System Optimization
&
Ganni Cycle

By

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Tentative Schedule

Chapter		Duration (min)
0	Questions on earlier materials	5
5	System Optimization & Ganni Cycle	40
	Floating Pressure Application to 12 GeV Cycle Design	15
10	Optimal Operation of the Existing Helium Systems	20
	Discussions	?

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**

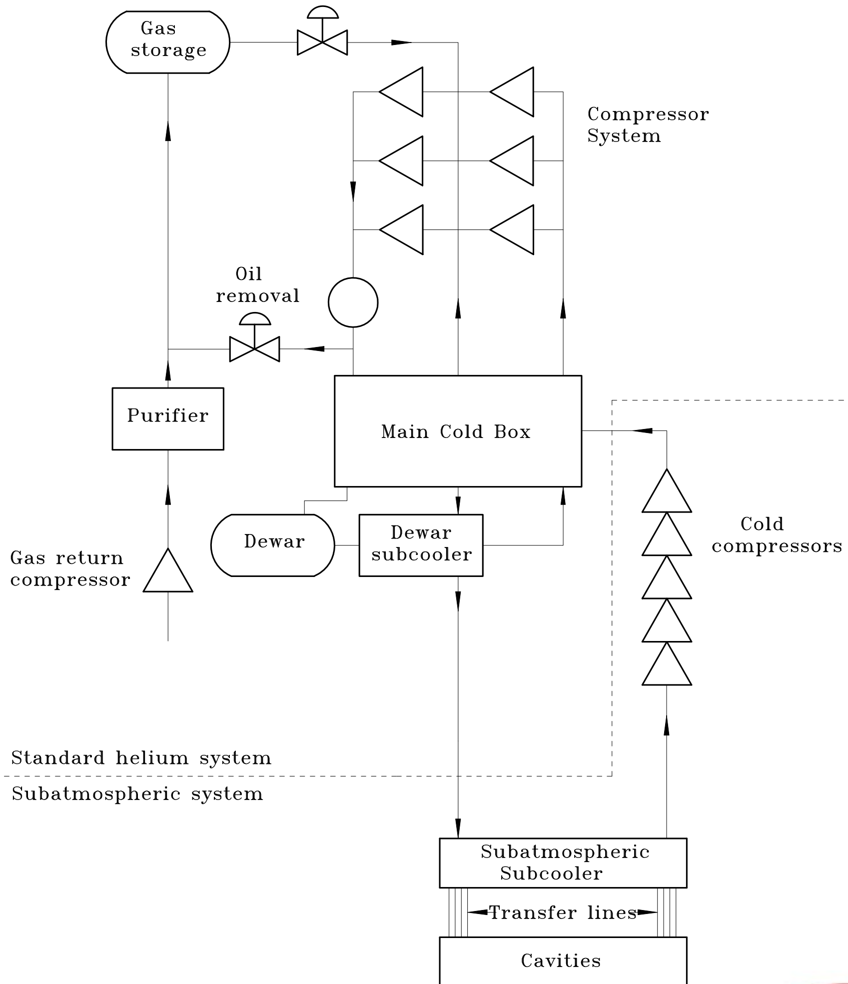
Traditionally a design for maximum efficiency at one operating point is referred as the optimum system design.

System Optimization (Cont.)

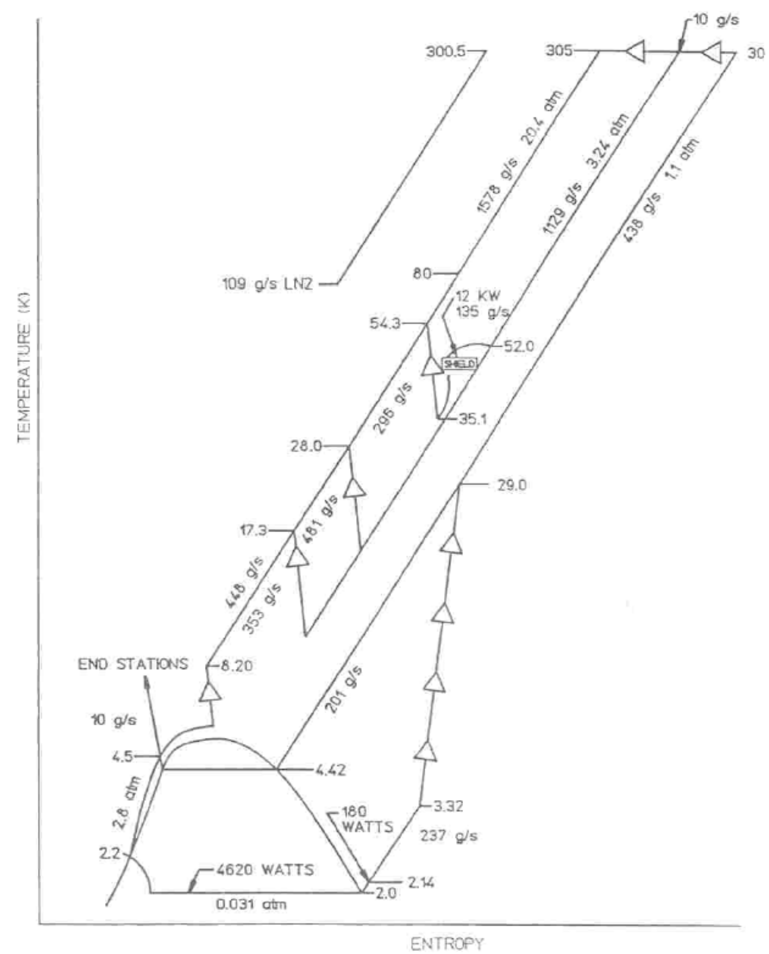
- **The above five factors (or perhaps more) are rarely looked at as the optimization goals.**
- **The demand on equipment varies substantially between operating as a refrigerator (i.e., Hx dominance) and liquefier (i.e., expander dominance).**
- ***The challenge is to envision a cycle considering these optimization goals, using real components, capable of operating close to maximum efficiency for a load varying from a maximum to minimum capacity and from full refrigeration to full liquefaction mode or in any partial load combinations.***

Operation of the Helium Refrigeration System

Central Helium Liquefier (CHL) at JLab



Original Design TS Diagram



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.

System Optimization (Cont.)

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.**

System Optimization (Cont.)

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 is between 2 and 5 at peak efficiency.**
- **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.**

System Optimization (Cont.)

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, since it plays a significant role in choosing the number of Carnot steps.**
- **For larger systems, pressure ratio, efficiency, arrangement and number of expanders will lead to the optimum number of Carnot steps.**

System Optimization (Cont.)

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 by speed**
- **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 inlet pressures to the turbo expander(s)**
- **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 size to ~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).***

System Optimization (Cont.)

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.

System Optimization (Cont.)

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.

System Optimization (Cont.)

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.

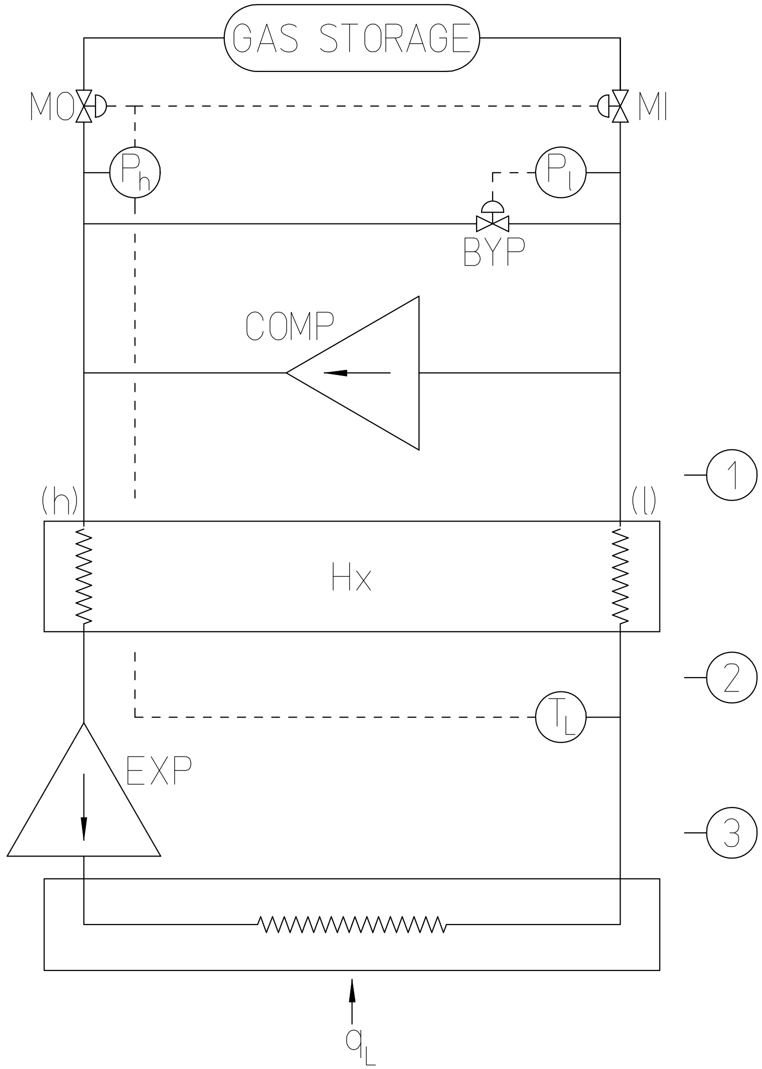
Floating Pressure Process - System Optimization (Cont.)

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

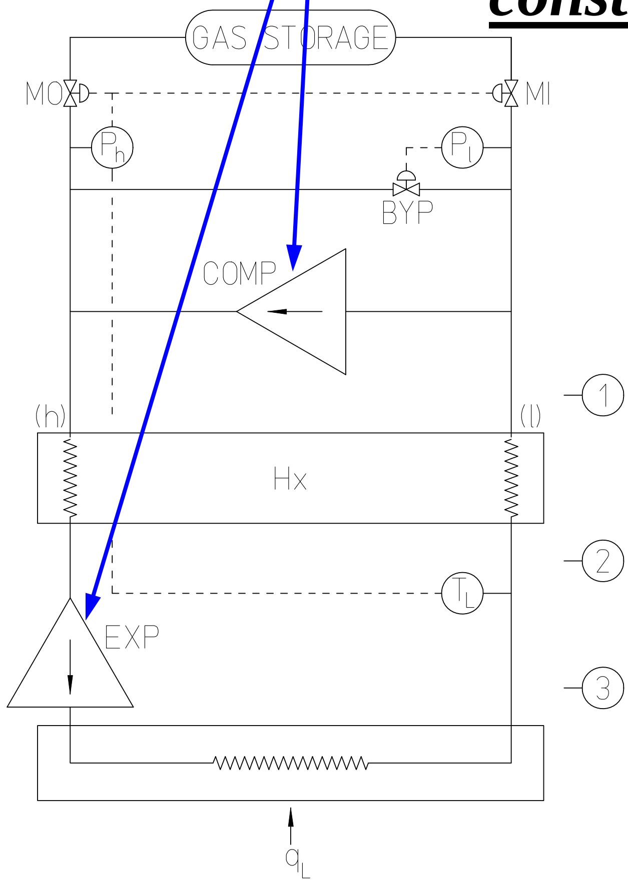
If left unconstrained, these two devices establish

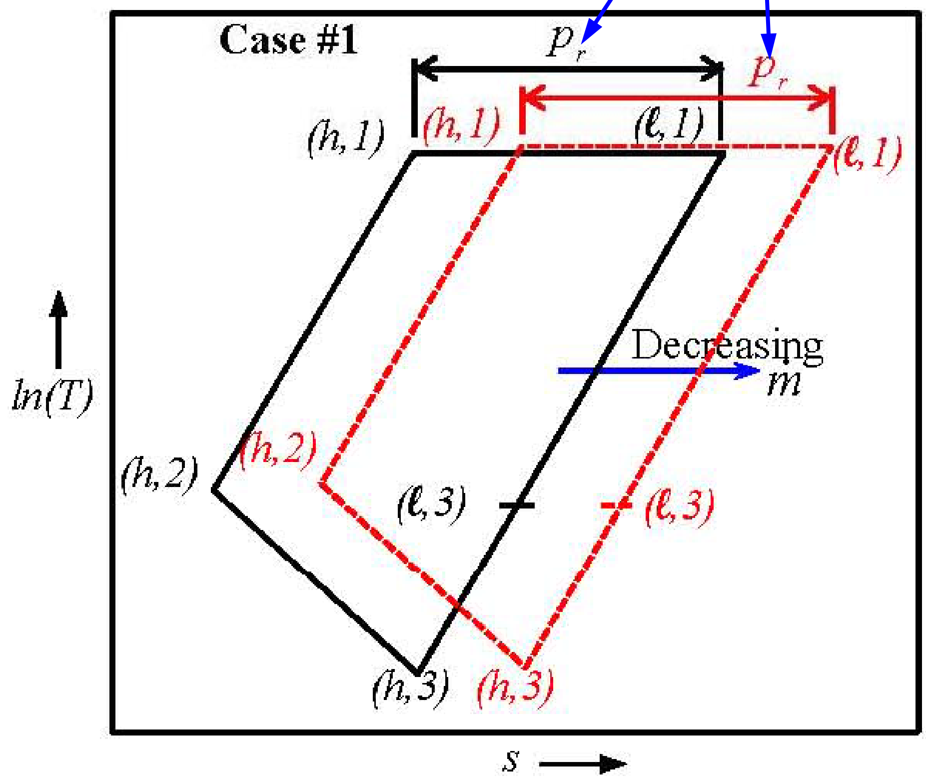
- ***Essentially constant pressure ratio* and,**
- ***Essentially constant Carnot efficiency***

For a given gas charge

Floating Pressure Process - System Optimization (Cont.)

The compressor and expander establish an *<u>essentially constant pressure ratio and constant system Carnot efficiency</u>*

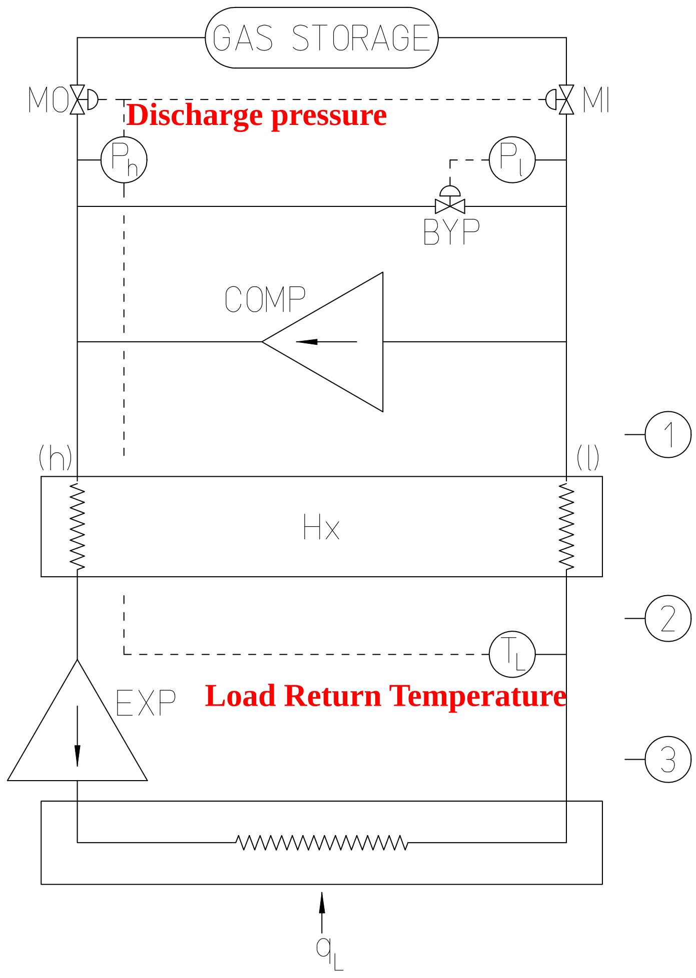




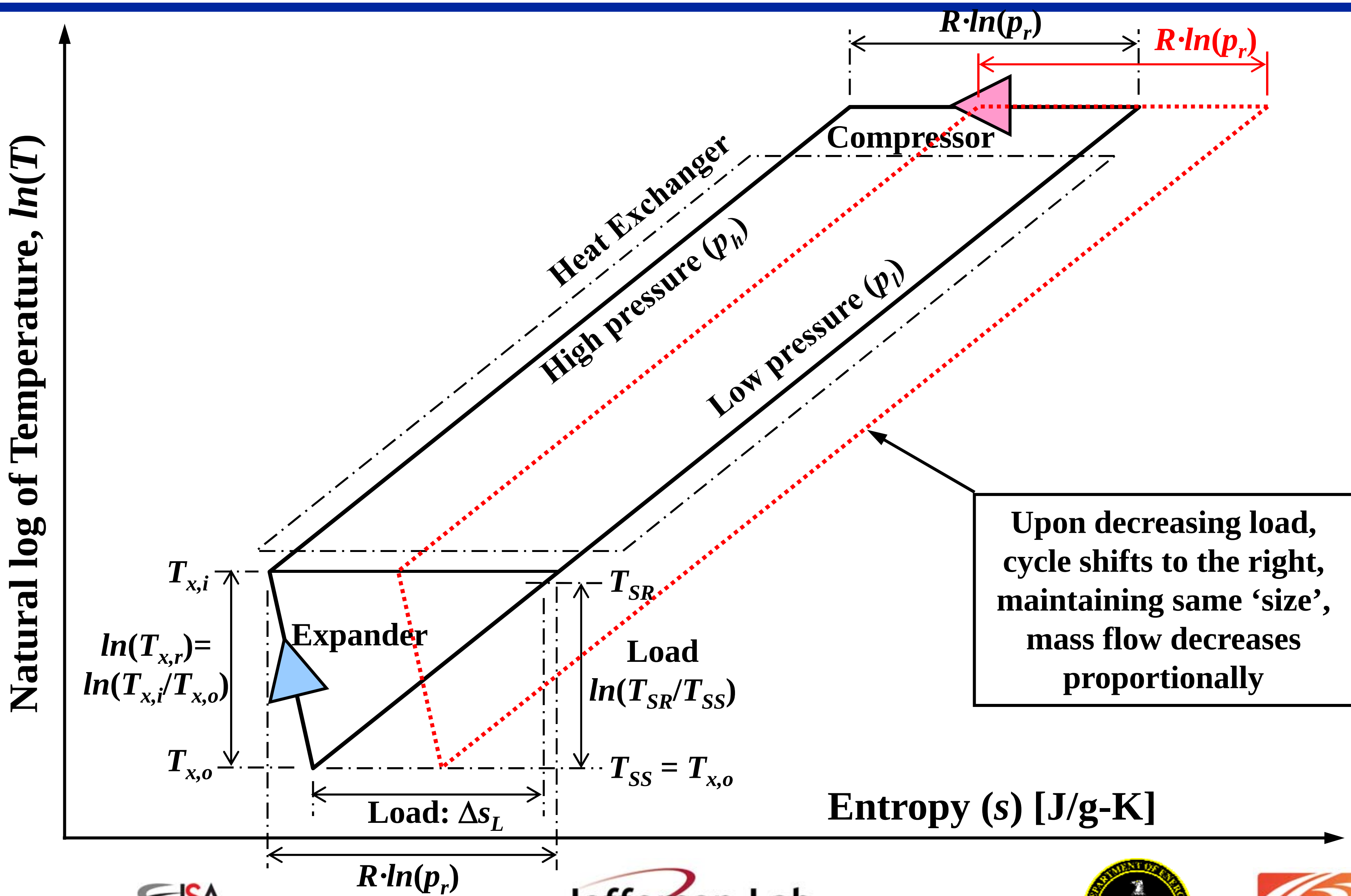
General Arrangement for Floating Pressure Process Cycle (*patent pending*)

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',

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

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

- 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

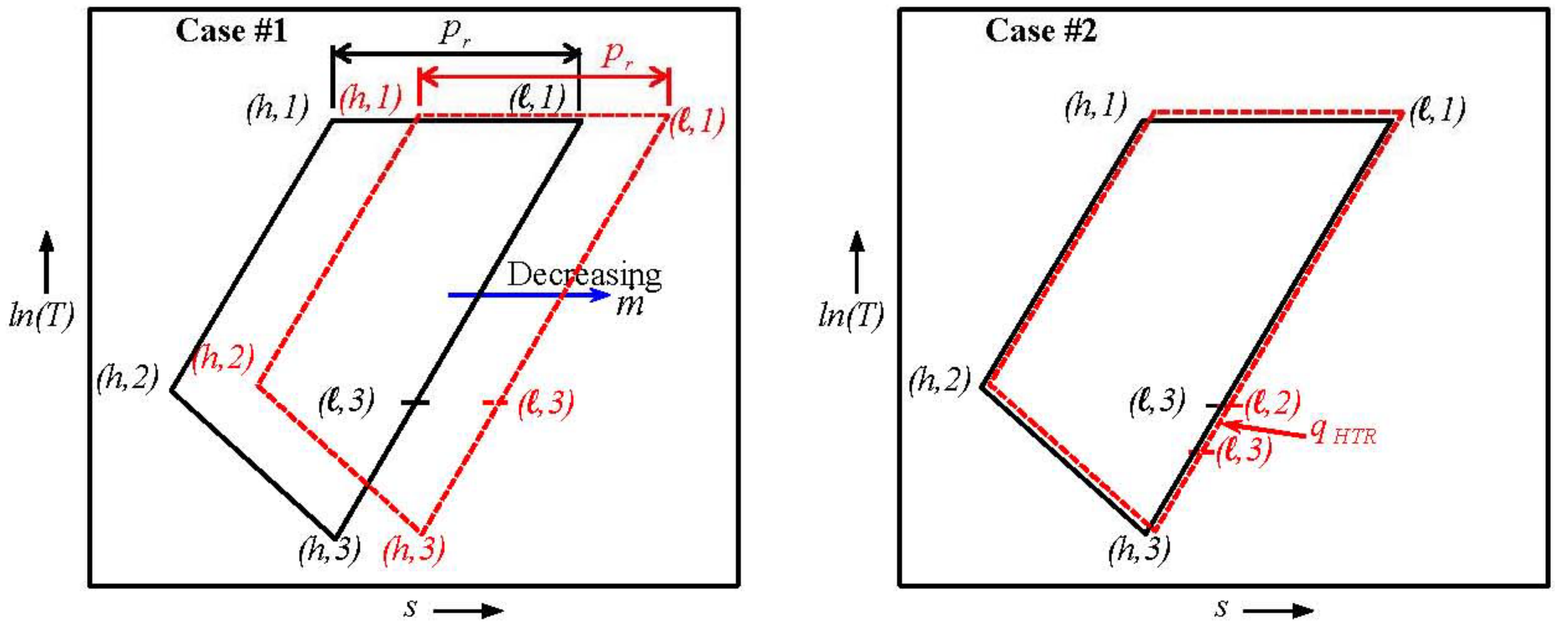
Methods to Control Shield Refrigerator Capacity

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.)

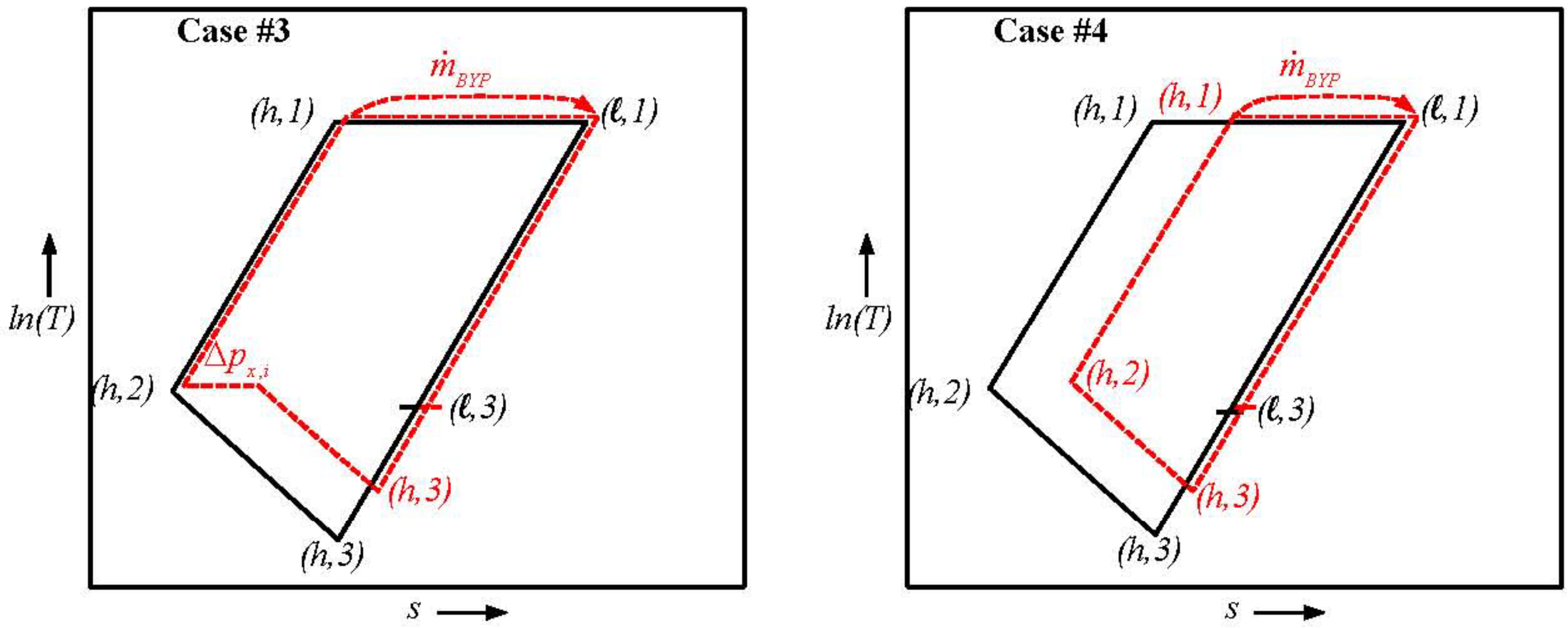
TS Diagram of Case #1 & #2



Note: Case #1 is the Floating Pressure Process

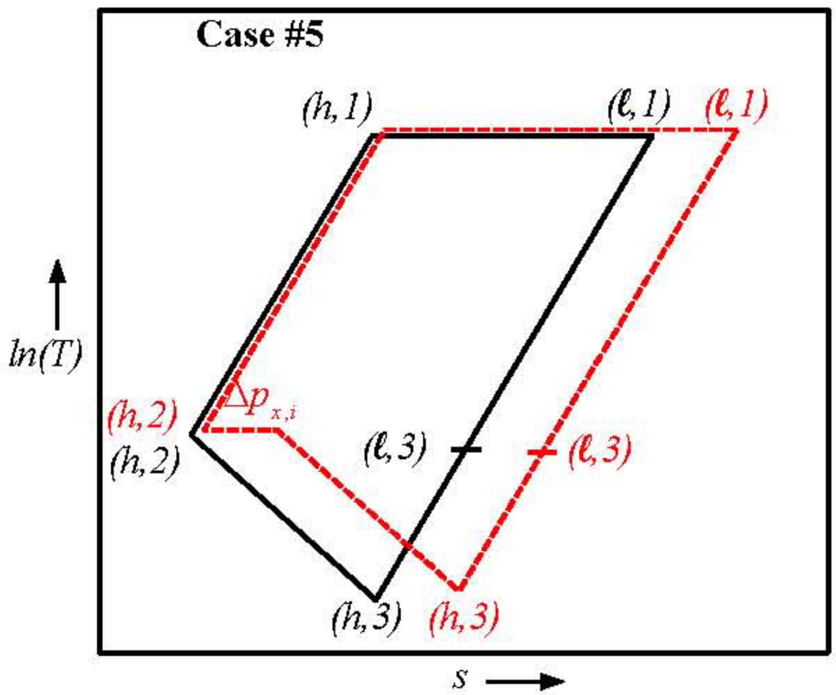
Capacity Modulation(Cont.)

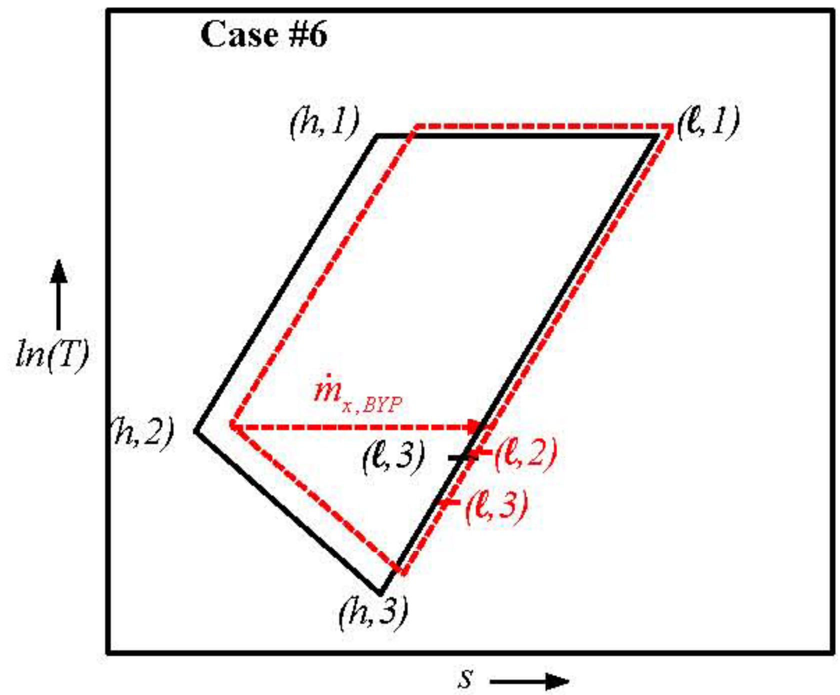
TS Diagram of Cases #3 & #4



Capacity Modulation (Cont.)

TS Diagram of Cases #5 & #6





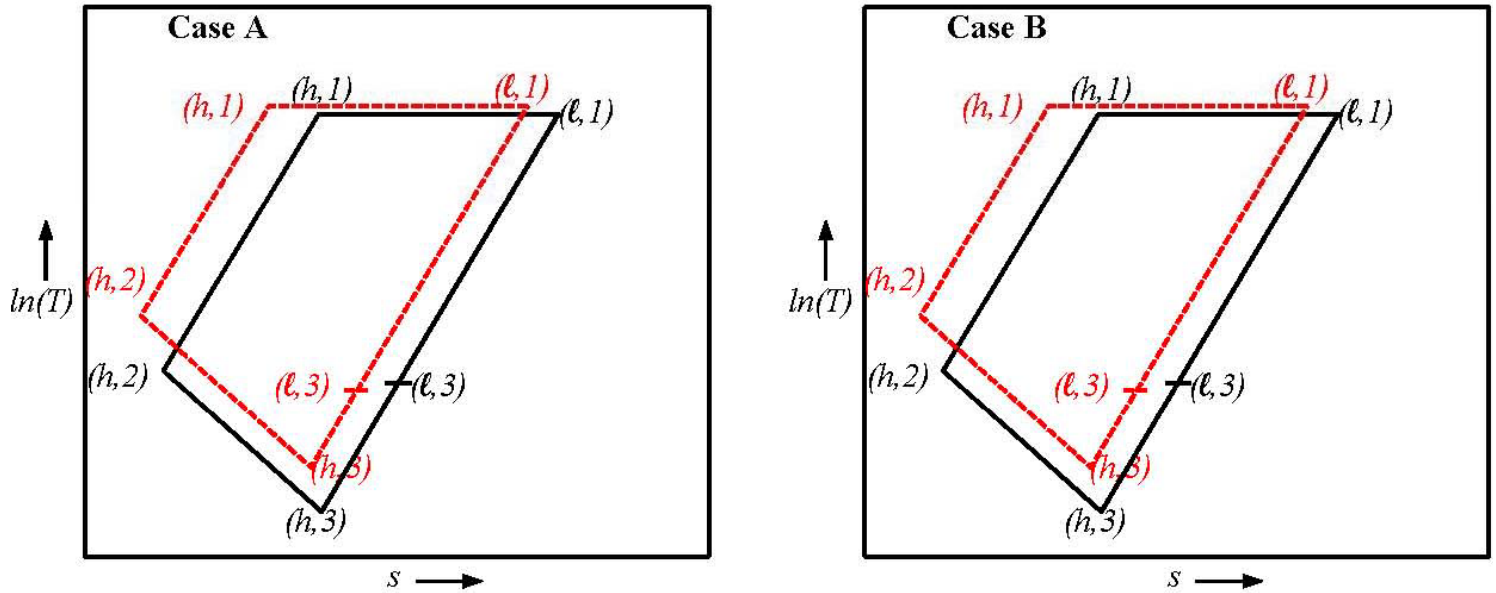
Variations in Equipment Parameters

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

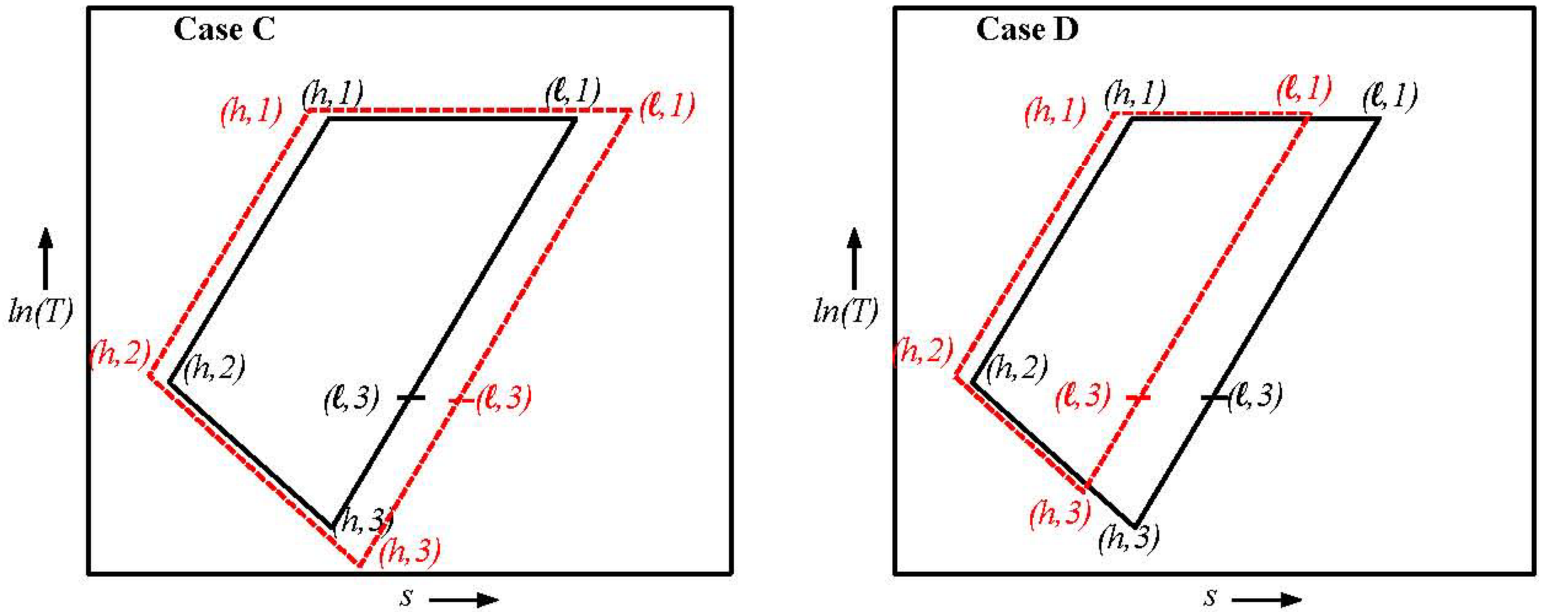
Variations in Equipment Parameters

TS Diagram of Cases A & B



Variations in Equipment Parameters

TS Diagram of Cases C & D



Floating Pressure Process - System Optimization (Cont.)

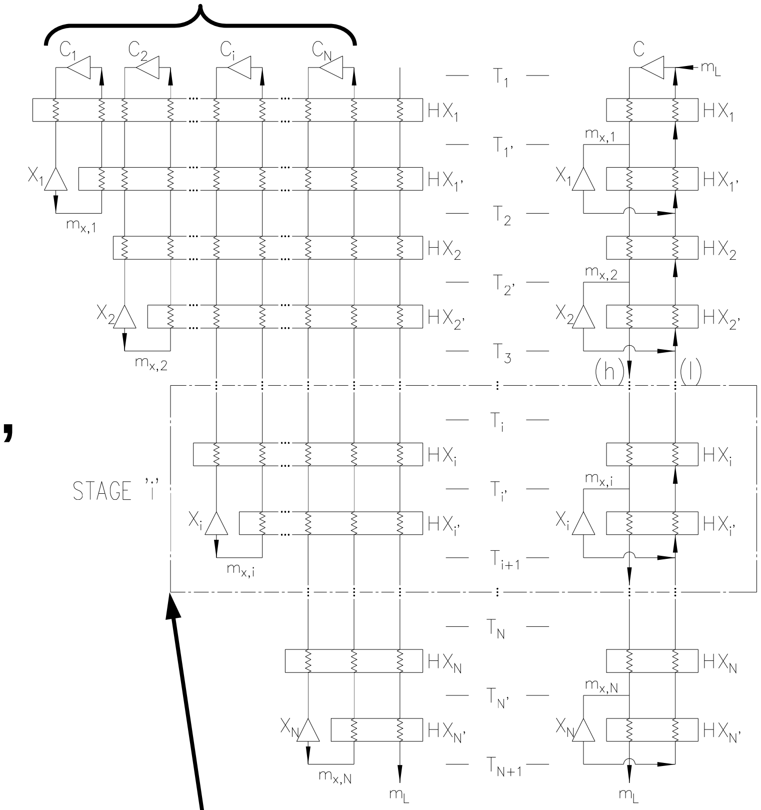
If, instead of using the Floating Pressure Process (as discussed in Case #1), one of the load adjustment mechanisms in Cases #2 to #6 were implemented in attempting to bring the off-design condition back to the TS design condition one of two results would occur:

- **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.)**

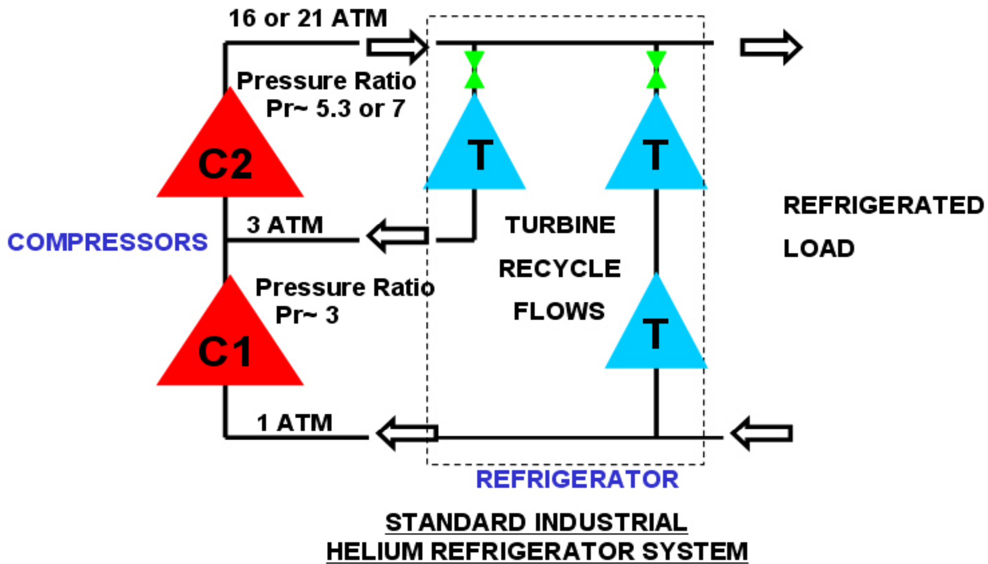
- **So, how does this apply to helium liquefiers and refrigerators?**
- **Recall that *each expansion stage is basically the cycle described in the Floating Pressure Process***
- **For liquefiers and mix-mode systems, 60 to 90% of the total system flow is through the turbines (providing the cooling)**
- **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.)

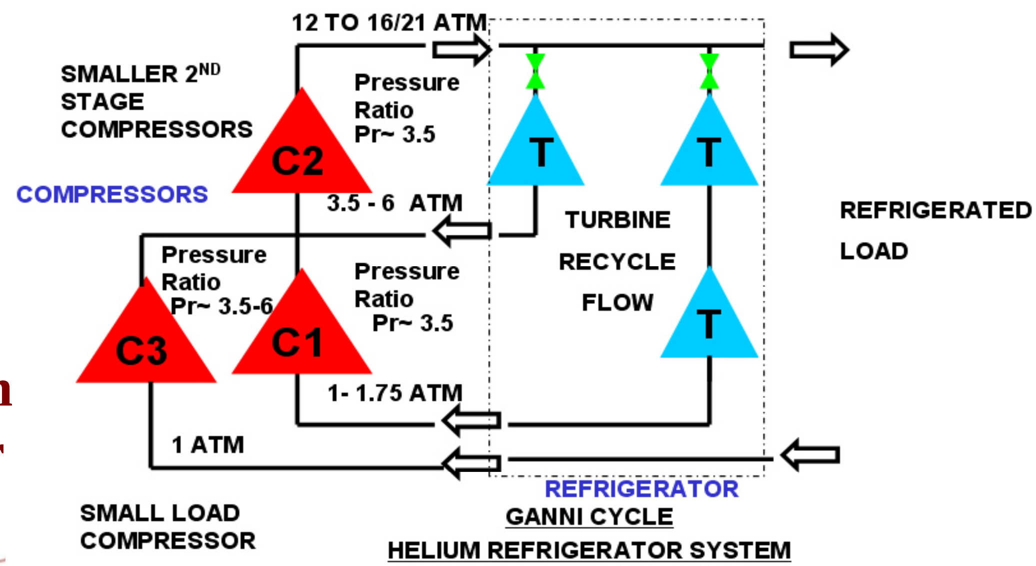


Traditional Helium Cycles

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).



Floating Pressure Process - System Optimization (Cont.)

Summary **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 (for the last 20 years) to stay status quo:

- **"We have done this before" – really???** (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.

Application of Optimization to
CHL-II Cycle Specification
for JLab 12 GeV Upgrade

P. Knudsen

JLab 12 GeV Helium Refrigerator Cycle Studies

- **Purpose**
  - —**Lay the "ground work" for the refrigeration system specifications by obtaining a thermodynamically <u>optimum practical cycle</u> configuration for all the load requirements**
    - **Establish the <u>probable optimum cycle</u>, for <u>all probable vendors</u>**
    - **Establish number and size of major components**
  - —**Support concurrent civil (building and utility) design**

- **Why?...to be able to,**
  - —**Effectively communicate our needs to the vendors**
  - —**Control the quality of equipment Jlab will receive**
  - —**Use the lessons learned form original CHL, SNS etc.; eliminating or minimizing the past mistakes**
  - —**Compare with other present state of the art systems of comparable size; e.g., CERN**

CHL Cryo Plant Capacities

- **Existing CHL #1 supporting Current 6 GeV**
  - **Capacity: 4.6 kW @2.1K,**
  - **12 kW @ 35K-55K**
  - **10 g/s liquefaction @ 4.5K**

- **New CHL #2 to support Future 12 GeV**
  - **Capacity: 4.6 kW @2.1K,**
  - **12 kW @ 35K-55K**
  - **15 g/s liquefaction @ 4.5K**

12 GeV Cycle Carnot Analysis

Jlab 12 GeV Carnot Cycle Analysis

Loads:

	w [g/s]	T_{sup} [K]	T_{ret} [K]
4.5K Liq.	15.0	4.59	78.43
CC Flow	238.0	4.59	35.00
Shield	192.1	35.70	47.73

Expander #1 Shield Flow

q_{sh}	12	[kW]	*shield load*
$T_{SS} = T_{h,2'}$	35.70	[K]	*shield supply temperature*
$T_{SR} = T_{l,3}$	47.73	[K]	*shield return temperature*
C_{p0}	5.193	[J/g-K]	*specific heat at const. pressure*
w_{sh}	192.1	[g/s]	*shield flow rate*

$w_{x,1}$	339	[g/s]	*expander #1 flow*
$w_{x,2/3}$	417	[g/s]	*expander #2 - #3 string flow*
$w_{x,4/5}$	393	[g/s]	*expander #4 - #5 string flow*
$w_{x,6}$	377	[g/s]	*expander #6 string flow*
w_c	**1541**	[g/s]	*est. total compressor flow*

Stage 1 (Shield & Refrigerator End)

N	1		*# stages*
T_1	80	[K]	*1st stage (h) inlet temperature*
$T_L = T_{N+1}$	35	[K]	*Nth stage (l) inlet temperature*
Θ	2.286		*total temperature ratio*
τ	0.020		$= \Delta T_{hl,i} / T_{l,i}$
θ_{hl}	1.020		$= T_{h,i} / T_{l,i}$
θ_i	2.286		*stage temperature ratio* $= T_{h,i} / T_{l,i+1}$
w_L	15.0	[g/s]	*liquefaction flow*
w_R	1126	[g/s]	*refrigeration flow*
w_{LR}	1141	[g/s]	$= w_L + w_R$
ν_L	0.013147		$= w_L / w_{LR}$
Γ	3.000		*expander pressure ratio*
ϕ	0.400		$= (\gamma-1)/\gamma$
η_x	75.0%		*expander isentropic efficiency*
θ_x	1.364		*expander temperature ratio*
f	0.025		
g	1.241		
h	0.344		
k	0.319		
$\beta_{r,0}$	0.129		$= w_{x,N} / w_{LR}$
β_r	1.078		$= w_{x,i} / w_{x,i+1}$

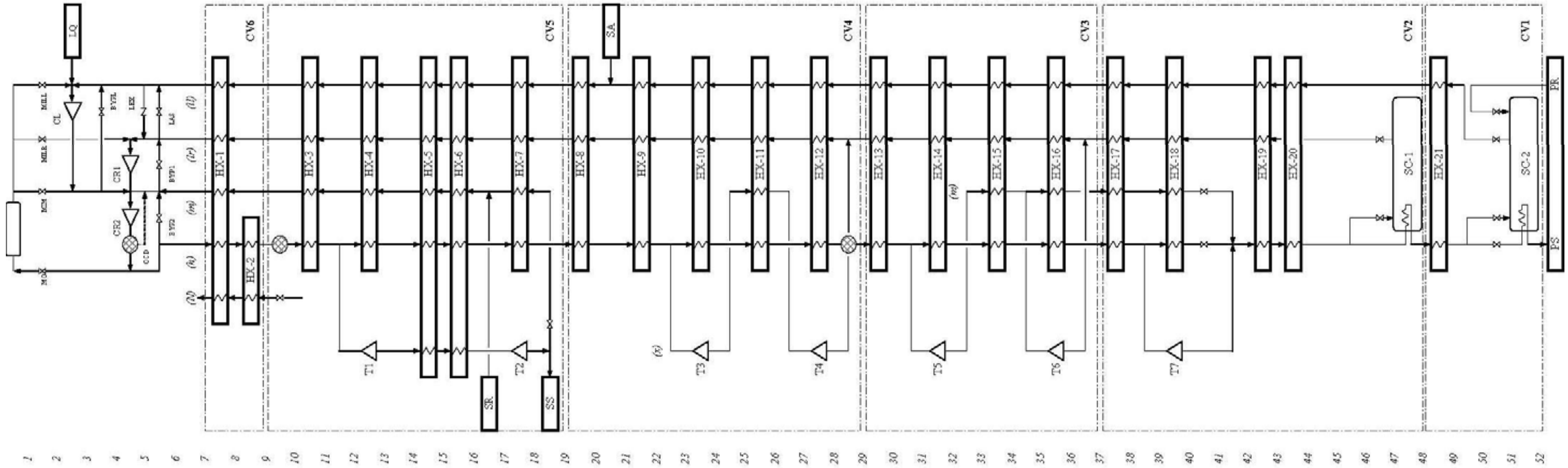
i	$T_{h,i}$ [K]	$T_{l,i}$ [K]	$w_{x,i}$ [g/s]
1	80.00	78.43	147
1'	47.73		
2	35.70	35.00	

Stages 2-6 (Liquefier End)

N	5		*# stages*
T_1	35.70	[K]	*1st stage (h) inlet temperature*
$T_L = T_{N+1}$	4.5	[K]	*Nth stage (l) inlet temperature*
Θ	7.933		*total temperature ratio*
τ	0.020		$= \Delta T_{hl,i} / T_{l,i}$
θ_{hl}	1.020		$= T_{h,i} / T_{l,i}$
θ_i	1.537		*stage temperature ratio* $= T_{h,i} / T_{l,}$
w_L	253.0	[g/s]	*liquefaction flow*
w_R	0	[g/s]	*refrigeration flow*
w_{LR}	253	[g/s]	$= w_L + w_R$
ν_L	1		$= w_L / w_{LR}$
Γ	3.118		*expander pressure ratio*
ϕ	0.400		$= (\gamma-1)/\gamma$
η_x	75.0%		*expander isentropic efficiency*
θ_x	1.378		*expander temperature ratio*
f	0.010		
g	0.507		
h	0.358		
k	0.347		
$\beta_{r,0}$	1.489		$= w_{x,N} / w_{LR}$
β_r	1.029		$= w_{x,i} / w_{x,i+1}$

i	$T_{h,i}$ [K]	$T_{l,i}$ [K]	$w_{x,i}$ [g/s]
2	35.70	35.00	423
2'	31.99		
3	23.69	23.22	411
3'	21.22		
4	15.72	15.41	399
4'	14.08		
5	10.43	10.22	388
5'	9.34		
6	6.92	6.78	377
6'	6.20		
7	4.59	4.50	

C4 – Process Flow Diagram



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C4 - Cycle Analysis Overview

Ganni Cycle Analysis: Overview: Jlab 12GeV Load - Cold Compressors Mode - C4

Energy Balance Violation	NO	Mass Balance Violation?	NO

Loads:

	w	Supply p	Supply T	Return p	Return T	T_{sat}	Δp	q	E_L	*Frac* E_L
	[g/s]	[atm]	[K]	[atm]	[K]	[K]	[atm]	[kW]	[kW]	[-]
Shield	152.5	5.00	35.00	4.50	50.00		0.500	12.00	83.8	6.6%
4K Ref.	0.0	3.00	4.52	1.250	4.47	4.468		0.00	0.0	0.0%
4K Liq.	15.0	3.00	4.52	1.050	298.94				101.9	8.1%
Sub-Atm.	238.0	3.00	4.52	1.19	30.00			37.84	1077.9	85.3%

Total Load Carnot Work: 1263.6 [kW]
Total Carnot Efficiency: 28.3%

Compressors:

		N_c	w_c	w_{byp}	p_S	p_D	p_r	T_S	T_D	η_v	η_i	P_m
		[-]	[g/s]	[g/s]	[atm]	[atm]	[-]	[K]	[K]	[-]	[-]	[kW]
220	1st Load	1	328.3	49.3	1.050	4.30	4.10	298.94	305.00	91.2%	54.8%	557.4
220	1st Recycle	2	832.2	0.4	1.33	4.30	3.24	298.94	305.00	91.5%	57.2%	1130.2
132	2nd Recycle	2	1491.0	0.0	4.30	18.40	4.28	305.00	305.00	86.1%	55.8%	2513.2
				49.7								

Total Input Power: 4200.8 [kW]
Effective Compr. Sys. Eff.: 52.0%

LN$_2$ Pre-Cooling:

w_{LN}	Q	ΔT_{hN7}	ΔT_{hN8}	Supply p	Supply T	Vent p	Vent T	q	E_x	η_{LN2}	COP_{INV}
[g/s]	[gph]	[K]	[K]	[atm]	[K]	[atm]	[K]	[kW]	[kW]	[-]	[-]
133.3	171.9	6.06	4.53	4.00	91.34	1.00	298.94	53.74	92.8	35.0%	4.93

Equiv. LN$_2$ P.C. Input Power: 265.0 [kW]
Availability to Coldbox 2183.6 [kW]
Cold Box Efficency: 57.9%

Control Volumes:

	T.L. #	HP Stream (h) T	HP p	HP w	LP Recycle Stream (r) T	LP (r) p	LP (r) w	LP Recycle Stream (l) T	LP (l) p	LP (l) w	ΔT_{hl}	$(\Delta T_{hl}/T_l)$	Σq_{LK}
		[K]	[atm]	[g/s]	[K]	[atm]	[g/s]	[K]	[atm]	[g/s]	[K]	[-]	[W]
	7	305.00	18.00	1481.0	298.94	1.33	831.8	298.94	1.05	264.0	6.060	0.020	
CV6													500
	9	80.49	17.90	1481.0	78.89	1.40	831.8	78.89	1.12	264.0	1.600	0.020	
CV5													600
	19	35.70	17.52	1110.8	35.00	1.46	831.8	35.00	1.18	264.0	0.700	0.020	
CV4													450
	29	15.79	17.38	719.8	15.46	1.50	440.8	15.46	1.22	26.0	0.325	0.021	
CV3													300
	37	7.52	17.34	372.6	7.38	1.52	93.5	7.38	1.24	26.0	0.148	0.020	
CV2													355
	48	4.75	3.00	279.0				4.66	1.25	26.0	0.093	0.020	
CV1													205
	52	4.52	3.00	253.0				4.47	1.25	0.0	0.049		

Expanders:

#	w_x	Δp_i	p_i	p_o	p_r	T_i	T_o	T_r	Δh_x	η_x	W_x	ϕ_x	$\phi_{x,D}$	% $\Delta\phi_x$
	[g/s]	[atm]	[atm]	[atm]	[-]	[K]	[K]	[-]	[J/g]	[-]	[kW]	[-]	[-]	[-]
1	370.2	0.10	17.48	13.72	1.27	60.57	56.42	1.07	22.57	75.0%	8.35	153.6	153.6	0.0%
2	370.2	0.10	13.54	5.05	2.68	46.30	35.00	1.32	60.23	75.0%	22.30	115.4	115.4	0.0%
3	391.0	0.10	17.41	5.50	3.16	30.64	22.04	1.39	44.96	75.0%	17.58	75.7	75.7	0.0%
4	391.0	0.10	5.32	1.55	3.43	22.03	15.47	1.42	33.17	75.0%	12.97	206.6	206.6	0.0%
5	347.2	0.10	17.26	5.50	3.14	15.78	11.05	1.43	20.56	75.0%	7.14	45.7	45.7	0.0%
6	347.2	0.10	5.34	1.57	3.40	10.94	7.40	1.48	14.34	75.0%	4.98	120.4	120.4	0.0%
7	372.6	0.10	17.24	3.06	5.64	6.65	5.414	1.23	7.73	70.0%	2.88	25.3	25.3	0.0%

Total Expander Work: 76.20 [kW]

Heat Exchangers:

	HX #	*W.E.* $(\Delta T_{hl}/T_l)$	*C.E.* $(\Delta T_{hl}/T_l)$	q_h	NTU	ε	*(Calc.)* (UA)	*(Proj.)* $(UA)_P$	%Δ(UA)	*Ref.* $(UA)_{ref}$
		[-]	[-]	[kW]	[-]	[-]	[kW/K]	[W/K]	[-]	[W/K]
CV6	*HX-1*	0.020	0.057	1675.45	42.16	0.980	318.75			937.44
	HX-1N	0.020	0.057	31.00	42.16	0.980	5.90			271.35
	HX-2	0.057	0.020	22.74	1.04	0.647	8.08			92.36
CV5	*HX-3*	0.021	0.035	155.35	11.09	0.925	84.56			246.87
	HX-4	0.035	0.020	24.44	2.71	0.789	15.97			56.65
	HX-5	0.020	0.029	66.26	7.06	0.887	53.90			157.36
	HX-6	0.029	0.045	13.42	1.41	0.633	8.05			23.51
	HX-7	0.045	0.020	63.70	8.53	0.938	51.23			175.74
CV4	*HX-8*	0.020	0.034	28.59	5.86	0.877	33.56			123.50
	HX-9	0.034	0.040	2.41	0.49	0.344	2.19			8.94
	HX-10	0.040	0.020	35.39	11.52	0.952	47.38			229.29
	HX-11	0.020	0.021	0.06	0.03	0.027	0.13			0.54
	HX-12	0.021	0.020	27.84	16.71	0.953	74.30			359.58
CV3	*HX-13*	0.021	0.022	0.05	0.06	0.057	0.15			0.95
	HX-14	0.022	0.040	12.20	12.56	0.934	31.73			255.28
	HX-15	0.040	0.049	0.41	0.34	0.269	0.87			5.46
	HX-16	0.049	0.020	8.17	11.74	0.959	27.90			224.54
CV2	*HX-17*	0.020	0.242	1.58	3.84	0.932	2.99			45.63
	HX-18	0.242	0.020	0.05	N/D	N/D	N/D			N/D
	HX-19	0.020	0.121	0.67	2.14	0.846	2.42			36.92
	HX-20	0.121	0.131	0.01	0.07	0.071	0.02			0.70
	SC-1	0.120	*0.011*	1.23	2.43	0.912	5.79			125.37
CV1	*HX-21*	0.020	0.057	0.05	1.19	0.673	0.31			12.97
	SC-2	0.057	*0.011*	0.46	1.63	0.805	3.64			164.72
	Min.	0.020	0.020	0.01		0.027				0.54
	Max.	0.242	0.242			0.980				937.44
	Total			2171.51	144.61		779.82			
	MIN $(\Delta T_{hl}/T_l)$	0.020			ε_{max}	0.980				
			300-80K	1729.19	43.20		332.73			
			80-35K	323.16	30.80		213.72			
			35-4.5K	119.16	70.62		233.37			

Dewars:

	w_{JT}	w_{BO}	w_{la}	w_a	p	T	$\Delta T_{hl,c}$	*x*
	[g/s]	[g/s]	[g/s]	[g/s]	[atm]	[K]	[K]	[-]
HX-2	133.3	115.6			1.200	78.89	1.60	13.3%
SC-1	93.5	73.1	0.0	0.0	1.527	4.70	0.05	21.9%
SC-2	26.0	24.1	0.0	0.0	1.250	4.47	0.05	7.6%

T5 Bypass:

$\xi_{x5,b}$	$w_{h,29}$
[-]	[g/s]
0.00%	726.4

T6 Bypass:

$\xi_{m6,b}$	$\Delta p_{m40,41}$
[-]	[atm]
0.00%	2.411

T7 Bypass:

$\xi_{h7,b}$	$T_{h,39}$	$T_{h,42}$	$(T_{h,40} - T_{x,41})$ MAX	$(T_{h,40} - T_{x,41})$ INPUT
[-]	[K]	[K]	[K]	[K]
0.00%	8.00	5.40	0.0000	0.0000

JLab 12 GeV Helium Refrigerator Cycle Studies

- **Examined four configurations of Jlab floating pressure cycle.**
  - **All configurations use (5) expander stages between 35 & 4.5K.**
  - **All configurations use LN_2 pre-cooling.**
  - **C1: (2) expanders between 80 & 35K. HP to shield expander (T2), T2 exhaust to T1 (warmer expander), T1 exhaust to LP recycle stream (LR).**
  - **C2: (2) expanders between 80 & 35K. HP to T1, T1 exhaust to shield expander (T2), T2 exhaust to LR.**
  - **C3: (1) expander between 80 & 35K. Shield expander (T1) exhaust to MP stream.**
  - **C4: (2) expanders between 80 & 35K. HP to T1, T1 exhaust to shield expander (T2), T2 exhaust to MP stream.**

Cycle Configuration Summary

Jlab 12 GeV Cycle Summary

		Cycle Configuration			
		1	**2**	**3**	**4**
Total Carnot Efficiency		27.6%	28.0%	28.0%	28.3%
Effective Compressor Sys. Eff.		51.7%	51.7%	52.0%	52.0%
Cold Box Efficiency		56.4%	57.1%	57.3%	57.9%
Load Carnot Work	[MW]	1.27	1.28	1.27	1.26
Total Input Power	[MW]	4.34	4.35	4.25	4.20
Availability to CBX	[MW]	2.24	2.25	2.21	2.18
LN_2 Requirement	[gph]	158	158	176	172
Load Compressor		(1) 220	(1) 220	(1) 220	(1) 220
1st Stage Recycle Compressor		(3) 193	(3) 193	(2) 220	(2) 220
2^{nd} Stage Recycle Compressor		(2) 132	(2) 132	(2) 132	(2) 132
Total Number Turbine Expanders		7	7	6	7
HP Flow to CBX	[g/s]	1361	1364	1515	1481
CBX MP Return?		NO	NO	YES	YES
Load Compressor Suction	[atm]	1.05	1.05	1.05	1.05
SC-1 (HP Pot) Pressure	[atm]	1.56	1.56	1.53	1.53
2^{nd} Stage Recycle Suction	[atm]	3.98	3.99	4.39	4.30
Shield Return Pressure	[atm]	4.21	1.47	4.59	4.50
T1 Enthapy Drop	[J/g]	80.0	72.7	76.3	22.6
T1 Flow Coefficient	[-]	311	70	99	154
Shield Expander Enthalpy Drop	[J/g]	80.0	80.0	45.0	60.2
Shield Expander Flow Coefficient	[-]	61	139	76	115
Total Expander Work	[kW]	84.6	82.2	76.4	76.2
Total HX NTU's	[-]	136	139	140	145
300 - 80K	[-]	35	37	43	43
80 - 35K	[-]	30	31	26	31
35 - 4.5K	[-]	70	71	71	71
Total HX (UA)	[kW/K]	681	691	756	780
300 - 80K	[kW/K]	249	262	339	333
80 - 35K	[kW/K]	199	191	183	214
35 - 4.5K	[kW/K]	233	238	234	233

Cycle Configuration Summary

Notes:

(1) JLab shield supply transfer-line relief valve S.P. 7.5 atm. & shield return relief valve S.P. 6.5 atm.

(2) Compressor frame: Howden WRVi 321 mm diameter rotors; no. of compressors in parenthesis and (L/D) ratio adjacent.

(3) Equivalent LN_2 power input calculated using 35% Carnot efficiency (for LN_2 Carnot work).

(4) Maximum allowed turbine enthalpy drop for cycle configurations is 80 J/g. This is not necessarily thermodynamically optimum.

(5) Expander flow coefficients are for magnitude comparison only. The actual value will be set by the selected wheel size.

(6) LN_2 flow based on saturated liquid at 4 atm.

(7) Compressor volumetric and isothermal efficiencies predicted using SSC data (ASST Compressor Data Analysis, 19-Jan-93).

(8) Assumed 1st stage motor efficiency of 94.5% operating at 3550 rpm; and, 2nd stage motor efficency of 96.7% operating at 3550 rpm.

(9) Assume 0.4 atm. pressure drop across main oil removal and 10 g/s of helium bypass (drain) back to MP stream.

(10) Total HP Δp through HX's 0.3 atm. plus 0.3 atm. & 0.1 atm. for 80K & 20K bed's, respectively. Total LP stream Dp 0.2 atm. (both recycle and load return).

(11) Used 75% adiabatic efficiency for all expanders except the coldest ('wet') expander (70%).

(12) Minimum allowed HX ($\Delta T/T$) ratio is 2% and maximum effectiveness allowed is 98%.

Cycle Configuration Summary

- **Appears that configuration C4 is best.**
  - **—Estimated 28.3% Carnot efficiency for the 4.5K refrigerator system.**
  - **—Requires:**
    - **(1) Load compressor: 321/220 (~ 0.56 MW)**
    - **(2) 1st Stage Recycle compressors: 321/220 (~ 0.57 MW each)**
    - **(2) 2nd stage Recycle compressors: 321/132 (~1.3 MW each)**
    - **Total input power ~4.2 MW, with ~172 gph LN_2 consumption.**
    - **(7) Expansion stages.**

Comparison with Other Comparable Sized Cycles

Comparison of Other Similar Size (18kW 4.5K Equivalent) Helium Refrigerators to JLab 12GeV Refrigerator Addition														
		Notes:	**CEBAF (design)** *[1]*	**CEBAF (original)** *[2]*	**CEBAF (present)** *[3]*	**CERN (Linde)** *[4]*	**CERN (AirLiq)** *[5]*	**SNS (design)** *[6]*	**SNS (original)** *[7]*	**SNS (present)** *[8]*	**JLab - C1 (Design)**	**JLab - C2 (Design)**	**JLab - C3 (Design)**	**JLab - C4 (Design)**
Year			1988	1992	2006	2002	2002	2002	2004	2006	2007	2007	2007	2007
Carnot Load	[MW]		1.22	1.04	1.01	1.26	1.27	0.75	0.65	0.69	1.27	1.28	1.27	1.26
Availability to Cold Box	[MW]	*[a]*	2.13	2.11	2.09	2.18	*N/A*	1.33	1.24	1.54	2.24	2.25	2.21	2.18
Equivalent LN_2 PC Input Power	[MW]	*[b]*	0.22	0.43	0.58	*N/A*	*N/A*	0.23	0.37	0.42	0.24	0.24	0.27	0.27
Compressor Input Power	[MW]		3.88	5.28	4.10	3.96	4.47	2.66	3.51	3.51	4.34	4.35	4.25	4.20
Est. Operating (2.1K) Carnot Eff.	[-]	*[c]*	18.5%	11.2%	13.0%	*N/A*	*N/A*	14.0%	10.0%	11.4%	16.6%	16.6%	16.9%	17.1%
Total 4.5K Carnot Efficiency	[-]	*[d]*	29.8%	18.2%	21.6%	31.7%	28.5%	25.8%	16.8%	17.7%	27.6%	28.0%	28.0%	28.3%
Cold Box Efficiency	[-]	*[d]*	57.3%	49.4%	48.5%	57.7%	*N/A*	56.1%	52.7%	45.0%	56.4%	57.1%	57.3%	57.9%
Effective Compressor System Eff.	[-]	*[d]*	55.0%	39.9%	50.9%	55.0%	*N/A*	50.0%	35.3%	43.9%	51.7%	51.7%	52.0%	52.0%
Cold Compressor Flow	[g/s]	*[e]*	237	200	190	194	194	126	121	140	238	238	238	238
HP Supply Flow to Cold Box	[g/s]		1578	1410	1552	1680	1680	1149	1092	1077	1361	1364	1515	1481
Discharge Pressure	[atm]		20.4	20.6	17.7	19.7	19.7	16.6	16.9	16.9	18.0	18.0	18.0	18.0
LN_2 Usage	[g/s]		109	216	290	*N/A*	*N/A*	117	184	210	123	123	136	133
No. of Compressor Skids (1^{st} / 2^{nd})	[-]		(2 / 3)	(2 / 3)	(2 / 2.5)	(3 / 2)	(3 / 2)	(2 / 2)	(2 / 2)	(2 / 2)	(4 / 2)	(4 / 2)	(3 / 2)	(3 / 2)
No. Expansion Stages	[-]	*[f]*	4	4	4	5	6	4	4	4	7	7	6	7
No. Turbines	[-]	*[f]*	4	4	4	10	8	5	5	5	*TBD*	*TBD*	*TBD*	*TBD*

Comparison with Other Comparable Sized Cycles

Notes:
[1] Effective compressor system efficiency from CEBAF Revised Design Handbook (Jan. 14, 1991), Table 11.4.1., p. 11.4-5.
[2] Unable to achieve pump-down with MP stream at 3.24 atm. due to inadequate 4.5K cold box capacity. This is when MP stream was reduced to 2.8 atm. and a pump-down was achieved.
[3] Cold compressor flow 190-195 g/s for FEL plus (5 pass) 5 GeV operations. Also supplying 18 g/s of 4.5K liquefaction. MP at 2.75 atm.
[4] Refer to LHC Project Report 796 (2004). Availability to cold box based on Linde cycle design (refer to Adv. Cryo. Eng., v45, p.1285); testing data not available. **Compressor efficiency cannot be verified against reported required power input**. For stated power input, 1st and 2nd stage compressors must have AT LEAST 59% and 57% overall isothermal efficiency, respectively.
[5] Refer to LHC Project Report 796 (2004). Process conditions for cycle design or testing not available.
[6] This refers to the Linde design; not to the 'Jlab Reference Refrigeration Cycle' (Appendix B of the Jlab-SNS 4.5K Cold Box specification). However, Linde was not responsible for the compressor system design (given to PHPK). <u>*An effective compressor system efficiency of 50% has been assumed*</u>.
[7] Maximum plant capacity at Linde TS conditions for refrigeration mode. Plant only able to support 19% of shield load at TS MP (4 atm.). Used input power from 'SNS (present)' data since the same number of 1st & 2nd stage compressors are running and the LP, MP & HP stream pressures are very close to the same.
[8] MP reduced to ~2.8 atm. to increase CBX availability to meet maximum load and prevent excessive oil carry-over from compressor bulk oil separators.
[a] This is the total exergy provided to the cold box from the compressor system.
[b] Except for the JLab C1-C4 cycles, the equivalent LN_2 pre-cooling input power (P_{LN}) is based on an estimate of 2 kW per 1 g/s of LN_2 (approx. 35% Carnot efficiency of net exergy).
[c] Estimated total 2.1K Carnot efficiency calculated assuming a nominal 2K COP of 160 and a 2K load supply interaface enthalpy difference of 20 J/g. So, $\eta_{2K} = COP^*\Delta h^* w_{CC} / (P_T + P_{LN})$. With COP = 160, Δh = 20 J/g and w_{CC} the cold compressor flow. CERN was not included since their system is at 1.8K.
[d] η - Total Carnot Efficiency, η_{CBX} - Cold Box Efficiency, $\eta_{C.sys}$ - Effective Compressor System Efficiency
E_L - Carnot Load [MW], P_T - Total (Electric) Power Input, P_{LN} - Equivalent Power for LN_2 P.C., E_{CBX} - Cold Box Availability, $E_{C.sys}$ - Compressor System Carnot Input Power
If the liquefaction load returns to the compressor system at the 1st stage suction pressure and it is not a large fraction of the total Carnot Load, then $E_{C.sys} \approx E_{CBX}$
$\eta = E_L / (P_T + P_{LN})$, $\eta_{CBX} = E_L / E_{CBX}$, $\eta_{C.sys} = E_{C.sys} / P_T \approx E_{CBX} / P_T$ <u>Note</u>: $P_{LN} = 0$ for turbine pre-cooling (300-80K)
[e] CERN operates at 1.8K with a nominal 41 g/s of liquefaction load.
[f] The number of expansion stages is not necessarily equal to the number of turbines. String turbines without a heat exchanger in between are considered one expansion stage.

Comparison Conclusion

- **From the comparison the following appear achievable:**
  - **~57% Cold Box Efficiency**
  - **~52% Overall Compressor System Efficiency**
  - **~28% Total Carnot (4.5K) Efficiency (including LN_2 pre-cooling)**

- **WELL MATCHED compressor system and cold box ARE ESSENTIAL for obtaining high total Carnot efficiency**

- **It is important to understand, before the design is finalized, how efficient a given cycle is at various off-design conditions (100% liquefier, reduced capacity modes, etc.).**
  - **—From this aspect, the JLab floating pressure cycle is believed to perform superior to any other known cycle.**

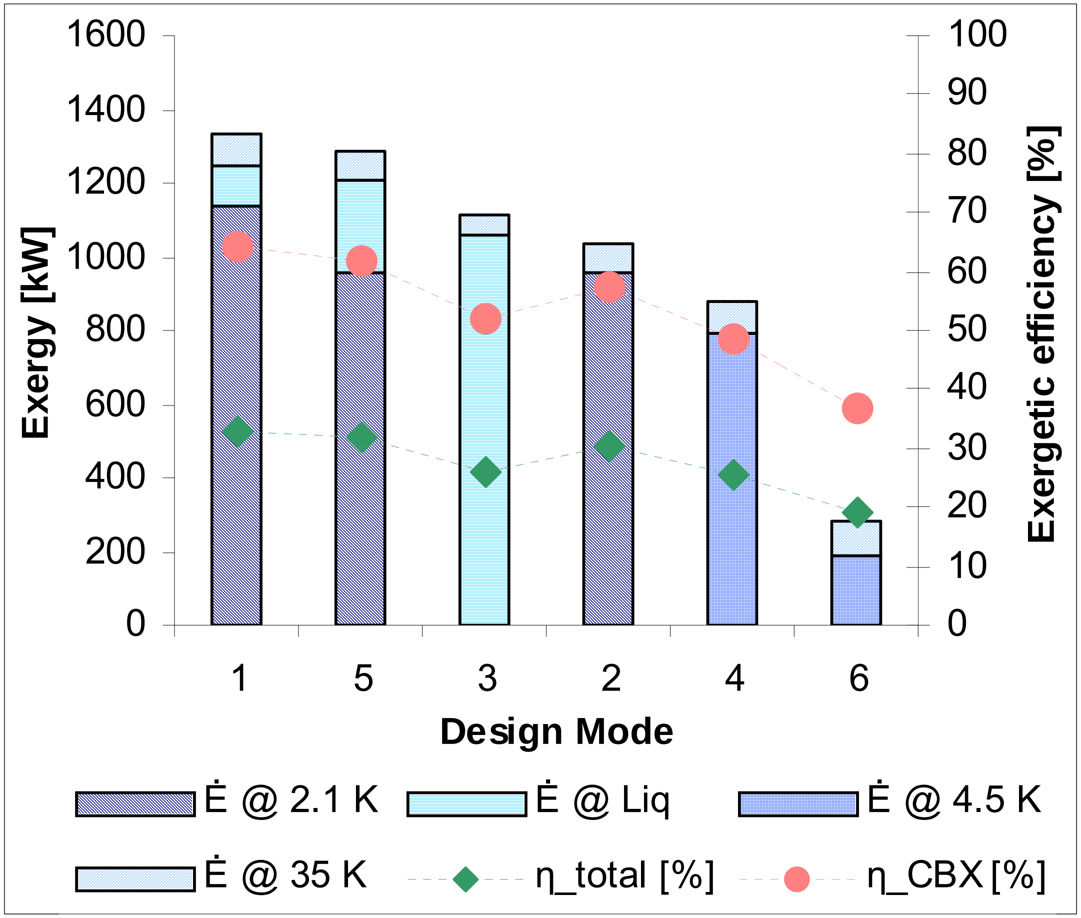
CHL-2 Design Modes of Operation

#	Design Mode	Load @ 2 K [g/s] *	Load @ 4.5 K [kW]	Liquefaction [g/s]	Load @ 35 K-55K [kW]
1	Maximum capacity (CBX supporting maximum cold compressor operation)	>238	0	>15	>12
2	Nominal capacity (CBX supporting nominal cold compressor operation)	>200	0	0	>7.5
3	Maximum 4.5-K liquefaction	0	0	>150	>7.5
4	Maximum 4.5-K refrigeration	0	>10.5	0	>12
5	Maximum fill (of Linac cryo-modules)	>200	0	>35	>12
6	Stand-by 4.5-K refrigeration**	0	>2.5	0	>12

•Load at 2.1 K means supply flow at 3.2 bar 4.5 K, with return flow at 1.2 bar 30 K

** Mode 6 requires a minimum amount of rotating equipment while supporting the LINAC loads at 4.5-K.

CHL-2 4.5K System Projected Efficiencies



10. Optimal Operation of the Existing Helium Refrigeration Systems

Generally helium refrigeration systems are designed to operate at one maximum capacity operating point.

In practice, the system capacity requirement often varies depending on the load characteristics, distribution system insulating vacuum pressure, experimental setup among other factors.

Operating the system at the maximum design point may not be advantageous when the full capacity is unnecessary or the required mode of operation has changed.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

Optimal operation addresses the following goals:

- **The design TS diagram parameters.**
- **The present loads.**

  Again, the same five questions of

  ***System Optimization,* need to be answered.**
- **Any modifications (may be as simple as a control philosophy change) to the system to fit the present load's operating conditions and current optimal goals.**

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

Normally TS diagrams are developed for a maximum design capacity.

Many systems are unnecessarily continuously operated at the design TS maximum capacity by wasting capacity with throttled valves, adding heater load and/or bypassing the compressor capacity.

***These methods are analogous to driving a car with a fully depressed gas pedal while controlling the actual speed of the car with the foot brake*.**

Helium refrigeration systems unnecessarily operating at the maximum design capacity not only use additional utilities (electric power, LN2, cooling water), but operate the components at higher stress, that often result in additional maintenance costs and down time.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

Operating plants can avoid these problems by incorporating the variable pressure(s) control philosophy described in earlier chapters.

<u>That method works similar to the variable gas pedal depression controlling the speed of the car.</u>

The difference between the analysis of a new design and an existing system is that the ability to select components to meet the requirements of the design case is constrained for the existing system.

TS diagrams are developed during the system design phase to select the operating process and define the process design requirements for the major components.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

The design TS diagram for the existing system is of ***limited*** use and often misleads less experienced users since the load requirements (characteristics) may be changed from the original system design or the system components may not have been optimally selected to meet the design TS diagram goals in the first place.

If no manufacturer design data is available to calculate the device flow at different operating conditions, the design TS diagrams can be used to establish these flow characteristics as a last resort.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

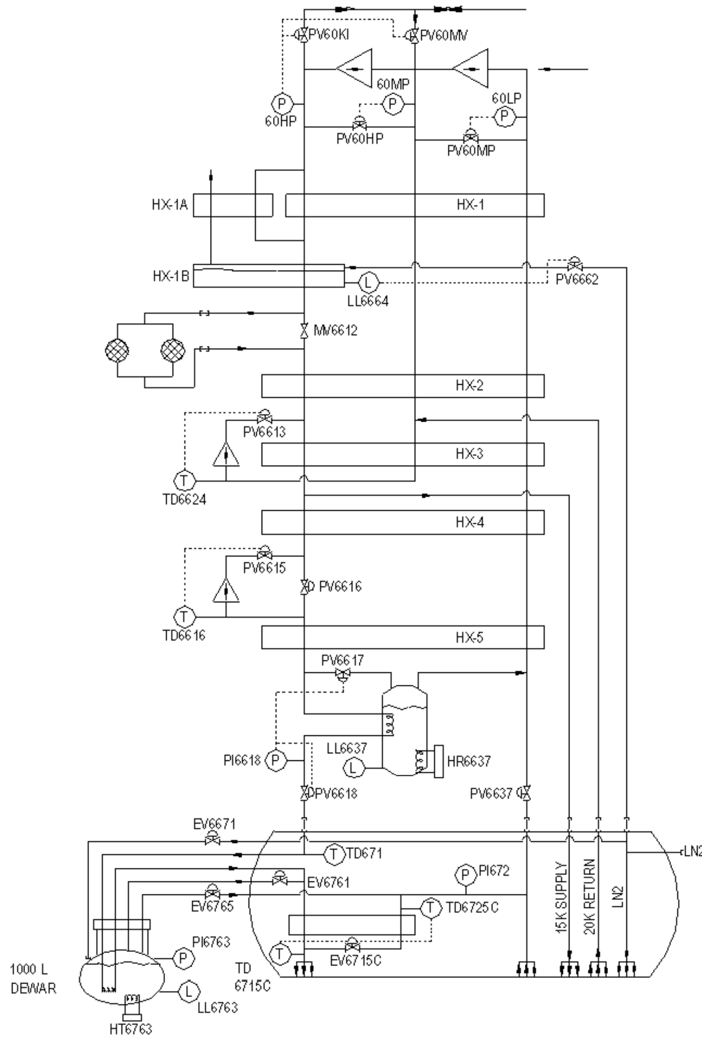
The Variable pressure operation key factors important in the design phase are:

- **The oil removal system should be designed to work at the minimum pressure required by the cold box for efficient minimum capacity operation.**
- **A varying liquid inventory (dewar) to establish the system pressure required to meet the load demand.**

We will look into some practical examples of existing systems modified in this way.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

End Station Refrigeration System (ESR) at JLab



Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

This system, originally designed for ESCAR experiment at University of California, Berkley (LBL), was relocated to JLab in 1993.

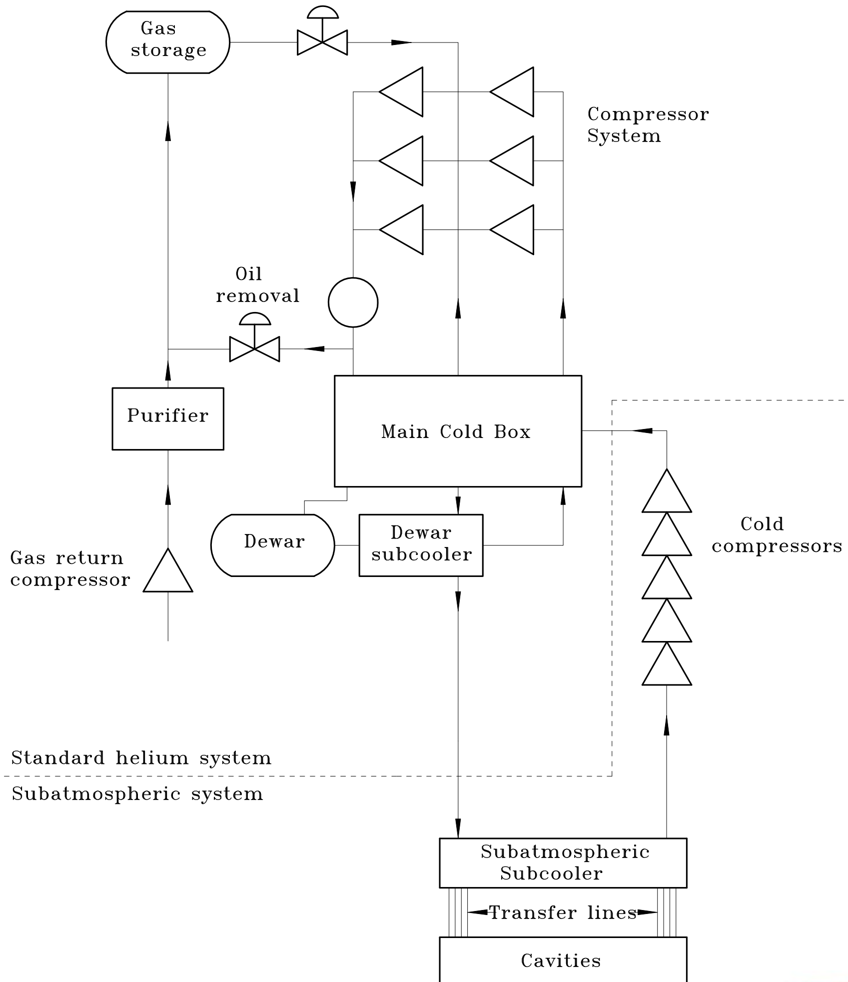
- It went through some equipment modifications before it was commissioned [18] at JLab in 1994 to include 80K beds, a separate LN2 stream cooling HX and a new 4.5K phase separator in the cold box.
- The original gas management and the cold box controls were based on single point load (maximum capacity) operation

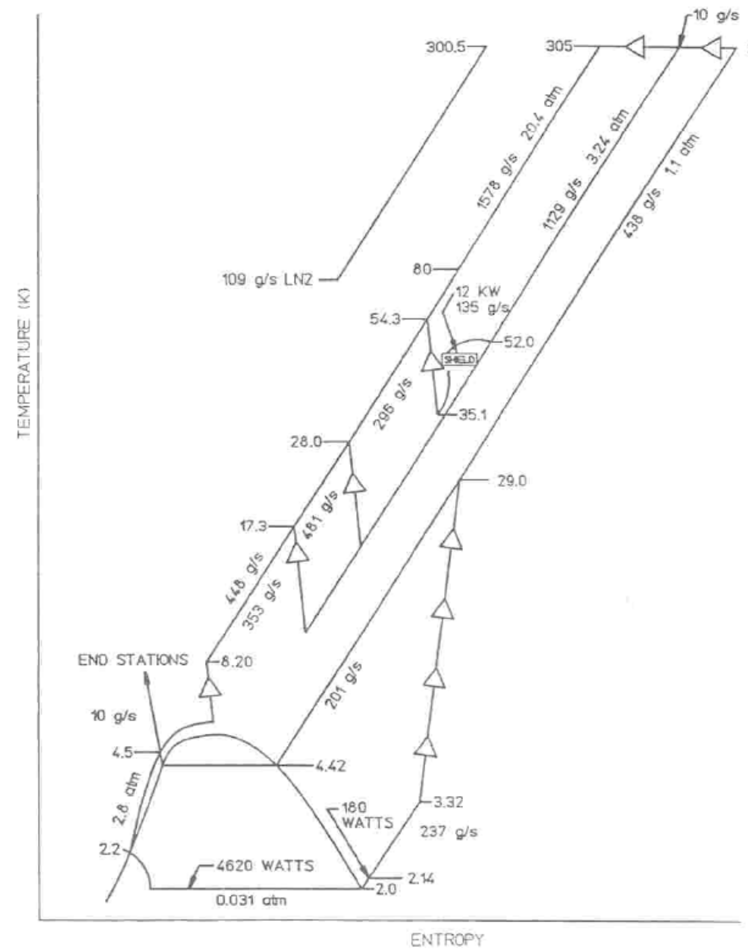
The system supports the experimental hall (end station) loads which vary in time depending on the number of magnets and the targets on line and the condition of the loads

The ESR has the highest reliability among all JLab cryogenic systems. This system has been operating continuously (24/7) for the past 10 years with an availability greater than 99%. Since it uses the originally installed variable system pressures control, stresses on the system components are routinely reduced.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

Central Helium Liquefier (CHL) at JLab





Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

The system was originally designed as shown on the above TS diagram

The capacity, efficiency and the operating parameters of both the 4K and the 2K cold boxes proved to be lower than the original design goals.

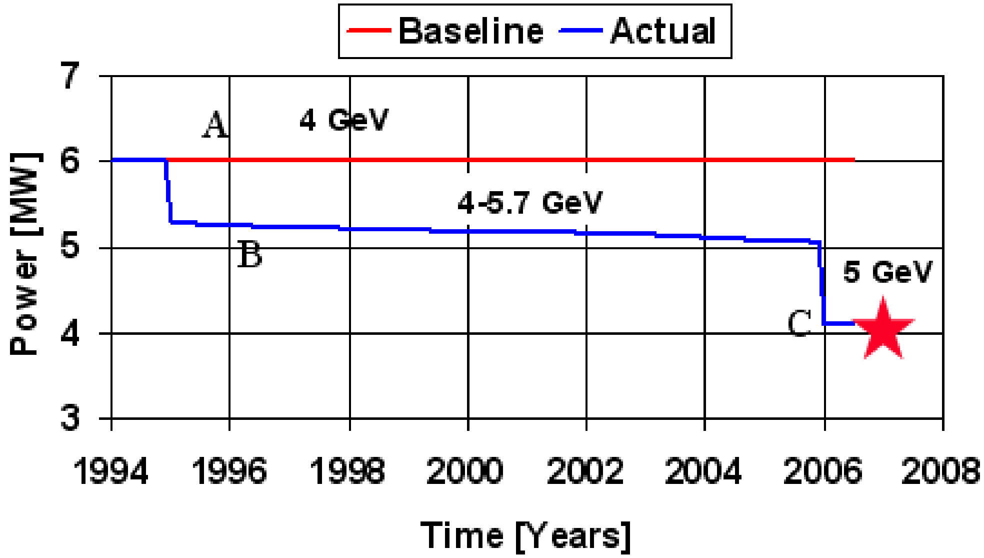
Initially, the system was operated close to the original design TS diagram.

Accounting for the modified components capacity and system performance, modifying the system operation resulted in a reduction of input power by ~ 1 MW and increased refrigeration capacity.

The original 2K cold box proved to be less efficient than design and extremely unstable.

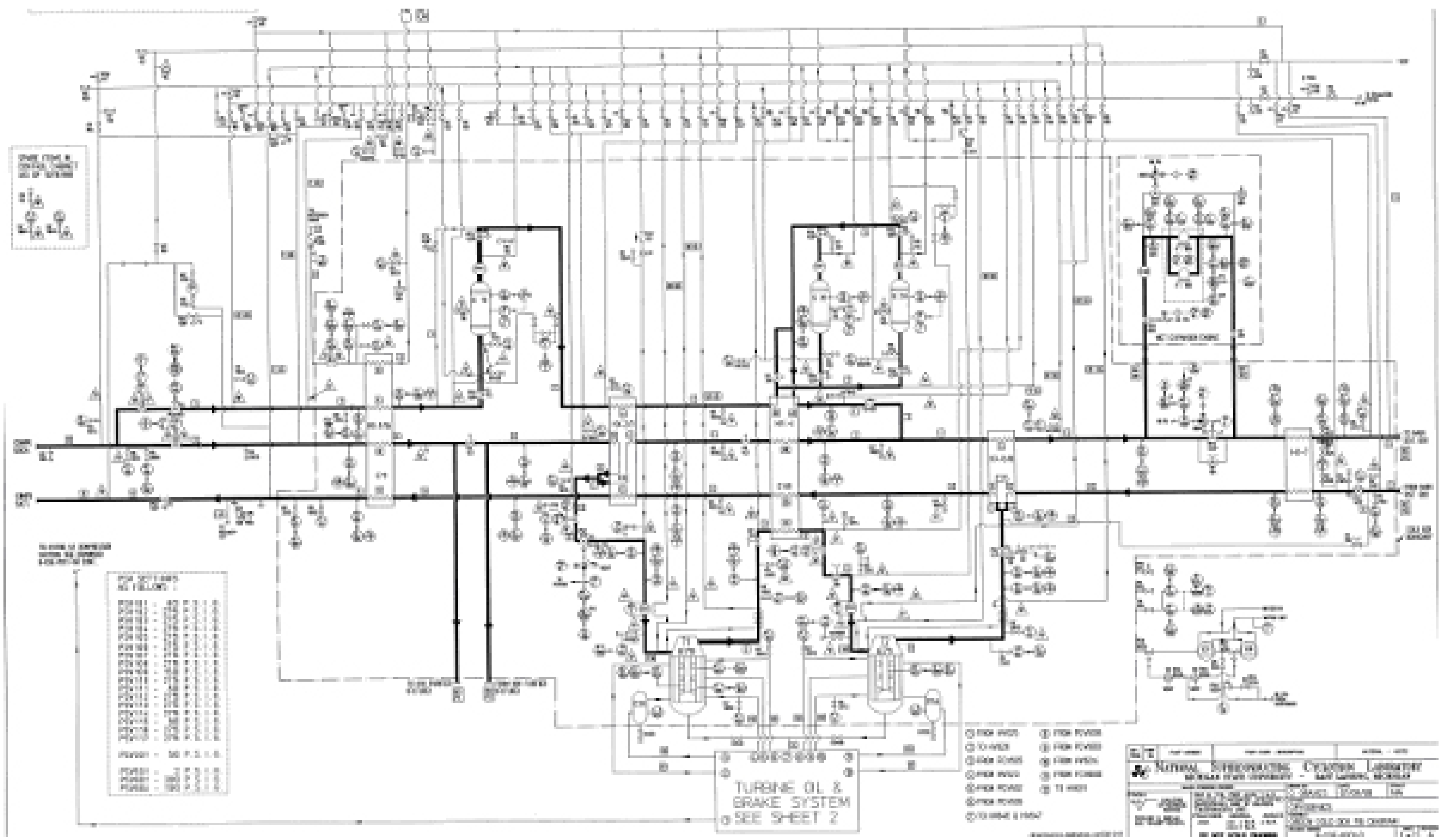
The new 2K cold box designed by JLab improved the 2K capacity by ~10%, increased its stability, and gave some insight to the cold compressors' variable frequency motor torque and other component limitations.

Optimal Operation of JLab-CHL-1 Helium Refrigeration System



This cryogenic plant supports operation of the Continuous Electron Beam Accelerator Facility (CEBAF) cryomodules in the tunnel. The accelerator power is adjustable from 500MeV to 6GeV but the original cryogenic plant was designed to operate only at one design capacity consuming more than 6MW of electrical power. Through the years the Cryogenics Group has completed several phases of technological improvement and increased the plants operational envelope to allow its capacity to be varied to better match the cryogenic load. The operational envelope now allows the plants power consumption to be varied from 4.2MW up to 6MW in conjunction with the CEBAF accelerator requirements.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)



Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

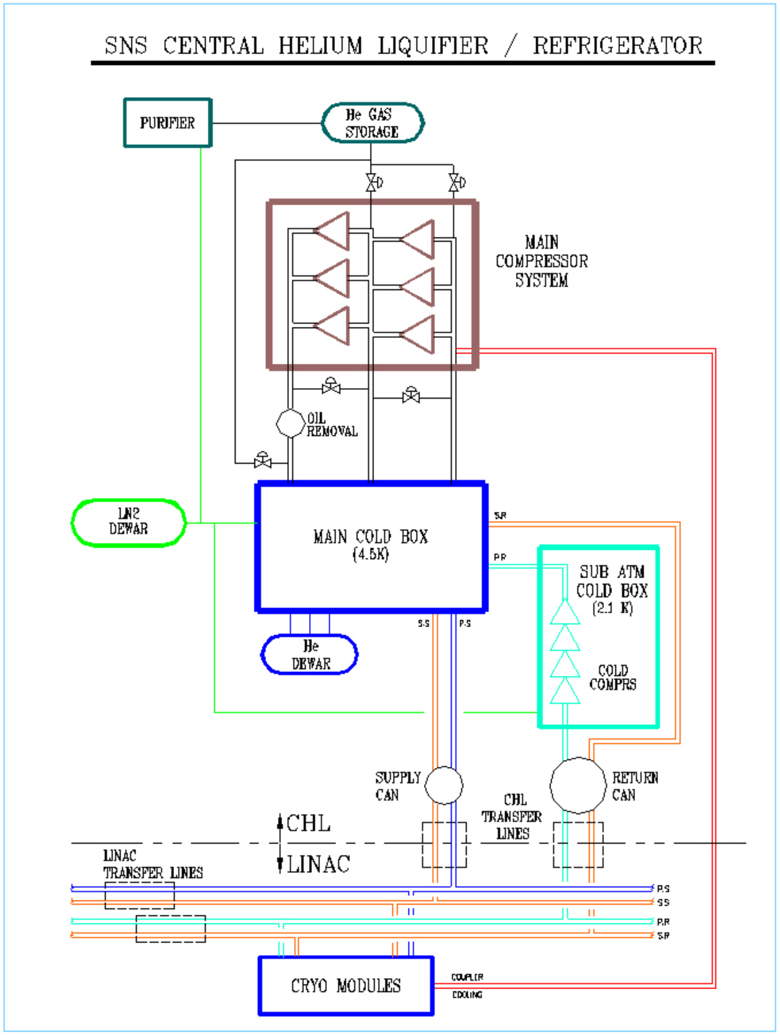
The Cryogenic System Upgrade for the National Superconducting Cyclotron Laboratory:

The MSU refrigerator was originally designed as a liquefier for Bureau of Mines [22] in Amarillo, TX (1979).

The original pure liquefier system design has been arranged to operate efficiently as a primary refrigerator over varying load requirements and to support a combination of refrigeration and liquefaction loads.

The maximum system pressure follows the load requirement, reducing the input utilities for reduced loads as well as reducing the wear and tear on the equipment. This system has been operating continuously for the past four years with more than 99% availability.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)



Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

SNS is the second JLab cryogenic team project to design, procure, fabricate equipment and support an installation outside JLab.

JLab was responsible for all the cryogenic system design aspects of the project. The SNS cryogenic system is Operating Continuously from 2005

The system is presently set to operate at approximately optimum conditions for the majority of the operating modes by utilizing the previously explained optimal operational concepts.

The SNS system would have used 3.8 MW of equivalent input power with out the floating pressure technology and it can be turn down to ~2.7 MW of equivalent input power or in between based on the refrigeration needs of the accelerator.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

Brookhaven National Lab (BNL):

This refrigeration system was originally designed for the Isabelle project with a capacity of 24.8 kW@ 3.8 K without LN2 pre-cooling and capable of supporting some 2.5K temperature operations.

With only minor modifications and using the original cold box T-S diagram and control philosophy as a starting point, it was adapted for the RHIC accelerator system requiring less than a third of the system refrigeration capacity and operating at 4.5K. It utilized the original compressor gas management system at the design system pressures of ~16 atm, requiring ~9.4 MW of input power to the compressor system Figure 1

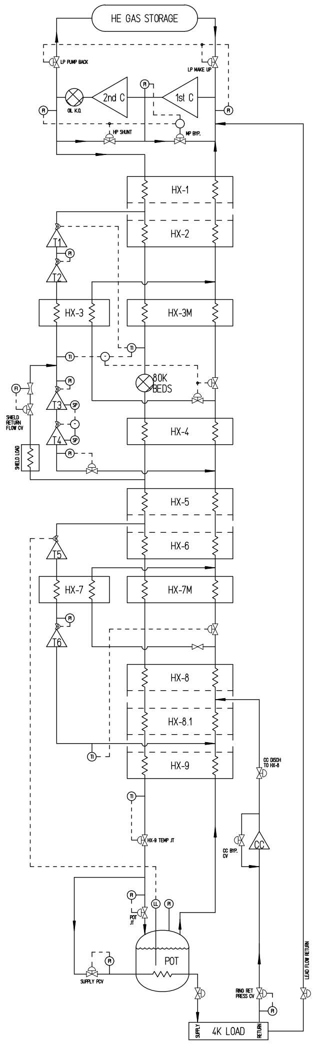
Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

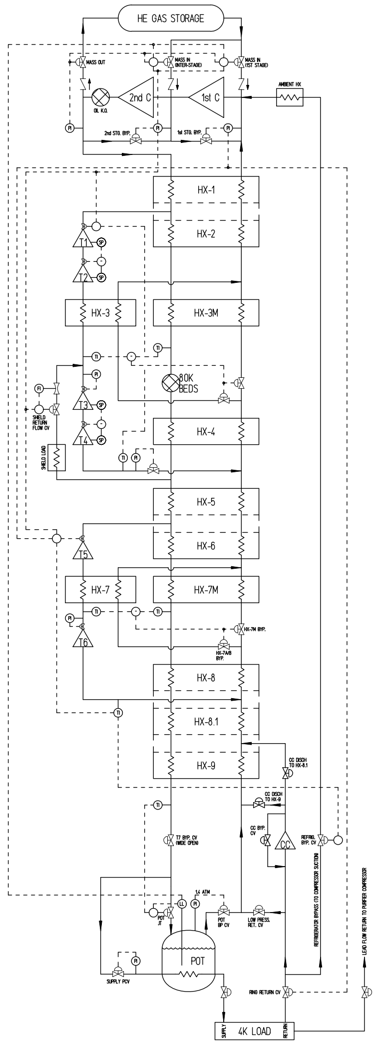
Brookhaven National Lab (BNL) (Cont.)

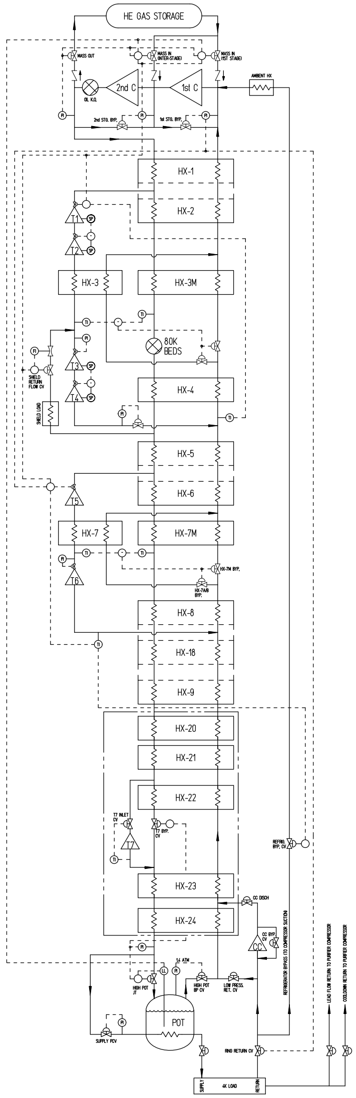
A JLab/BNL team recognized the original T-S diagram process conditions for the RHIC loads resulted in a non-optimum refrigerator cold end performance. After this discovery, BNL began a series of modifications to the cold end piping, its cycle operating temperatures and modified the gas management system (as explained in Chapter 8), that resulted in a substantial, ~ 2MW reduction of input power Figure 2.

Phase-III of the JLab/BNL project is currently in progress and anticipated to further reduce the input power and improve the system's capacity, efficiency, stability, operational flexibility, reliability and availability. A process diagram of the proposed concept for the next Phase is shown in Figure 3.

Optimal Operation of the Existing Helium Refrigeration Systems (cont.)

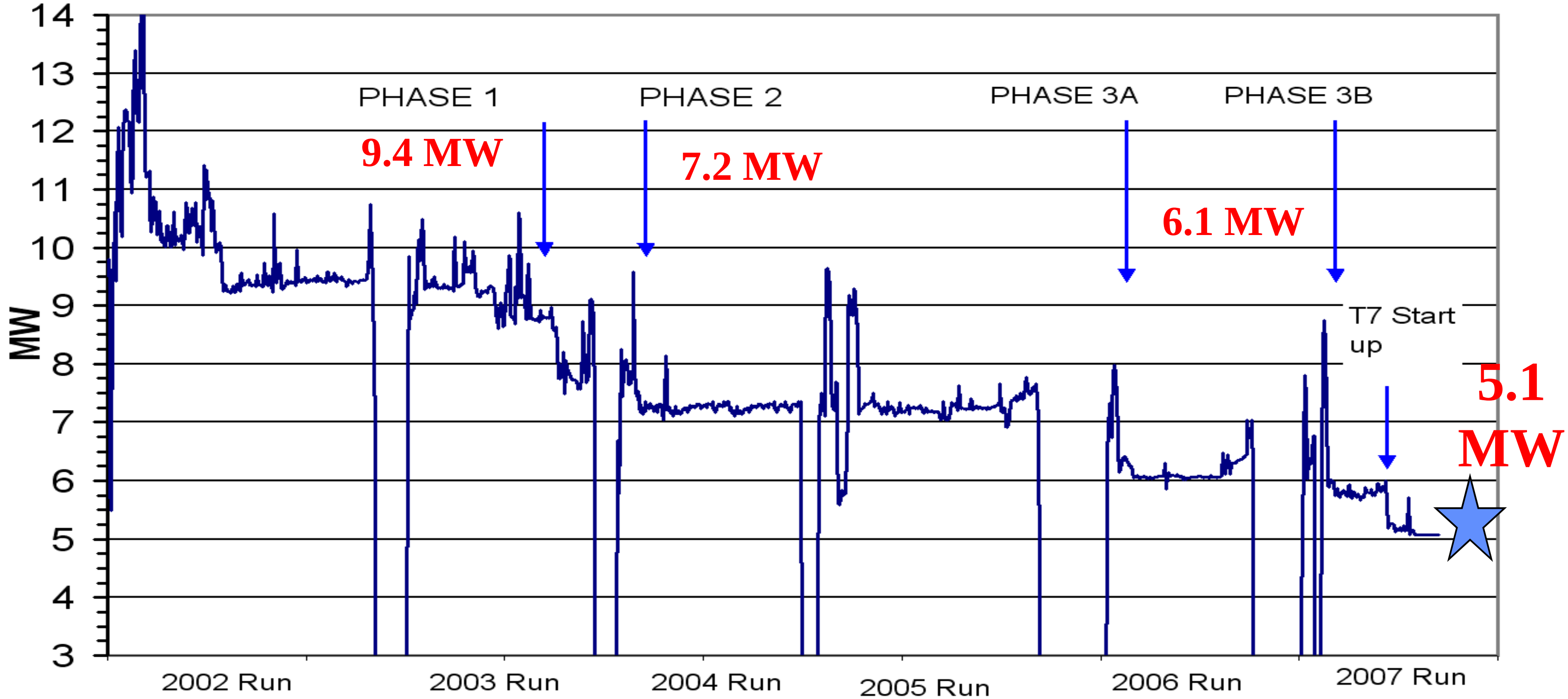






BNL RHIC Energy Savings at the Completion of Phase III

Electric Power History Graph, (Phase III "Goal" 5.4MW)



Exceeded 2003 Goal of 5.4MW……46% Electrical Power Reduction Presently (2010) it is at 4.8 MW

NASA-JSC/JLab Collaboration

James Webb Telescope

Replaces Hubble
~1 million miles out

Telescope Mockup at the National Mall, D.C.

Floating Pressure Technology For Telescope Testing

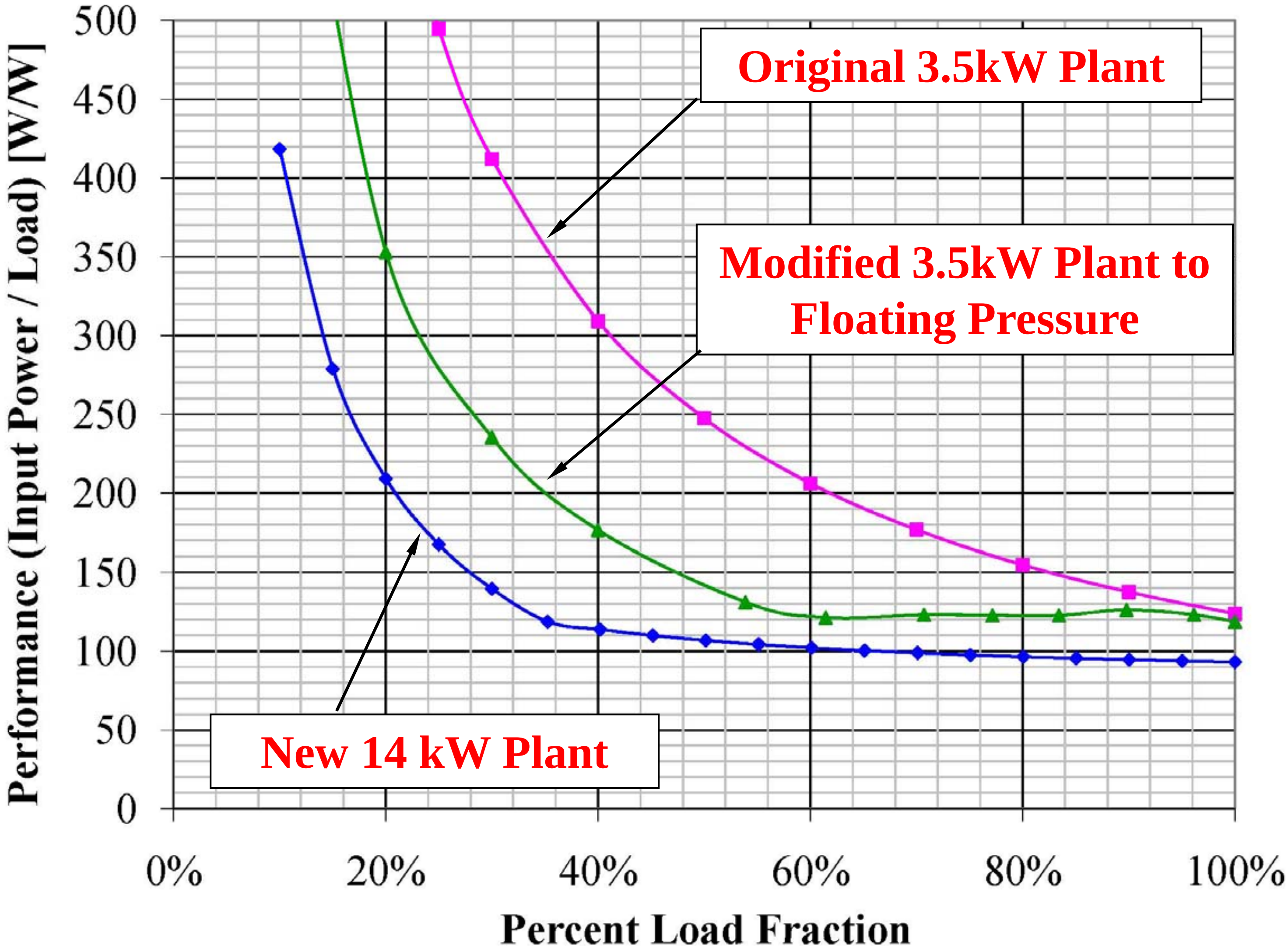
Environmental Space Simulation Chamber-A, NASA, Houston

The existing 3.5 kW 20K cryogenic systems are converted to JLab's Floating Pressure Technology.

Improved temperature stability from 2.5K to 0.25K and efficiency (follows)

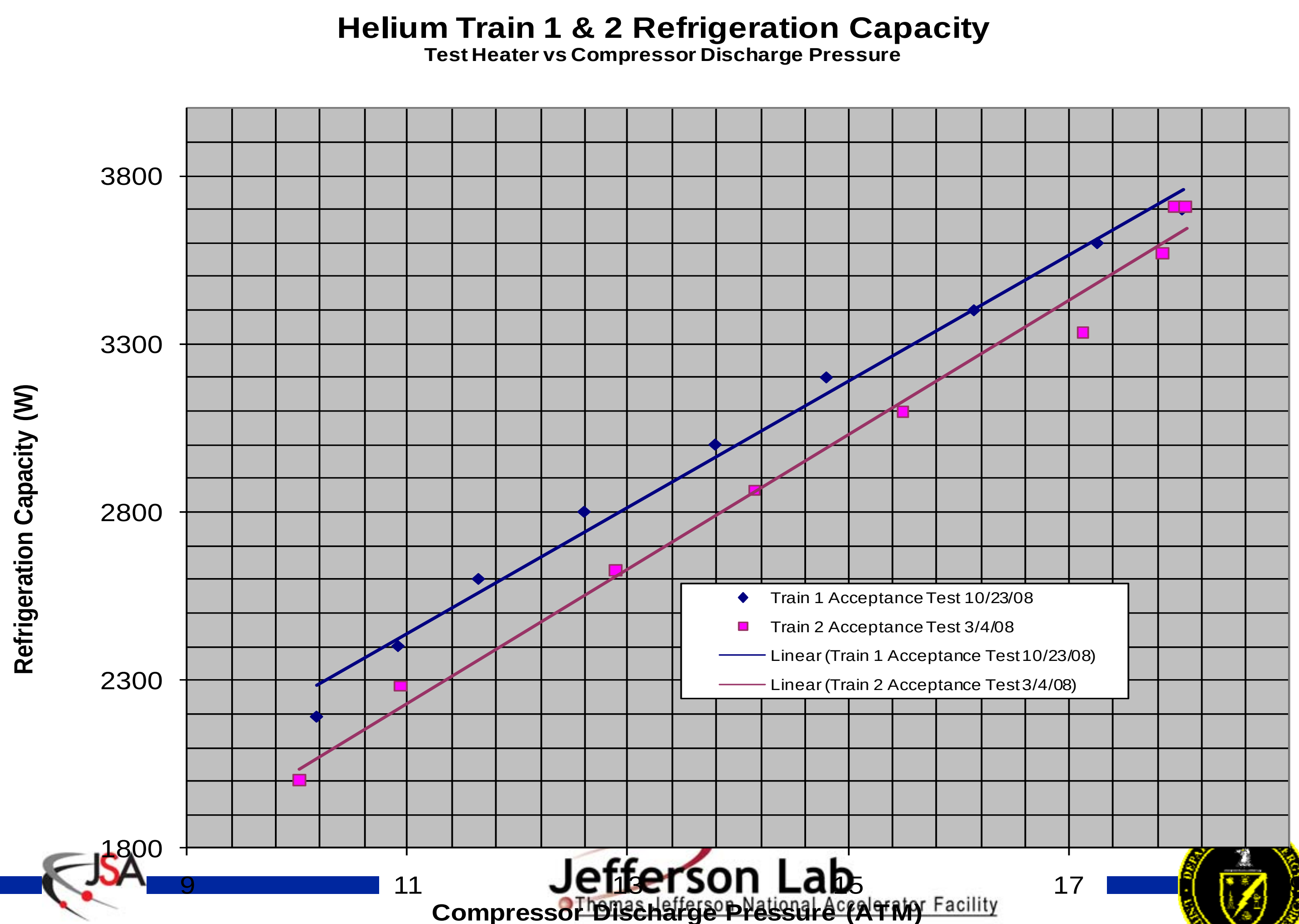
- **New 20K, 13kW refrigerator design is**

NASA-JSC 2008 3.5kW Plant Test Results



NASA-JSC 3.5kW Plants Test Results

Refrigeration Power Produced vs. Compressor Discharge Pressure



3.5kW Plant conversion to Floating Pressure

Reaults

Change over to the floating pressure Ganni Cycle control:

- Greatly improve the system performance
  - System Carnot efficiency is constant from
    55 to 100% of the capacity
  - Power savings and reduced LN2 consumption

- Improved system operational stability
  - Improved load temperature stability
    ~2.5K to 0.25K

3.5kW Plant conversion to Floating Pressure

Results (Cont.)

- Operator intervention requirement is
  - substantially reduced (or practically eliminated)

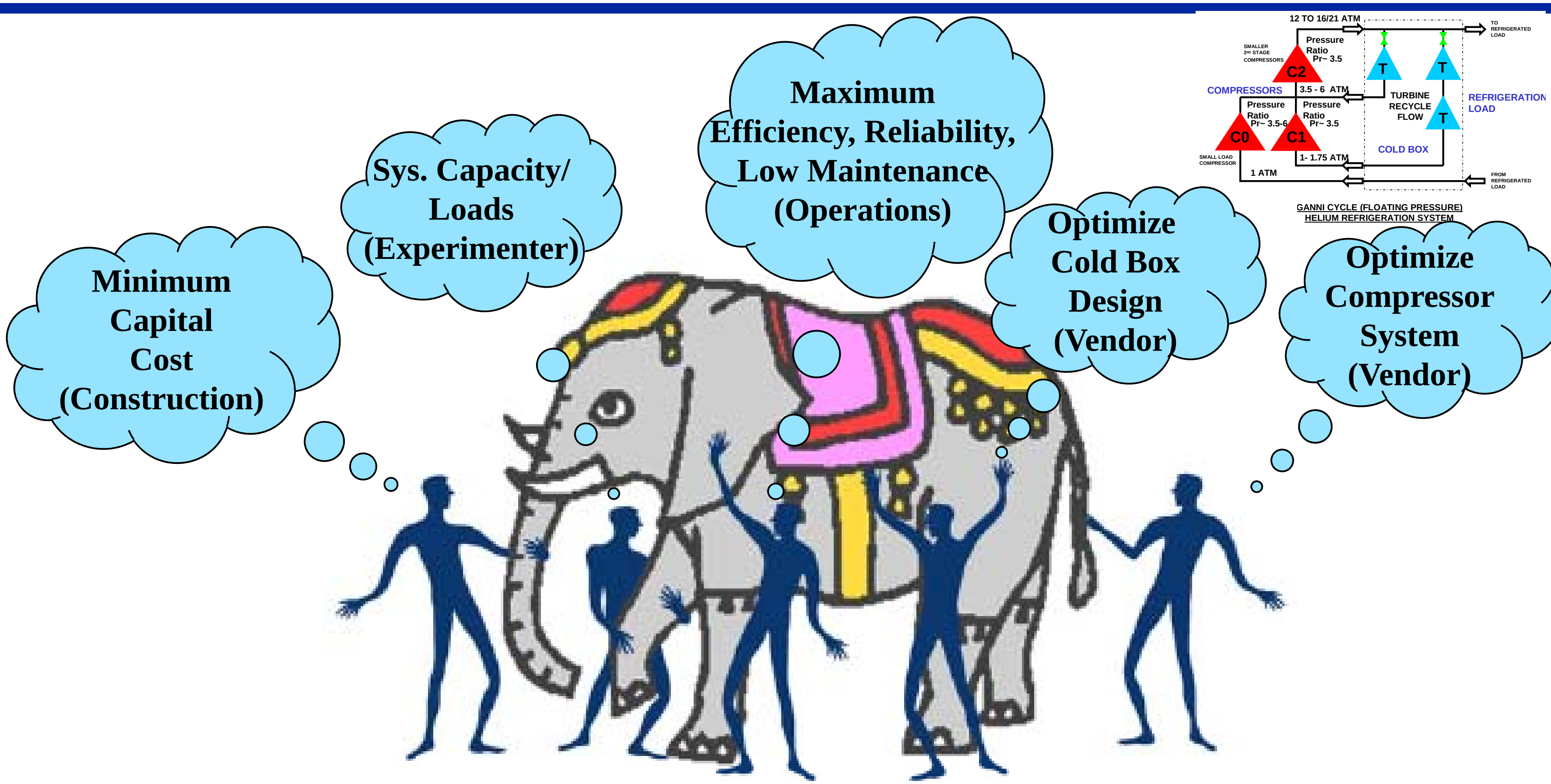
- Maintenance requirements are expected to
  - be reduced on the compressor
  - improve system reliability

- Proved that two identical systems designed to the same design TS have different optimal performance characteristics, i.e
  - <u>*disproved the notion that the design TS is the optimum for a given (as built) systems*</u>

What is common in all these Jobs is:

The Variable pressure operation and is one of the key factors in able to adopt to different load conditions efficiently.

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