OPTIMAL DESIGN AND OPERATION OF HELIUM REFRIGERATION SYSTEMS USING THE GANNI CYCLE

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ABSTRACT

The constant pressure ratio process, as implemented in the floating pressure - Ganni cycle, is a new variation to prior cryogenic refrigeration and liquefaction cycle designs that allows for optimal operation and design of helium refrigeration systems. This cycle is based upon the traditional equipment used for helium refrigeration system designs, i.e., constant volume displacement compression and critical flow expansion devices. It takes advantage of the fact that for a given load, the expander sets the compressor discharge pressure and the compressor sets its own suction pressure. This cycle not only provides an essentially constant system Carnot efficiency over a wide load range, but invalidates the traditional philosophy that the ('TS') design condition is the optimal operating condition for a given load using the as-built hardware. As such, the Floating Pressure-Ganni Cycle is a solution to reduce the energy consumption while increasing the reliability, flexibility and stability of these systems over a wide operating range and different operating modes and is applicable to most of the existing plants. This paper explains the basic theory behind this cycle operation and contrasts it to the traditional operational philosophies presently used.

KEYWORDS: helium, cycles, refrigerator, screw compressor, efficiency

INTRODUCTION

Traditional cryogenic helium refrigeration and liquefaction process cycles are designed at specified maximum capacity operating point(s). In practice however the actual refrigeration and/or liquefaction loads often vary. In addition the components used in the system do not always perform exactly as envisioned in the cycle design cases, which are traditionally represented by the TS design diagrams. 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. Common methods for plant capacity reduction are the use of pressure-throttling valves, adding a load using heaters and/or bypassing the cold and/or warm helium gas. These methods in themselves introduce inefficiencies, presumably to maintain the TS design condition or close to it. So, for traditional process designs, the actual operating utility requirements (electric power, liquid nitrogen and cooling water requirements) per unit load (of refrigeration and/or liquefaction) significantly increases at reduced loads. Although these mechanisms reduce plant production, they have only a limited effect on reducing the required utilities to maintain high plant efficiency. *These traditional methods are analogous to driving a car with a fully depressed gas pedal while controlling the actual speed with a brake*.

Thus, the underlying assumption for traditional process designs is that the TS design condition is considered the optimum operating condition for the actual equipment and actual loads. *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*. It assumes that most controls are for protecting the equipment; e.g., preventing expander temperatures from getting too cold (to maintain the required bearing capacity etc.), or preventing the 1st stage compressor suction from going sub-atmospheric or some minimum pressure to ensure oil removal effectiveness etc. However, this can be easier said than done, since it is not uncommon for equipment manufacturers to provide very narrow operating limits to 'protect' the equipment from unknown or non-optimum conditions rather than truly protecting the equipment from damage. Also, the Floating Pressure Process – Ganni cycle only utilizes key process parameters that are the independent system process variables. This is contrary to many traditional process cycles that attempt to manipulate a sizeable number of variables presumably for process optimization and equipment protection. In this paper, the authors have attempted to demonstrate that *the Floating Pressure Process – Ganni cycle invalidates the traditional philosophy that the TS design condition is the optimal operating condition for as-built hardware and actual loads*.

BASIC FLOATING PRESSURE CYCLE

Consider the basic system consisting of one compressor working with a cold box containing a heat exchanger and a turbo expander as shown in FIGURE 1. This is a simplified arrangement for a typical gas (shield) refrigerator used in many applications (e.g., 20-K systems).

The control scheme for the Floating Pressure Process shown in FIGURE 1 operates as follows:

(a) Compressor bypass (BYP) will respond to prevent the compressor suction ($p_{l,1}$) from going below the set (minimum) pressure (usually ~1.05 atm.). Basically, the sole function of the compressor bypass is to prevent the compressor suction from becoming sub-atmospheric.

(b) Mass-in valve (MI) will respond by opening, charging the system with gas from the gas (or liquid) storage, if the compressor discharge pressure ($p_{h,1}$) falls below the set point.

(c) Mass-out valve (MO) will respond by opening, discharging the system by sending gas out to gas storage, if the compressor discharge pressure ($p_{h,1}$) rises above the set point.

(d) There is a fixed offset in set points between the MI and MO valves (say ~0.2 to 0.3 atm.); with the MI valve set point lower than the MO valve set point. This offset is important for adjusting the charge in the system for a given load.

(e) The compressor discharge set point is a function of the load. For this application, the discharge pressure set point is the output of a control loop that looks at the shield return temperature for its process variable.

(f) During steady state operation, the MO, MI and BYP valves are ***ALL CLOSED***.

Nomenclature:

(h)	*HP Stream*
(ℓ)	*LP Stream*
(b,x)	*Expander bypass stream*
(b,L)	*Load heater bypass stream*
MO	*Mass-out*
MI	*Mass-in*
BYP	*Compressor bypass*
EXP	*Expander*
HX	*Heat exchanger*
HTR	*Load heater*
(1), (2), etc.	*Temperature level (T.L.) #*

Subscripts:

$c,S = \ell,1$	*Compressor suction*
$c,D = h,1$	*Compressor discharge*
$SS = h,3$	*Shield supply*
$SR = \ell,3$	*Shield return*
L	*Shield load*

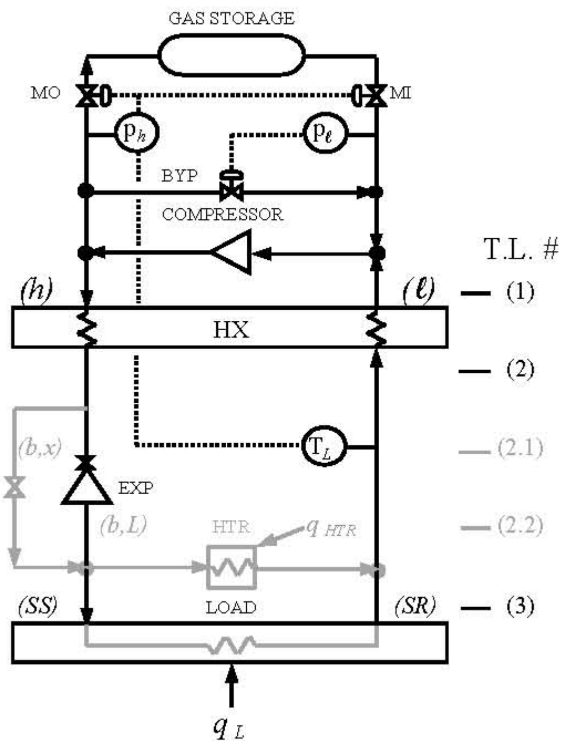

FIGURE 1. General Arrangement for Floating Pressure Process Cycle (***patent pending***)

The availability to the cold box (and to the load) is set by the compressor system and is proportional to both the system mass flow and the logarithm of the (high to low) pressure ratio. For the Floating Pressure Process, the gas charge (i.e., system gas mass in the cycle) is manipulated by the MO and MI valves. Since *the expander and compressor are essentially constant volume flow devices*, they each set their own respective inlet pressures for a given mass flow rate at their operating temperatures. Since both devices have the same mass flow rate, *this characteristic establishes a pressure ratio that is essentially invariant to the mass flow*. So, for a given system gas mass charge, the discharge pressure is set by the expander flow coefficient and the suction pressure is set by the compressor displacement. Although, in theory, either the discharge or suction pressure signals could be used as the key process variable used (to adjust) to match a given load, in practice, using the discharge pressure provides larger signal and thus leads to a very stable system. The (essentially) constant pressure ratio maintains a (nearly) constant enthalpy drop across the expander (assuming constant efficiency) which results in an (approximately) invariant mass specific load enthalpy difference. Since isothermal compressor efficiency is primarily dependent on the pressure ratio [1], with an essentially constant pressure ratio, the mass specific compressor input power is nearly constant. Further, as the system temperatures vary a picayune amount (even under varying load conditions), the mass specific load exergy is nearly invariant. This results in an *essentially constant system Carnot efficiency* over a very wide load range. The following illustrates this mathematically.

TABLE 1. Symbols and Subscripts

	Symbols:			**Symbols:**	
C_p	Specific heat at constant pressure	[J/g-K]	ϕ	$=(\gamma-1)/\gamma$	[non-dim.]
h	Enthalpy (mass specific)	[J/g]	γ	Ratio of specific heats	[non-dim.]
Δh	Enthalpy difference	[J/g]	η	Efficiency	[non-dim.]
$\dot{m}$	Mass flow rate	[g/s]	ρ	Density	[kg/m^3] or [ℓ/s]
N_0	Units conversion constant (=101.325)	[kPa/atm]	τ	$=\Delta T_{hl}/T_l$	[non-dim.]
Ntu	HX number of transfer units	[non-dim.]	ξ	Actual to design mass flow ratio	[non-dim.]
p	Pressure	[atm]			
p_r	Pressure ratio	[non-dim.]		**Subscripts:**	
Δp	Pressure difference	[atm]	C	compressor	
q	Heat transfer	[W]	D	Design (or observed)	
Q	Volumetric flow	[ℓ/s]	h	High pressure stream	
s	Entropy (mass specific)	[J/g-K]	hl	Difference between 'h' & 'l' streams	
Δs	Entropy difference	[J/g-K]	i	isothermal	
T	Temperature	[K]	l	Low pressure stream	
ΔT	Temperature difference	[K]	L	(shield) load	
(UA)	HX thermal rating	[W/K]	m	motor	
$\dot{W}$	Power	[W]	r	ratio	
w	Specific work	[W/(g/s)]	v	volumetric	
E	Exergy	[W]	x	expander	
ε	Physical exergy (mass specific)	[J/g]	0	Reference (state)	
κ	Flow coefficient (units defined by equation)		1,2,3	Temperature level	

From the specified design load conditions, the compressor displacement capacity, expander flow coefficient and heat exchanger (HX) size are determined, considering the component limitations and best operating ranges for overall optimum efficiency [2, 3]. Now, assuming all the flow is going to the load (i.e., there is no bypass etc.),

$$\dot{m}_x = \dot{m}_L$$

Most of the helium refrigeration/liquefaction cycles are constructed with constant volume displacement compressors (e.g., screw, reciprocating compressors) and centrifugal turbo expanders. Also, assuming that there is no compressor bypass,

$$\dot{m}_C = \dot{m}_x$$

With, $\dot{m}_C = \eta_v \cdot Q_C \cdot \rho_{l,1}$ (1)

and, $\rho_{l,1} = \dfrac{N_0 \cdot p_{l,1}}{\phi \cdot C_p \cdot T_{l,1}}$

Since the expander is essentially a constant volume flow device, the flow through the expander is (essentially) choked (or critical) flow,

$$\dot{m}_x = \kappa_x \cdot \frac{p_{h,2}}{\sqrt{T_{h,2}}} \quad (\text{for } p_{r,x} \geq 2) \tag{2}$$

For this analysis, we will assume C_p to be constant and the same for both high and low pressure streams. Equating the compressor and expander mass flows, we have a characteristic pressure ratio that is essentially constant.

$$p_r \equiv \frac{p_{h,2}}{p_{l,1}} = \left(\frac{\eta_v \cdot Q_C}{\kappa_x}\right) \cdot \left(\frac{1}{\phi \cdot C_p}\right) \cdot \frac{\sqrt{T_{h,2}}}{T_{l,1}} \cong \underline{\boldsymbol{Constant}} \tag{3}$$

Now, examining $T_{h,2}$ and $T_{l,1}$ for the HX shown in FIGURE 1, since there is balanced flow in the HX,

$$\Delta T_{hl,2} = \frac{\Delta T_{\max}}{(1 + Ntu)} \quad \text{and, } \Delta T_{hl,2} = \Delta T_{hl,1}$$

with,

$$T_{l,3} = T_{l,2}, \quad \Delta T_{hl,1} = T_{h,1} - T_{l,1}, \quad \Delta T_{hl,2} = T_{h,2} - T_{l,2}, \quad \Delta T_{\max} = T_{h,1} - T_{l,3} \quad \text{and,}$$

$$Ntu = \frac{(UA)}{\dot{m} \cdot C_p}$$

Keep in mind that $T_{h,1}$ is set by the ambient temperature (or 80-K liquid nitrogen pre-cooling) and $T_{l,2}$ $(= T_{l,3})$ is assumed to be maintained by the control system to satisfy the load requirements; so, $\Delta T_{\max}$ is constant. Now, assuming that the (UA) scales with the mass flow by approximately, the relationship between the (UA) at a given $\dot{m}$ and the design (or observed) (UA) at the design (or observed) $\dot{m}_D$ is,

$$\frac{(UA)}{(UA)_D} = \left(\frac{\dot{m}}{\dot{m}_D}\right)^n \quad \text{where, } n \approx 0.67$$

letting, $\xi = \dot{m}/\dot{m}_D$,

then, $Ntu = \dfrac{(UA)_D}{\dot{m}_D^n \cdot \dot{m}^{(1-n)} \cdot C_p} = \dfrac{(UA)_D}{\xi^{(1-n)} \cdot \dot{m}_D \cdot C_p} = \dfrac{Ntu_D}{\xi^{(1-n)}}$. (4)

This means that if the mass flow decreases to 35% of the design flow (ξ = 0.35), the HX *Ntu*'s increase by ~40%. Further, recalling the equation for $\Delta T_{hl,2}$, for large *Ntu*'s (20+), this will have a diminishing effect on $\Delta T_{hl,2}$ and $\Delta T_{hl,1}$ (and therefore, $T_{h,2}$ and $T_{l,1}$). Since $T_{l,1}$ and $T_{h,2}$ change very little (and $\sqrt{T_{h,2}}$ even less), even over wide variations in mass flow, we find that the characteristic ***pressure ratio*,** $p_r = p_{h,2}/p_{l,1}$ ***is approximately and practically constant***.

The specific input power to the compressor is,

$$w_C = \frac{\dot{W}_C}{\dot{m}_C} = \frac{T_{l,1}\cdot\phi\cdot C_p\cdot\ell n(p_{r,C})}{\eta_i\cdot\eta_m} \tag{5}$$

where the compressor pressure ratio is

$$p_{r,C} = \frac{p_{h,1}}{p_{l,1}} = p_r + \frac{\Delta p_h}{p_{l,1}}, \quad \text{with, } \Delta p_h = p_{h,1} - p_{h,2}$$

Note that the compressor isothermal efficiency η_i is primarily a function of the compressor pressure ratio $p_{r,C}$, which is essentially constant [1].

The specific load exergy (reversible work) is,

$$\frac{\mathrm{E}_L}{\dot{m}_L} = \Delta\varepsilon_L = \Delta h_L - T_0\cdot\Delta s_L \tag{6}$$

with,

$$T_{r,x} = \frac{T_{h,2}}{T_{h,3}} = \left[1-\eta_x\cdot(1-p_{r,x}^{-\phi})\right]^{-1}$$

$$\Delta h_L = \frac{q_L}{\dot{m}_L} = C_p\cdot\left(T_{l,3}-T_{h,3}\right) = C_p\cdot T_{l,3}\cdot\left[1-\frac{\left(\tau_2+1\right)}{T_{r,x}}\right] \tag{7}$$

$$\tau_2 = \frac{\Delta T_{hl,2}}{T_{l,2}}$$

$$\Delta p_l = p_{h,3} - p_{l,1}, \quad \Delta p_L = p_{h,3} - p_{l,3}$$

$$p_{r,x} = \frac{p_{h,2}}{p_{h,3}} = \frac{p_r}{\left(1+\Delta p_l / p_{l,1}\right)}$$

$$\frac{p_{l,3}}{p_{h,3}} = 1 - \frac{\Delta p_L / p_{l,1}}{\left(1+\Delta p_l / p_{l,1}\right)}$$

$$\Delta s_L = C_p\cdot\left[\ell n\left(\frac{T_{h,3}}{T_{l,3}}\right) - \phi\cdot\ell n\left(\frac{p_{h,3}}{p_{l,3}}\right)\right] = C_p\cdot\left[\ell n\left(\frac{\left(\tau_2+1\right)}{T_{r,x}}\right) + \phi\cdot\ell n\left(1-\frac{\Delta p_L / p_{l,1}}{\left(1+\Delta p_l / p_{l,1}\right)}\right)\right] \tag{8}$$

since, $p_r \cong$ constant, and $T_{l,3} \cong$ constant,

$$\frac{\Delta p_l}{p_{l,1}} \cong \text{ constant, } \frac{\Delta p_L}{p_{l,1}} \cong \text{ constant, } p_{r,x} \cong \text{ constant, } T_{r,x} \cong \text{ constant}$$

Further, since changes in τ_2 are small, the specific load exergy ($\Delta\varepsilon_L$) remains approximately constant. So for all practical purposes, the Carnot efficiency is

$$\eta_{carnot} = \frac{\mathrm{E}_L}{\dot{W}_C} = \frac{\Delta\varepsilon_L}{w_C} \cong \underline{\textbf{\textit{Constant}}} \tag{9}$$

There are a few additional key observations from the above analysis:

1. Since $Q = \dot{m}/\rho \sim \dot{m}/p$ and $\dot{m} \sim p_l$ and $p_h \sim p_r \cdot p_l$, the volume flow (and thus the velocity) at any point in the system remains approximately constant as the system pressure varies. This is readily apparent recalling that both the compressor and expander are constant volume flow devices. As such both the expander efficiency (i.e., one utilizing a variable brake) and the oil removal efficiency remain approximately constant (recalling that the control scheme will not allow any compressor bypass flow until the actual compressor suction pressure falls below the set point pressure). So, *the Floating Pressure Process does not pose any additional threat to the compressor system oil removal as long as the* ***compressor bypass is not used*** *at reduced operating pressures*.

2. In the first order, the pressure loss Δp_l reduces the refrigeration capacity (by reducing the compressor suction pressure and thus the mass flow rate) and Δp_h increases the compressor input power by increasing the pressure ratio.

3. The minimum turn-down (load decrease) before throttling and/or load heaters would be necessary is determined by the minimum compressor suction pressure (i.e., the set point, below which the compressor bypass opens) and/or the expander operational limit (equipment limitations such as bearing thrust or shaft natural frequency coincidence).

In summary so far, the Floating Pressure Process allows the system pressures to adjust (as previously described) at a nearly constant pressure ratio. In turn this provides *an essentially constant system Carnot efficiency over a wide load range* (i.e., capacity turn-down from the design load).

TS DIAGRAM DESCRIPTION

FIGURE 2 depicts the TS diagram for a shield refrigerator (as in FIGURE 1) at the ideal design conditions. That is, assuming ideal gas behavior with constant specific heat (i.e., fluid ideality), neglecting stream/load pressure drops, heat leak and non-constant rotating machinery efficiencies (i.e., process idealities). From this diagram, there are several initial observations to be made:

- Y-axis is the natural logarithm of temperature
- Between any two arbitrary points '1' and '2', the difference in 's' values between these points is, $\Delta s = (s_2 - s_1) = C_p \cdot \{\ell n(T_2/T_1) - \phi \cdot \ell n(p_2/p_1)\}$, or $\Delta s = C_p \cdot \{\ell n(T_r) - \phi \cdot \ell n(p_r)\}$; where $T_r = T_2/T_1$ and $p_r = p_2/p_1$
- So, at constant temperature (isotherms), $\Delta s = -\phi \cdot C_p \cdot \ell n(p_r)$, and,
- At constant pressure (isobars), $\Delta s = C_p \cdot \ell n(T_r)$
- Slope of isobars is equal to the specific heat at constant pressure (C_p).

If all components and loads were exactly as designed, the Floating Pressure Process would automatically adjust the system to operate at the TS design condition. However, in practice, no actual system operates exactly as per the 'TS' design conditions [4]. The main reasons are:

1. Given manufacturing and performance tolerances, it is practically impossible to exactly predict the performance of the components (e.g., compressors, expanders, heat exchangers, pressure drops, heat leaks, etc.).

2. The system is designed for maximum capacity or for some rare operating mode (like cool down) but is not required for normal operation.

3. The margins allocated (e.g., load, system capacity) in the system design are in the actual operating case either in excess of actual needs or not sufficient.

4. The load characteristics (e.g., refrigeration, liquefaction, actual load size) have changed or a different than as specified in the original design.

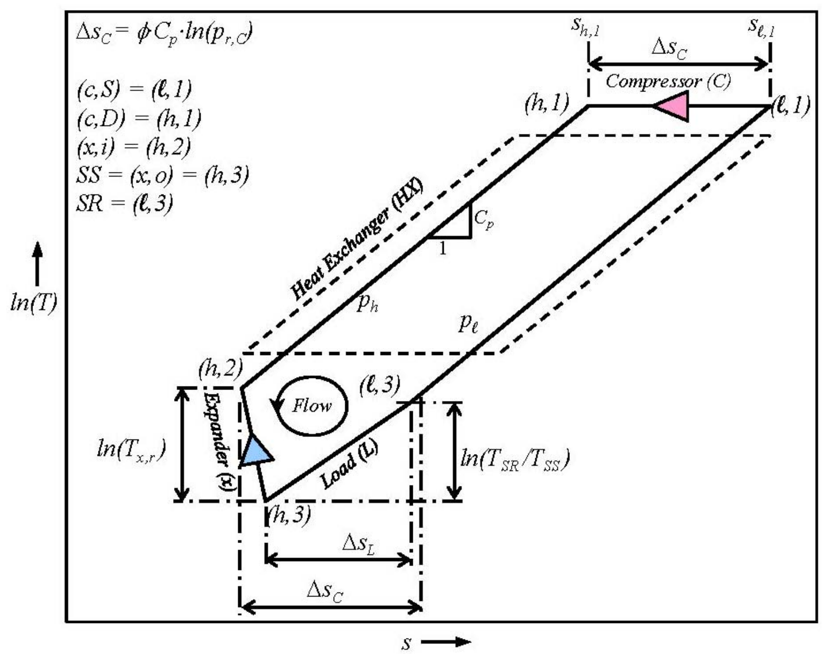

FIGURE 2. TS Diagram of Shield Refrigerator

CAPACITY MODULATION

So, to effect capacity modulation while maintaining optimal efficiency (i.e., a proportional decrease in input power to a given reduction in load) at off-design process conditions, it is crucial to maintain a (nearly) constant entropy difference (Δs) at each temperature throughout (from warm end to cold end) and to allow the mass flow ($\dot{m}$) to decrease proportionally with the load (q_L).

Now, as is the case for the Floating Pressure Process, ***if*** p_r ***is constant***, then recalling the observations of the TS diagram description, it is straight-forward to recognize that $\Delta s = \Delta s_C = (s_{l,1} - s_{h,1})$ and $T_{r,x} = (T_{h,2}/T_{h,3})$ are (essentially) constant. With these, the following additional observations can be made, again referring to FIGURE 2:

- The mass specific availability to the cold box is essentially constant, and is equal to the area under the process path from *(h,1)* to *(ℓ,1)*; it is approximately equal to the mass specific isothermal compressor power,
$$\dot{w}_{C,i} \cong \Delta s_C \cong \phi \cdot C_p \cdot T_{l,1} \cdot ln(p_{r,C}) \tag{10}$$
- Mass flow ($\dot{m}$) is proportional to the system gas charge (mass) which is proportional to the absolute system pressure levels; i.e., p_h or p_l
- The mass specific (shield) load $(q_L/\dot{m})$ is essentially constant and, neglecting load irreversibilities, is equal to the area under the process path from *(h,3)* to *(ℓ,3)*; it is approximately,
$$(q_L/\dot{m}) = \Delta h_x - \Delta h_{hl,1} \cong C_p \cdot \left\{ \left(T_{SR} + \Delta T_{hl,2}\right) \cdot \left(1 - T_{r,x}{}^{-1}\right) - \Delta T_{hl,1} \right\} \tag{11}$$

This floating pressure process is also a variable (mass) charge system. Referring to FIGURES 1 and 2 and Case #1 in FIGURE 3, as the load (q_L) decreases and the mass out (MO) valve responds by releasing gas back to gas storage, *<u>the process cycle translates right without size change (i.e., 'floats') from the 'black' lined cycle to the dashed 'red' lined cycle</u>*. The pressure ratio remains essentially the same, but the system has decreased its gas charge, thereby decreasing the mass flow rate ($\dot{m}$). Therefore, the 'width' and 'height' of the process cycle remains essentially unchanged as the process responds to changes in the load. That is, the entropy difference between *(h)* and *(ℓ)* streams at each temperature is nearly constant. It should be recalled that $T_{h,1}$ is fixed (by the compressor cooling water or 80-K liquid nitrogen pre-cooling) and the control system is maintaining $T_{l,3} = T_{SR}$ at the desired set-point. *<u>This load capacity modulation is done without introducing process mechanisms that in themselves produce **exergy losses**</u>*. Further, it is implicit that the expander adiabatic efficiency does not decrease with a load reduction (i.e., which is a good approximation for a variable brake expander and no flow throttling).

TABLE 2. 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)

Cases #2 to #6 in TABLE 2 and FIGURES 3 to 5 are traditional methods to achieve a turn-down (i.e., load reduction) in plant capacity. Also, Cases #2 to #4 and #6 maintain the same total compressor mass flow upon a decreasing load. Interestingly, only Case #2 follows the 'TS' path upon load turn-down (though it will be later shown that it is not necessarily more efficient to do so). Case #2 adds a heat load, so that the total heat load equals the design load. Cases #3 and #5 throttle the expander inlet valve to waste the availability generated by the compressor that is not required by the load. Case #3 maintains the design compressor suction pressure, but accumulates compressor bypass upon a decreasing load. Case #5 maintains zero compressor bypass, but this results in the compressor pressure ratio increasing upon a decreasing load (even though mass flow decreases). Case #6 reduces refrigeration produced by the expander by bypassing mass flow around the expander. Case #4 decreases discharge pressure upon a decreasing load, but maintains the design compressor suction pressure. Cases #4 and #6 can present compressor system oil removal problems due to decreasing discharge pressure, but no reduction in mass flow. In some instances, a combination of Cases #2 to #6 are also used for capacity modulation. In summary, except the floating pressure case (#1), all other cases waste some of the specific exergy developed by the compressor and do not allow the input power to decrease, in a significant way, upon a decreasing load. As such, only the Floating Pressure Process (Case #1) can offer an essentially constant efficiency upon a decreasing load.

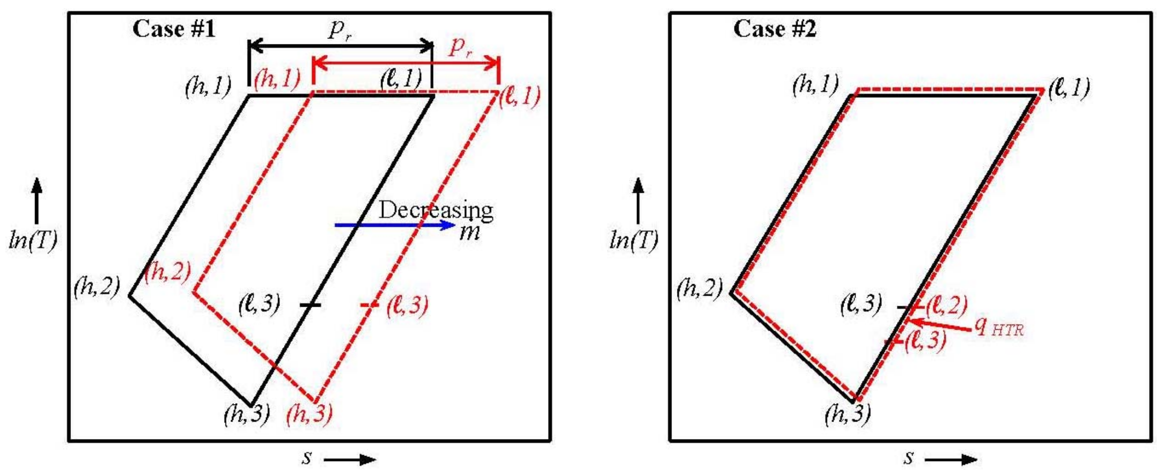

FIGURE 3. TS Diagram of Floating Pressure Process (Case #1) and for Case #2

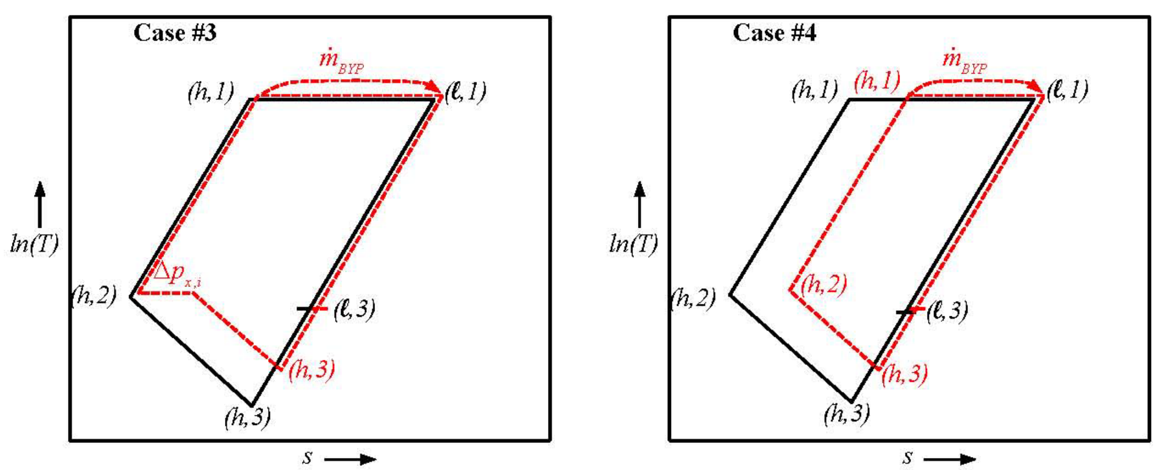

FIGURE 4. TS Diagram of Cases #3 & #4

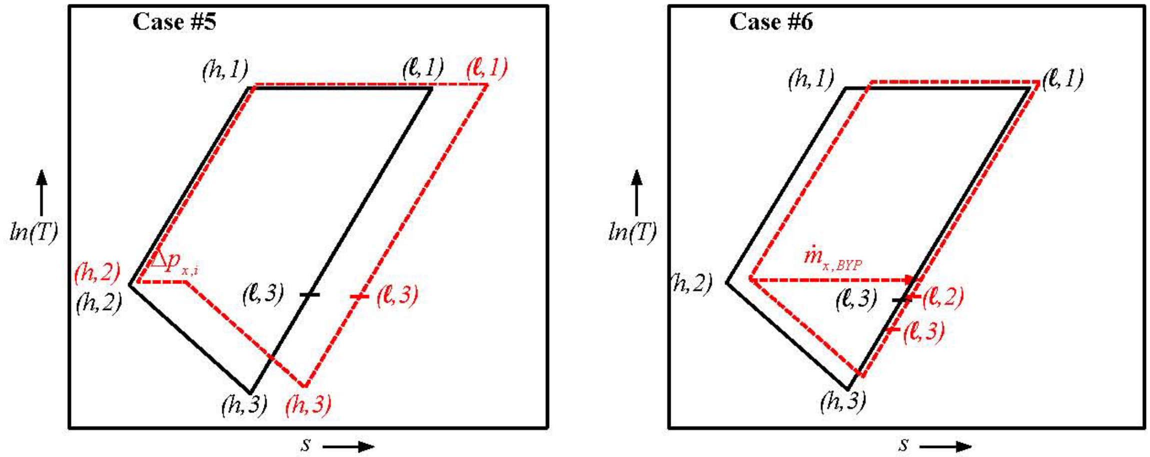

FIGURE 5. TS Diagram of Cases #5 & #6

By relieving over-determinate process constraints, the Floating Pressure Process entreats the question of whether the TS design conditions are optimal for as-built hardware. *<u>That is, should the TS design conditions be coerced on as-built hardware to achieve optimal efficiency for the design load?</u>* The answer to this question can be found in examining off-design conditions by introducing a small variation in one of the equipment parameters.

TABLE 3. Effect of Variations in Equipment Parameters

Case #	Selected Equipment Parameter Less Than Design Value	Consequence at Same Load	
		Pressure Ratio (p_r)	**Mass Flow ($\dot{m}$)**
A	HX Size (Ntu)	Increase	Increase
B	Expander Efficiency (η_x)	Increase	Increase
C	Expander Flow Coefficient (κ_x)	Increase	Decrease
D	Compressor Vol. Efficiency (η_v)	Decrease	Increase

Referring to TABLE 3 and FIGURES 6 and 7 (Cases A to D), the 'black' lined cycle is the (intended) TS design condition and the dashed 'red' lined cycle is how the actual cycle would operate *<u>under the design load</u>* using the Floating Pressure Process for a small decrease from the design value for in the selected equipment parameter. 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 (i.e., the 'black' line in FIGURES 6 and 7) one of two results would occur:

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

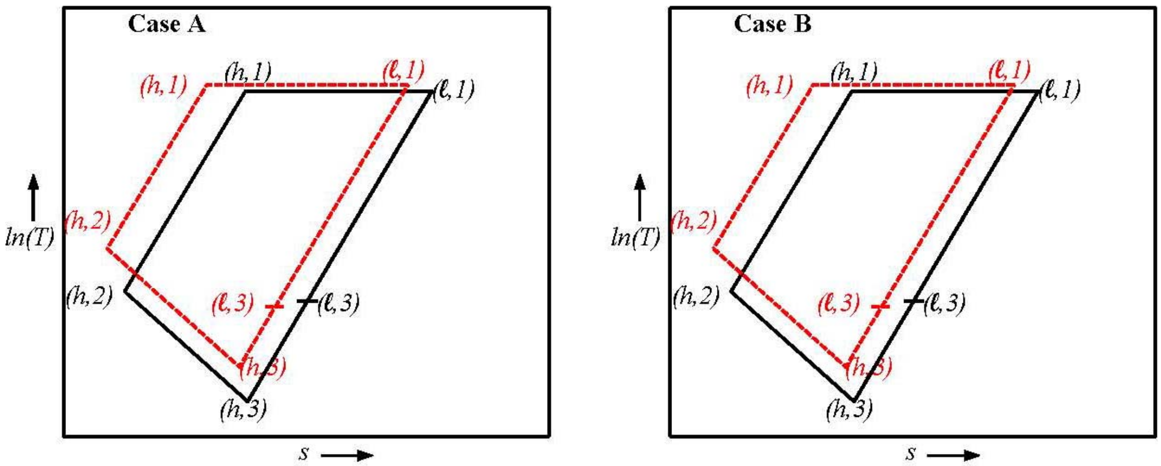

FIGURE 6. TS Diagram of Cases A & B

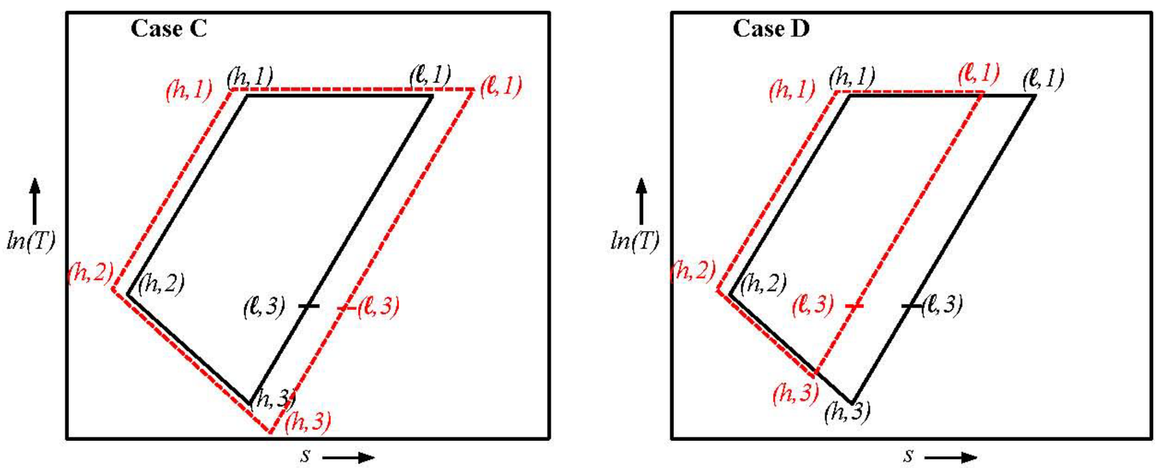

FIGURE 7. TS Diagram of Cases C & D

It is important to notice that the Floating Pressure Process ***is not contingent on precise instrumentation for successful operation***, and in fact, will operate quite contently regardless of the calibration or accuracy of the instrumentation. This is due to decoupling specific values of process variables from presumed system load capacities.

THE GANNI CYCLE

The Floating Pressure Process is applicable not only for a shield refrigerator but also an isothermal refrigerator and/or a liquefier. FIGURE 8 depicts the superposition of several shield refrigerators [3, 4], each operating at a progressively colder temperature level. This superposition of shield refrigerators is the ‘recycle flow’ or expander loop(s) in the traditional Claude cycle, and comprises 60 percent or more (up to 90 percent) of the total compressor flow. Of course, the ‘shield’ load for each expander in the Claude cycle is the cooling required for the liquefaction load, heat exchanger losses, heat leak and any shield loads that may be present.

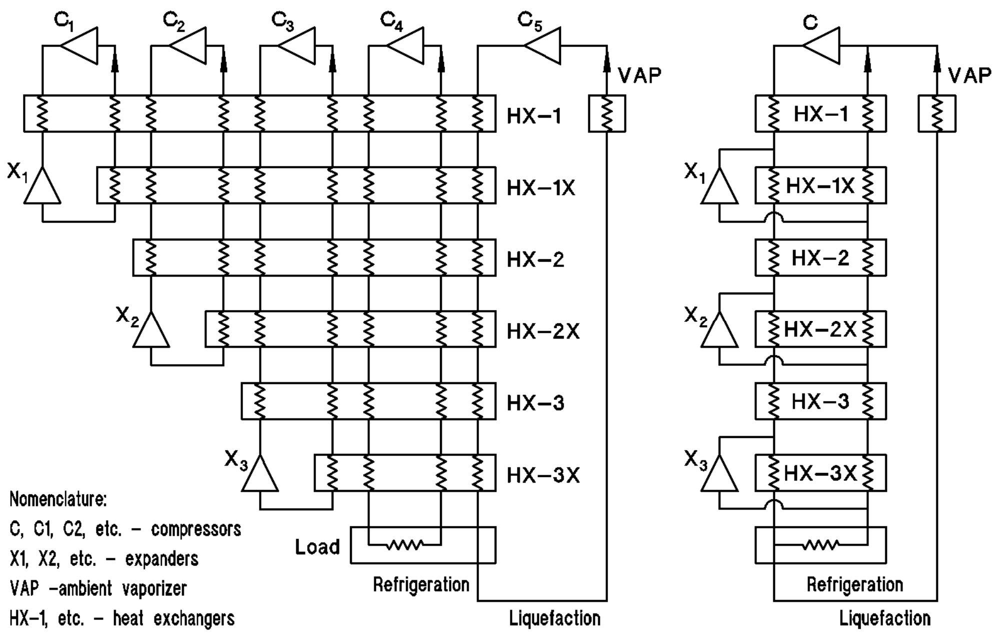

FIGURE 8. Superposition of Shield Refrigerators

The majority of the helium refrigeration and liquefaction system exergy losses (up to ***approximately*** ***2/3 of the total loss*** [5]) are a result of compressor system inefficiencies. As such, it is important to integrate the compressor efficiency characteristics in the cycle design. As shown in [1], the optimum (maximum) isothermal (and volumetric) compressor efficiency is primarily dependent on the pressure ratio, and it is around 3 to 4 for screw compressors. Now, consider if the expanders' recycle flow is allowed to operate using the Floating Pressure Process at the optimum pressure ratio, and the refrigeration load return is segregated from the expander recycle return, so as to maintain the lowest possible constant refrigeration load temperature. Such an arrangement is the Ganni cycle. FIGURE 9 depicts a possible multi-stage compressor arrangement for maximizing the exergy supply to the cold box and achieving good overall system efficiency within practical pressure limits. Although it is not necessary (nor perhaps practical) to completely segregate the expander recycle flow from the refrigeration load return flow, it is capable of achieving greater efficiency and stability. The guidelines for optimal arrangement of expanders in the cold box using the Carnot step are given [2, 3]. Some more detailed applications to helium cycles are given in the various arrangements for Ganni Helium Process Cycle (US patents 7,278,280 & 7,409,834 and the patent pending for the Floating Pressure Process).

Although the Floating Pressure cycle can be applied to systems produced by most manufacturers, there are some systems to which it can be applied over a wider range than the others. As such, there are two notes of caution in applying the Floating Pressure cycle:

1. When the compressor bypass valve is used for reasons based on other control needs, the velocity through the oil removal system ceases to remain constant. In these cases the functionality of the oil removal system should be carefully checked.

2. For systems with at least a modest liquefaction load, an efficient design requires the expander mass flows to be relatively close to each other [2, 3]. In systems where this is not the case, careful attention should be given in balancing between trying to achieve more optimal Carnot steps (expander temperature level spacing) and a turbine's safe operating range.

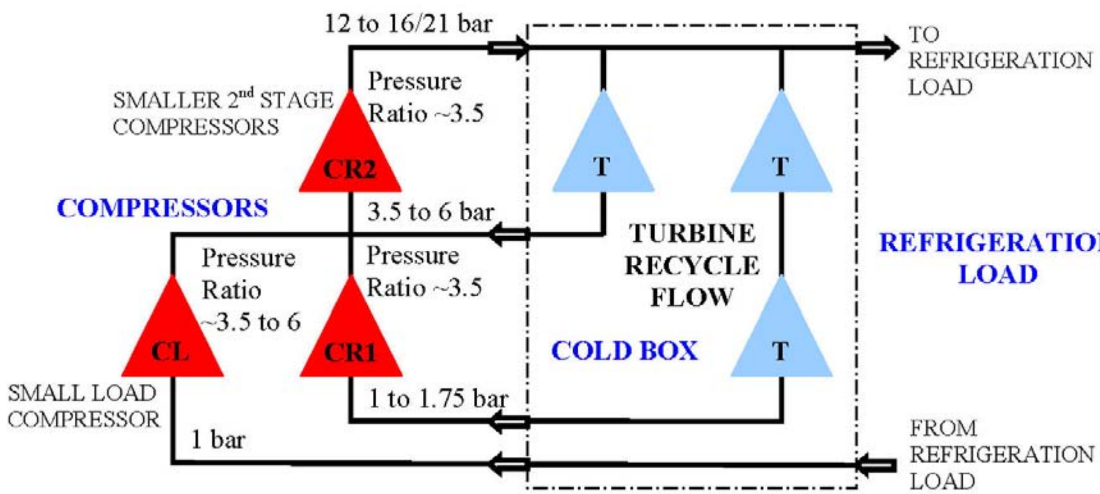

FIGURE 9. Simplified Ganni Helium Process Cycle

APPLICATIONS TO DATE

Fundamental aspects of the Floating Pressure Process were originally applied to the cryogenic system for the Superconducting Super Collider Laboratory (SSCL) string test plant (known as ASST-A) in 1992 to allow the refrigerator to respond efficiently to various modes of operation, including magnet string quench recovery. In 1994-95, the Floating Pressure Process was applied to all four major cryogenic plants at Jefferson Lab (JLab), which are manufactured by different vendors. Later it was applied to Michigan State University (MSU) [6], the Spallation Neutron Source (SNS) [7], Brookhaven National Laboratory (BNL) [8] and for NASA at the Johnson Space Center (JSC) [4]. In all cases, it has resulted in substantial improvements in the system's efficiency, capacity, reliability and stability. Presently JLab licensed the Ganni Cycle Floating Pressure Process technology to "Cryogenic Plants and Services" a Division of Linde BOC Process Plants, LLC for world-wide commercialization.

SUMMARY & CONCLUSIONS

In 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
5. Is a variable gas charge system, whose *gas charge is automatically adjusted* to satisfy the given load
6. 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
7. Has been licensed by JLab to Linde Cryogenics, Division of Linde Process Plants, Inc. and Linde Kryotechnik AG for world wide commercialization

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