 60HZ multiple. The PF peak occurred at the product of the slip and the 60HZ multiple. As the speed increased, the PF and the slip improved and the peaks occurred closer to the 60HZ multiples. The min and max PF's are clearly influenced by the standard 60HZ power supply frequency.

Type-3: Cold Compressor System (cont.)

Initial thoughts are:

may be the result of the physical combinations of passive and active electronic components in the output section of the drive and their characteristic frequency responses.

The output of the drive may be largely affected by the resonance tune of the electronics and their signal response at various frequencies.

The reasons for this behavior and the ways to improve the minimum PF for these variable frequency drives is of continued interest.

Attempts to find a device that is compatible with the output of the variable frequency drive that will correct for variable PF over the wide operating frequency range has not successful to date, but that effort is continuing.

Type-3: Cold Compressor System (cont.)

- It is strongly believed that the pump downs can be accomplished with constant design speed (gear) ratios, if the minimum PF is brought close to the maximum. This will simplify the pump down path selections and provide a more robust cold compressor operation. The PF improvement can also help the turn down range for these systems and thus improve the overall system efficiency over a large operating range

- The Carnot specific power for this (Type-3) process is

= $\left[4381\right]$ w/(g/s) / 20 w/(g/s)

= **~220 w/w**

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

Type-3: Cold Compressor System (cont.)

Advantages:

- Least possibility for contamination, since there are no sub-atmospheric connections at the warm end
- High efficiency at and near the design point operating conditions
- Very reliable system
- Pressure drop on the low pressure side of the cold box is not the limiting case
- Require less compressor floor space, since the displacement required for the sub-atmospheric volume gas is reduced by compression at low temperature

Disadvantages:

- Limited commercially available cold compressor options
- Increases the 4.5K system capacity required
- Higher capital cost: cold compressors (presently)
- Higher operating costs at reduced capacity (poor turn-down)
- Slower to reach the system operating conditions

Type-4: Hybrid systems

The Type-4 process can benefit from design trade offs between the Type-2 and Type-3 systems. This option draws a balance between the cold compressor input work to the 4.5K system and the overall process efficiency.

Figures 7.4.1 (a) and (b) present two variations; many other process options depending on the other constraints are available for this hybrid process concept.

The Tore-Supra system (300 W @ 1.75K) [16] & [17] was designed with the Type-4 concept.

The CERN LHC cold compressor systems are also of this type; where the warm compressors assist the cold compressors in achieving the total pressure ratio required.

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

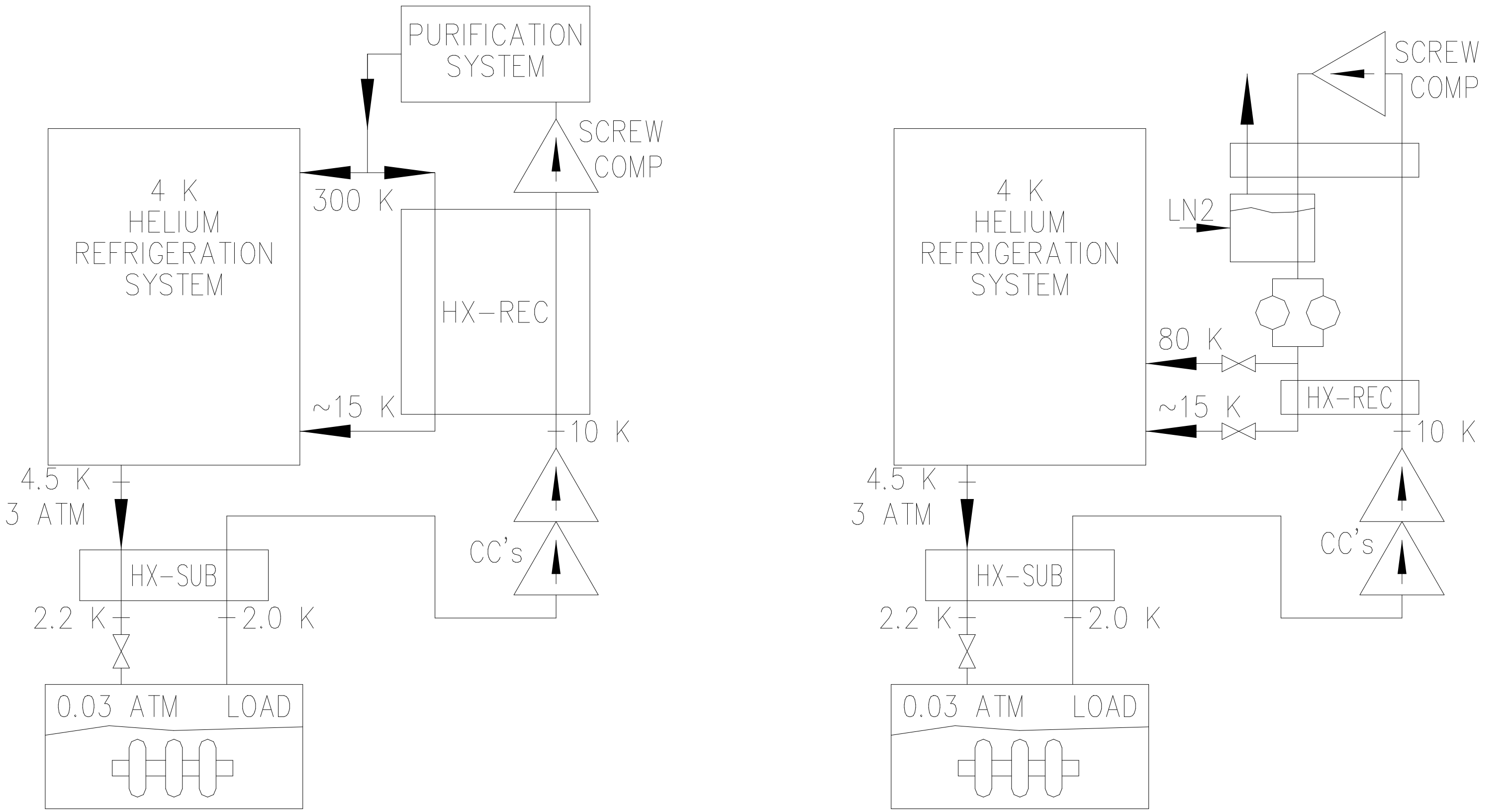


Figure: 7.4.1 A Hybrid Concept

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

The Carnot specific power for this (Type-4) process is

$$= \left[\left(0.1*6830\right)+\left(0.9*3628\right)+\left(2.077*300*\ln\left\{1/0.25\right\}\right)\right] \text{ w/(g/s) / 20 w/(g/s)}$$

= [683 + 3265+ 864] / 20 = ~240 w/w

The recovery heat exchanger (HX-REC) contributes to pressure drop and its sensitivity is as given in Type-4d calculation.

The Carnot specific power for Type-4d is

$$= \left[\left(0.1*6830\right)+\left(0.9*3628\right)+\left(2.077*300*\ln\left\{1/0.20\right\}\right)\right] \text{ w/(g/s) / 20 w/(g/s)}$$

= [683 + 3265+ 1003] / 20 = ~248 w/w

Type-4: Cold Compressor System (cont.)

Advantages:

- Pressure drop on the low pressure side of the cold box is not a limiting case
- Can add additional system capacity by adding expanders and compressors to the sub-atmospheric cold box system
- Easy to pump down and reach the operating conditions

  Can reduce the system capacity (turn down) efficiently

Disadvantages:

- Susceptible to air leaks at the warm end that can contaminate the system
- Refrigeration capacity for cold compressor compression work still needs to be supplied by the 4 K system capacity
- Sub-atmospheric warm compressors with large pressure ratios are less efficient and can also lead to air leak contamination
- More compressor room floor space is required to handle the low pressure helium gas volume

Helium Refrigeration Systems for Below 4.2 K Operations (Cont.)

Typical Expected values:				
	2K Process Specific Power	**Assumed Carnot Eff. of support refrigerator**	**Overall Specific Power**	**Overall Carnot Efficiency**
	W / W		**W / W**	
Type-1 (a)	**600**	**0.1**	**6000**	**0.027**
Type-1 (b)	**450**	**0.1**	**4500**	**0.036**
Type-2	**280**	**0.2**	**1400**	**0.114**
Type-2 (c)	**284**	**0.19**	**1495**	**0.107**
Type-3	**220**	**0.25**	**880**	**0.182**
Type-4	**240**	**0.24**	**1000**	**0.160**
Type-4 (d)	**248**	**0.23**	**1078**	**0.148**

Summary

It is helpful to compare the nominal overall specific work (total input power to refrigeration power, W/W) and efficiency for these various processes. The approximate overall specific power and Carnot efficiency for system sizes *typical for the Type* and the efficiency of the *4K refrigerators typically used* with the effect of the minor modifications in each (to indicate the efficiency penalty due to modification) are given in the following table 7.5.1

8. Conclusions

Carefully develop new system requirements that consider loads including all anticipated transients while allowing for the practical strengths and weakness of all system components considered.

In addition, assure that the design goals are carried through the selection of all components and into the detailed system design and that they are not skewed with perceived constraints.

If the end user has difficulty defining the requirements or how to achieve them in the specification, then it certainly can not be assumed that the vendor will be able to provide them.

Conclusions (Cont.)

What is an optimum system? Does it result in the:

1) **Minimum operating cost?**
2) **Minimum capital cost?**
3) **Minimum maintenance cost?**
4) **Maximum system capacity?**
5) **Maximum availability of the system?**

Or, A combination of some or all the above

Or, Some other factors?

What do you think?

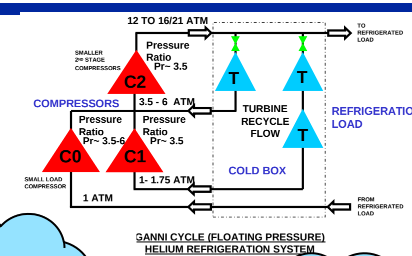
Conclusions (Cont.)

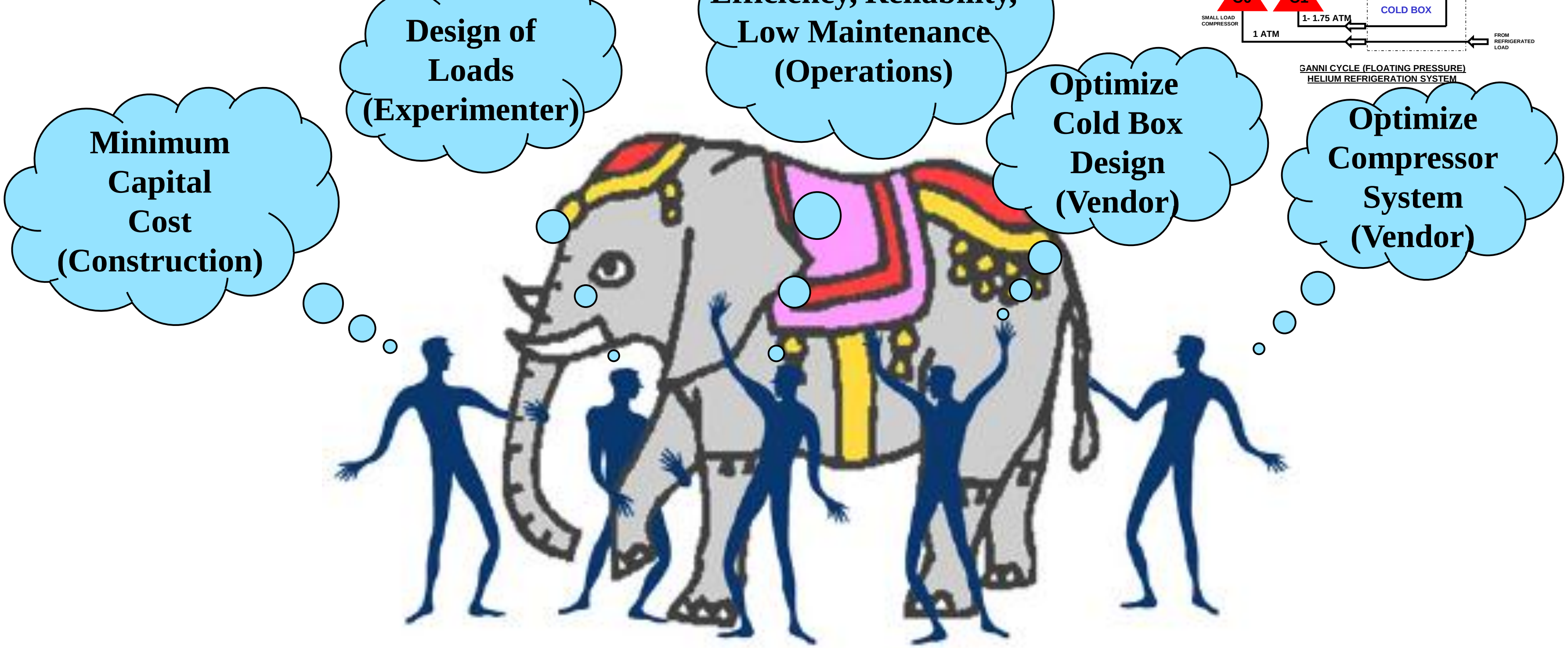
Hopefully the information presented in these notes and other information available will help you to design and operate cryogenic systems at optimal conditions and to answer the above questions. Floating pressure-Ganni Cycles are a step in that direction. The central theme of these notes is to minimize the input power for all required operating conditions. This will help to save our natural resources; an objective that is worth pursuing.

Hope you saw the

"Need for understanding the fundamentals"

What is an "Optimal" System





- **One's viewpoint can be based only on their role and focus within a project**
- **Easy to believe that one's goals are mutually exclusive of others**
- **Many believe that maximum system efficiency occurs only at one set of fixed operating conditions**

Conclusion

We like to see all of us to take personal interest

in how we use the precious commodity

energy

in accomplishing the end goal !!!

Thank you all for your interest

Conclusions (Cont.)

My special thanks to the members of JLab cryo department and especially to *Peter Knudsen* for making substantial contributions and all the other reviewers for helping me develop the notes.

Thank you all for your interest in this course

VenkataRao Ganni

Note: **All suggestions to correct any errors and to improve this manuscript will be greatly appreciated.**

Conclusions (Cont.)

Discussions & Remarks

Open Session