OPTIMAL DESIGN AND OPERATION OF HELIUM REFRIGERATION SYSTEMS *

V. Ganni, JLab, Newport News, VA 23606, U.S.A.

Abstract

Helium refrigerators are of keen interest to present and future particle physics programs utilizing superconducting magnet or radio frequency (RF) technology. They typically utilize helium refrigeration at and below 4.5-Kelvin (K) temperatures and are very energy intensive. After an overview of the quality of energy, the Carnot step (as defined by the author) and cycle design theory, the concept of overall process optimization is presented. In particular the question of 'what is an optimum system' will be discussed. In this regard, the Ganni cycle and floating pressure process will be examined with respect to a more traditional approach as a solution to obtain an optimum system for new designs and existing systems.

INTRODUCTION

In the late 1960s, the first Collins helium refrigerator located at Brookhaven National Laboratory (BNL) (~200W at 4.5K) and the refrigerator at Stanford University (300W at 1.8K), designed by Sam Collins, were considered as large cryogenic plants. In these earlier systems the input power was of secondary importance to a working plant and achieving the desired capacity. By the 1980's the plant sizes grew to more than 25 kW at 4.5K (BNL) and 4.6 kW at 2K (Jefferson Lab, or JLab). Thus, there was a relatively rapid change to large systems that were not consistently used at their maximum capacity and/or were used for modes different than those for which they were designed. Yet, these later systems were designed and operated with the philosophy of the earlier systems. That is, the design specified by the end user is at a single process condition (i.e., the maximum capacity, and perhaps one or two other conditions) and then forced to operate at the presumed optimum of the manufacturer's theoretical ('T-S') design, which was usually centered on the cold box design. Often because of conservativeness in specifying the system size, the actual operating loads are smaller than the design capacity, resulting in an inefficient turndown. Even though these systems are very energy intensive, the motivation to design and operate these plants efficiently was not as pressing as it is today. Helium refrigeration systems adapted a majority of components from the traditional refrigeration, air separation and oil/gas industries. Some limited characterization was done on these components for helium systems. However, the design of these systems remained very compartmentalized between the major sub-systems (i.e., load, cold box and compressor systems). These factors amalgamated so as to promulgate the myths that the T-S design is optimal for as built systems, a single (or a few) point design specification from the end user is reasonable and components have been well characterized.

Over these years, as operational experience was gained on many systems, using the now accepted and proven main components (e.g., screw compressors, turbo expanders and plate-fin heat exchangers), it has been recognized and proven that a system design based on a constant pressure ratio (known as "Floating Pressure Process" or "Ganni Cycle" ; patented) will lead to an optimal system design and improved efficiency of operating systems with the added benefits of stability, reliability and flexibility for most applications. Presently, due to the rapid increase in energy costs and an increased need for greater capacity, there is an increased interest to design new systems and convert the existing systems to this constant pressure ratio cycle.

*Work supported by U.S. Department of Energy, Contract no. DE-AC05-06OR23177.

QUALITY OF ENERGY

Clausius (In)equality

The inequality of Clausius, which forms the basis for the definition of the intrinsic fluid property of entropy, is key in understanding the ***'quality'*** or ***'availability'*** of energy. The inequality of Clausius is of course an ***equality*** only if the cycle is ***reversible***. In such a case, the relationship is then,

$$\frac{\Delta Q_L}{T_L} = \frac{\Delta Q_H}{T_H} \quad \text{e.g.,} \quad \frac{300\ \text{W}}{300\ \text{K}} = \frac{4\ \text{W}}{4\ \text{K}} = \frac{2\ \text{W}}{2\ \text{K}}$$

This equation is a statement of thermal energy quality equivalence. So, Q_L = 1W at T_L = 4.22 K is equivalent in quality as Q_H = 70 W at T_H = 300K.

Exergy

Physical exergy is defined [1] as, $\varepsilon = h - T_0 \cdot s$

where h is the enthalpy, s is the entropy and T_0 is the reference temperature (i.e., 300 K or the reference environment temperature). The usefulness of this intrinsic fluid property is to quantify the reversible (Carnot) work required for given process conditions. This allows the portion of actual input work that is unproductively spent (i.e., wasted due to irreversibilities; the lost work) to be quantified. As an example, consider the table below [2]:

Temperature Range [K]	$T_0 {*} \Delta s$ [J/g]	%	Δh [J/g]	%	$-\Delta\varepsilon$ [J/g]	%
300 to 80	2058	24.5	1143	73.0	915	13.4
80 to 4.22	6329	75.5	421	27.0	5908	86.6
300 to 4.22	8387	100	1569	100	6823	100
4.22 Latent	1469	17.5	20.7	1.3	1449	21.2

Where, $T_0 {*} \Delta s$ = heat rejected = isothermal compressor work

Δh = refrigeration load = expander output work

$\Delta\varepsilon = \Delta h - T_0 {*} \Delta s$ = ideal net input work

Note that the units for enthalpy and exergy are the same and are equivalent to [W/(g/s)]; i.e., mass specific power. $\Delta\varepsilon$ is also known as the mass specific Carnot power.

CARNOT STEP

Typically in helium refrigeration systems, there are multiples of some similar non-simple process steps; e.g., warm compression stages, expansion stages in a cold box, etc., to accomplish a given process.

The ***Carnot Step*** is defined (by the author) as the distribution (or step "spacing") of a given number of similar process steps that yields the minimum irreversibility. This optimal arrangement of process steps is applicable to ideal and real processes and will typically yield the minimum energy expenditure (for that process and selected components). It is important to note that the Carnot Step is not necessarily a reversible 'step', since it depends on whether the process and/or components are reversible.

Before beginning, let's distinguish the three main parts of a typical helium system, the (1) load, (2) cold box and (3) compressor. Clearly, an efficient overall system depends upon these main parts being well matched with each other and an efficient design of each part. Each part may require a number of Carnot steps for minimal losses (i.e., maximum exergy utilization).

The Load

A helium refrigeration system is designed to support specific external load(s); it can be a refrigeration load or liquefaction load, or any combination of the two. The heat energy from the loads is absorbed in the minimum entropy (liquid) helium, thereby increasing the fluid's entropy. Every attempt should be made at the load level (temperature) to minimize the entropy increase of the helium, recovering the exergy of the load return while satisfying the load requirements. Unfortunately, many times a substantial amount of refrigeration (exergy) leaving the load is un-recovered (i.e., wasted). Typical examples of this include targets returning 20K helium or magnet lead flow returning 80K helium which is then warmed to 300K (wasting the refrigeration) or loads designed with excessive pressure drop. This wasted refrigeration as well as losses introduced from a distribution system becomes a load as well, requiring a larger system and resulting in greater capital and operating costs. As an example in attempting to minimize load losses, consider the temperature selection of a single thermal shield to be used between the 300K and 4K to minimize the heat input to the primary (colder) load. Assuming equal conductance on both sides of the shield, the idealized choice for the shield temperate to minimize the total reversible input power (i.e., the load Carnot Step) is found by equating the temperature ratios and is 35K.

The Cold Box

The cold box ***bridges*** the temperature difference from the load temperature to ambient conditions by transferring the entropy increase at the load to the ambient temperature compressors. The cold box is given no input power and can only utilize the availability (i.e., exergy) supplied to it by the compressor(s). Obviously, it is critically important for the cold box to minimize its use of exergy supplied by the compressors and pass it to the load. The cold box provides a process path analogous to carrying a load from a deep basement floor (4.2K) to the ground floor (300K) by walking up the stairs (Figure 1). So, given the 'height' between the 'floors' (4.2K to 300K), we would like to know the ***optimal spacing of a given number of steps*** that will yield a minimum irreversibility. Excluding liquid nitrogen (LN) pre-cooling and Joule-Thompson (JT) effects, the similar non-simple process steps in a cold box are the expansion stages. It is the expanders that provide refrigeration by extracting work, so that the number of expanders (i.e., expansion stages) is equal to the number of steps. If there are expander strings without a heat exchanger in between the expanders, each string is counted as one step (rather than each expander). The cold box Carnot Step provides a means for evaluating the efficiency of a given cold box system design by establishing the ideal temperature 'step' distribution for the expanders.

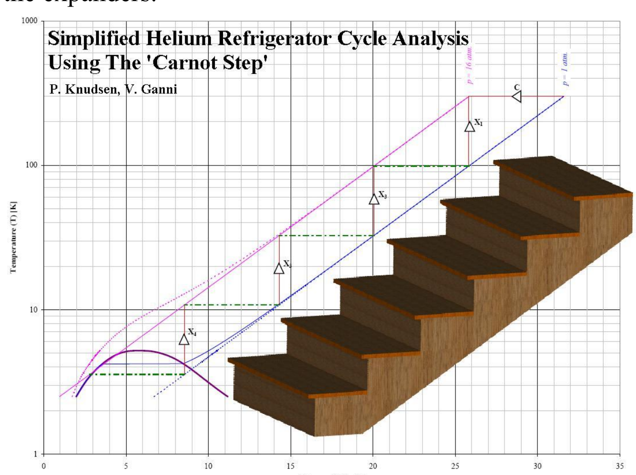

Figure 1: Example of carnot step.

For a liquefier (Figure 2), assuming a constant specific heat ideal gas and 100% effective HX's (called an Ideal Claude Liquefier, or ICL), it can be shown that the temperature distribution resulting in the minimum compressor mass flow for a given (high to low) pressure ratio is an equal temperature ratio for each expansion stage, which corresponds to an equal mass flow though each expander [3]. This is the cold box Carnot step for a liquefier. Note that in Figure 2, the right hand side is the super-position of the left hand side.

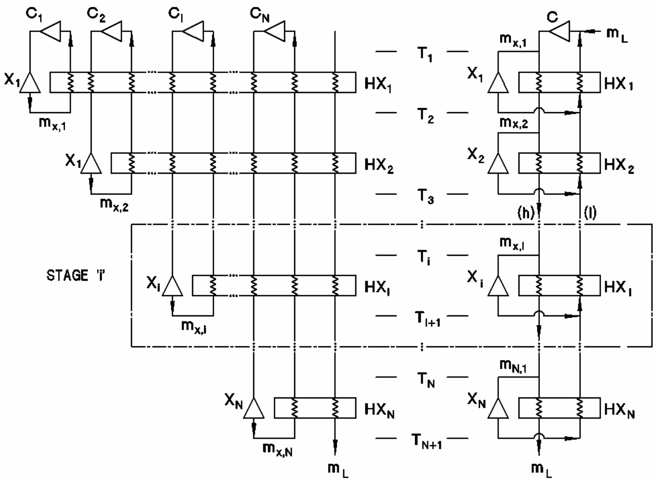

Figure 2: Ideal claude liquefier (ICL).

So, for '*N*' total expansion steps (given), and an arbitrary expansion stage (Carnot step) '*i*',

$$T_r = T_{r,i} = T_i / T_{i+1} = \text{constant}$$

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

And, $\dot{m}_x = \dot{m}_{x,i} = \text{constant}$, for *i*=1 to *N*.

Conversely, the ideal number of Carnot steps is,

$$N = \ln\left(T_{r,T}\right) / \ln\left(T_r\right)$$

Further, if the expander isentropic efficiency is 100% then,

$$T_r = T_{r,i} = P_r^{\phi} \quad \text{where, } \phi = (\gamma - 1)/\gamma$$

With, $P_r = p_h / p_l$, and, γ the ratio of specific heats.

As an example, for a 300 to 4.2K liquefier (e.g., T_1 = 300K, T_{N+1} = 4.2K) with an expander pressure ratio of 16 (i.e., P_r = 16), the total temperature ratio is $T_{r,T}$ = 300 / 4.2 = 71 and the temperature ratio for each expander stage is, T_r = $(16)^{0.4}$ = 3.03 (assuming isentropic expanders). So, the (ideal) number of expander stages required for the ideal Claude liquefier (ICL) is N = *ln* (71) / *ln* (3.03) = 3.85 ≈ 4.

In summary, as long as the expander isentropic efficiencies are equal for each stage, the mass flow and the temperature ratio for each expansion stage (or Carnot step) should be equal for a liquefier [3]. Actual, non-isentropic, expanders result in more expansion stages and/or greater expander mass flow per stage. Additionally, heat exchanger (HX) irreversibilities and the difference in specific heat (C_p) between high and low pressure streams will increase the required mass flow (for a given number of expansion stages). However, the C_p effect has more influence on cold expanders, where the fluid is non-ideal, and the HX irreversibilities will tend to have more influence on the warm expanders since the mass flow is greater at warm end. So, for practical systems these effects will tend to balance each other to some extent and result in the expanders having close to equal flows.

The Compressor System

The compressor system uses the input energy (which is usually electrical) to increase the availability (i.e., exergy) of the helium gas being supplied to the cold box. It compresses helium gas from (nominally) 1 atm to the high pressure required by the cold box design. In practice, this compression is accomplished using multistage compressors. For a multistage polytropic compression process, an equal pressure ratio among each of the equal efficiency stages yields the minimum (mass) specific input work. So, the Carnot step is an equal pressure ratio for each equal efficiency compression stage. The Carnot Step provides a means for evaluating the efficiency of a given compressor system design by establishing the ideal pressure 'step' distribution for the compressor stages.

REAL GAS HELIUM LIQUEFACTION AND REFRIGERATION SYSTEMS

The real gas, non-ideal property influences are theoretically less important to a liquefier than a refrigerator because the sensible cooling load of the fluid [which is 1569 W/(g/s) or a Carnot value of 6823 W/(g/s)] is large and it dominates the process in contrast to condensing the gas into a liquid [20.7 W/(g/s) or the Carnot value of 1449 W/(g/s)]; which, for a refrigerator is ideally the only real cooling requiring work extraction [2].

Main Differences Between a Refrigerator and Liquefier

The arrangement depicted in Figure 2 is also valid for a practical refrigerator in the sense that the liquefaction flow has a similar flow capacity unbalancing effect on the HX as the real fluid property variations and heat leaks. In refrigeration systems the coldest (load, 'wet' or 'JT') expander(s) provide the Carnot power to condense the helium with the remaining expanders dealing with the other previously mentioned losses. The number of expansion stages strongly depends on the size of the system. However, the Carnot steps should be such that the total isothermal compression work required for the losses (referring to Figure 2) is equally divided among the number of stages used. This will not necessarily correspond to an equal temperature ratio distribution.

The main differences between a highly efficient refrigerator and liquefier that influence component choices are:

- A highly efficient liquefier requires many stages of expanders, including the coldest (or load, 'wet', or 'JT') expander. The output power for these expanders is higher than a refrigerator for the same amount of Carnot work. Also, the HX's inherently have an unbalanced flow which results a smaller size HX then would be needed in a refrigerator.
- A highly efficient refrigerator requires much larger HX's (i.e., large HX thermal rating, known as 'UA', and number of transfer units, known as 'NTUs') as they have more balance flow than a liquefier.

However, there are fewer expanders and less output power from these expanders than the liquefier.

In most practical helium systems the expander work is not recovered. In these that do not use LN pre-cooling, the liquefier's Carnot efficiency is lower (by 82.5%) relative to the refrigeration system [2] due to this un-recovered work.

Figure 3 depicts the relative effect of HX's and expanders on a system's load capacity. Note that on an equal Carnot (i.e., equal reversible) work basis, 100W of 4.5K refrigeration is approximately equivalent to 1 g/s of 4.5K liquefaction. For various system designs, it shows the combination of refrigeration and liquefaction loads that particular system can support. For example, an ideal system can support 100W of refrigeration or 1 g/s of liquefaction or 50W + 0.5 g/s of both. A system that is designed as a refrigerator but used as a liquefier (i.e., it is expansion stage limited) will not be able to provide the Carnot equivalent liquefaction capacity that it can provide in refrigeration (e.g., in Figure 3, 150 W of refrigeration compared to 1 g/s of liquefaction). A system that is designed as a liquefier but used as a refrigerator (i.e., it is HX size limited) will not be able to provide the Carnot equivalent refrigeration capacity that it can provide in liquefaction (e.g., in Figure 3, 50W of refrigeration compared to 1 g/s of liquefaction). It is possible to design a system that operates well as a liquefier and as a refrigerator (refer to 'A Balanced Design…" in Figure 3).

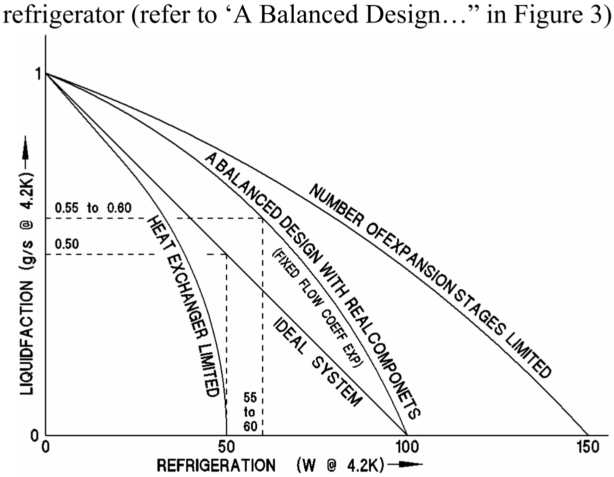

Figure 3: Effect of components on system load capacity.

OPTIMUM SYSTEM

It is non sequitur to ask a system designer (supplier) to provide an optimum system to support a given load, without specifying what the optimum results in, namely:

1. Minimum operating cost
2. Minimum capital cost
3. Minimum maintenance cost
4. Maximum system capacity
5. Maximum system availability

Traditionally a design providing maximum efficiency (minimum operating cost) at a single design point(s) is referred to as the optimum system design.

The optimization of all the above listed constraints has been traditionally considered contradictory. However, the Floating Pressure Process and Ganni cycle is an attempt to provide a system that is optimum in all if the above, considering real component characteristics, and 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.

BASIC FLOATING PRESSURE CYCLE

Consider the basic system shown in Figure 4, consisting of one compressor working with a cold box containing a HX and a turbo expander. This is a simplified arrangement for a typical gas (shield) refrigerator (e.g., 20K systems). For such a system, it can be shown that [2,4] using the Floating Pressure Process (patent pending), the pressure ratio and Carnot efficiency (i.e., the Carnot load power divided by the total input power) remains essential constant over a very wide operating range; i.e. approximately down to 35% for practical systems.

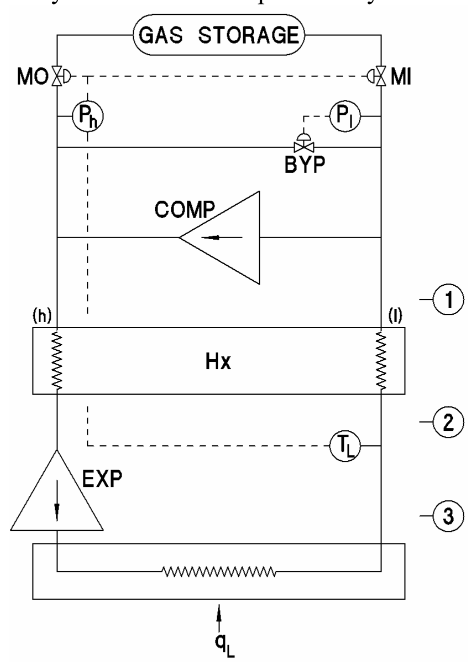

Figure 4: General arrangement for floating pressure process cycle (***patent pending***).

Referring to the TS diagram of the Floating Pressure Process in Figure 5, the area under process path *(h,3)* to *(l,3)*, represents the load (and therefore Carnot load power) per unit mass flow, and process path from *(l,1)* to *(h,1)*, represents the isothermal input work per unit mass flow. It is important to note that these remain constant as the system mass flow varies.

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 properly integrate the compressor efficiency characteristics in the cycle design. Figure 6 shows a typical isothermal efficiency for a 2nd stage (i.e., high pressure stage) compressor [6]. Note that the isothermal efficiency is primarily dependent on the pressure ratio across the compressor.

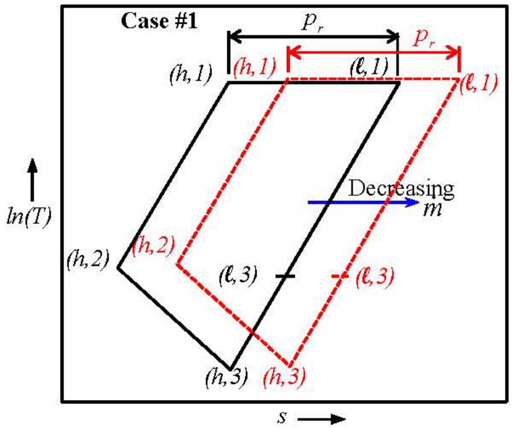

Figure 5: TS diagram of floating pressure process.

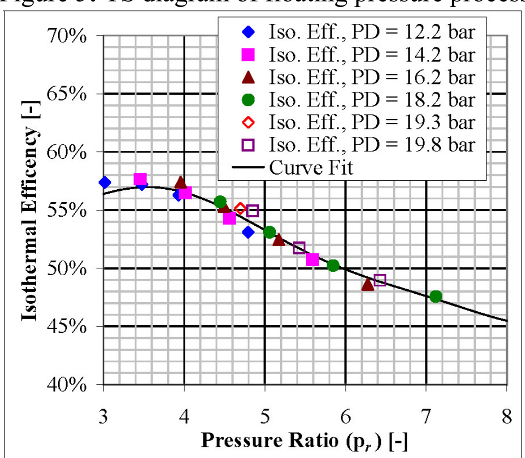

Figure 6: Typical 2nd stage screw compressor isothermal efficiency.

Consider if the turbine's recycle flow is allowed to operate using the Floating Pressure Process and the refrigeration load return is segregated from the turbine recycle return, so as to maintain the lowest possible refrigeration load temperature. Such an arrangement is the Ganni Cycle (US patents 7,278,280 & 7,409,834). Figure 7 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. Such an arrangement allows each stage to operate close to its maximum isothermal efficiency by keeping each stage close to its optimum compression ratio.

APPLICATIONS TO DATE

Some of the aspects of the Floating Pressure Process were adapted to all four plants at JLab in 1994-95. Each of these was manufactured by different vendors. Later similar adaptations were implemented for Michigan State University (MSU) [7], the Spallation Neutron Source (SNS) [8], BNL [9] and for NASA at the Johnson Space Center (JSC) [10]. Each implementation resulted in a substantial improvement of the system's efficiency, capacity, reliability and stability.

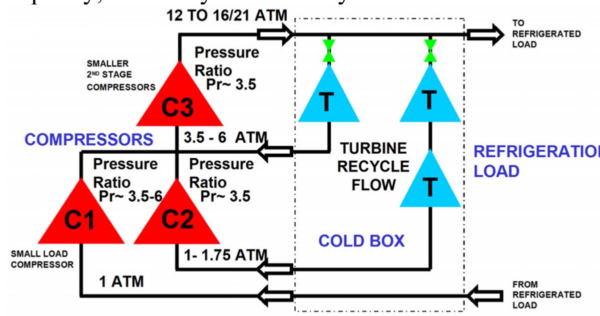

Figure 7: Simplified ganni helium process cycle.

SUMMARY & CONCLUSIONS

This paper is an attempt to present some of the fundamental thermodynamic principles necessary to recognize the quality of energy and provide some practical guide lines for the design and operation of these systems at optimal conditions. The constant pressure ratio Floating Pressure Process and Ganni Cycle application to existing systems and new designs addresses the quality and efficient use of energy and offers a solution to the "Optimal Design and Operation of Helium Refrigeration Systems". This process has been licensed by JLab to "Cryogenic Plants and Services" a Division of Linde BOC Process Plants, LLC for world wide commercialization.

REFERENCES

[1] Kotas, T.J., The Exergy Method of Thermal Plant Analysis, Krieger Publishing, 1995.

[2] Ganni, V., Design of Optimal Helium Refrigeration and Liquefaction Systems, CEC-ICMC 2007 - CSA Short Course Symposium.

[3] Knudsen, P., Ganni, V., "Simplified Helium Refrigerator Cycle Analysis Using the Carnot Step", Adv. Cryo. Eng., 51B, 2006, pp. 1977-1986.

[4] Ganni, V., Knudsen, P., "Optimal Design and Operation of Helium Refrigeration Systems Using the Ganni Cycle", *to be published in* CEC-ICMC Proc., Tucson, AZ, June 2009.

[5] Ziegler, B. O., "Second Law Analysis of the Helium Refrigerators for the HERA Proton Magnet Ring", in Advances in Cryogenic Engineering 31, edited by R. Fast, Plenum Press, New York, 1985, pp. 693-698.

[6] Ganni, V., et al., "Screw Compressor Characteristics for Helium Refrigeration Systems", Adv. Cryo. Eng., 53A, 2008, pp. 309-315.

[7] McCartney, et al., "Cryogenic System Upgrade for the National Superconducting Cyclotron Laboratory", Adv. Cryo. Eng., 47A, 2002, pp. 207-212.

[8] Arenius, D., et al., "Cryogenic System for the Spallation Neutron Source", Adv. Cryo. Eng., 49A, 2004, pp, 200-207.

[9] Than, R., et al., "The RHIC Cryogenic System at BNL: Review of the modifications and upgrades since 2002 and planned improvements", Adv. Cryo. Eng., 53, 2008, pp. 578-587.

[10] Homan, J., et al., "Floating Pressure Conversion and Equipment Upgrades of Two 3.5kW 20-K Helium Refrigerators", *to be published in* CEC-ICMC Proc., Tucson, AZ, June 2009.