PROCESS OPTIONS FOR NOMINAL 2-K HELIUM REFRIGERATION SYSTEM DESIGNS

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ABSTRACT

Nominal 2-K helium refrigeration systems are frequently used for superconducting radio frequency and magnet string technologies used in accelerators. This paper examines the trade-offs and approximate performance of four basic types of processes used for the refrigeration of these technologies; direct vacuum pumping on a helium bath, direct vacuum pumping using full or partial refrigeration recovery, cold compression, and hybrid compression (i.e., a blend of cold and warm sub-atmospheric compression).

KEYWORDS: helium, refrigeration, 2 K process, sub-atmospheric process, cycles, efficiency

INTRODUCTION

Refrigeration below 4.5 K typically involves sub-atmospheric helium at some point in the process. We will refer to these as nominally 2-K, since processes used for large accelerators operate close to this temperature and typically between 1.8 and 2.1 K. These processes may use cold (cryogenic) compressors, ambient temperature ('warm') vacuum pumps or a combination ('hybrid'). The cold compressors are usually centrifugal machines and can range from wheel diameters of 8 cm to 37 cm, operate as high as approximately (~) 700 Hz, have isentropic efficiencies from 50 to 70% and usually use magnetic bearings [1-6]. The warm vacuum pumps are usually some combination of lobe blowers, rotary vane pumps, liquid ring pumps and rotary screw compressors. Their size can range up to 225 kW, usually operate at a suction pressure of 16 to 600 mbar and involve at least two stages (of usually differing types) [1-6].

With this range of equipment options available for process arrangements and variance in equipment performance and operation, before selecting a 2-K process and the

corresponding equipment required, we should begin by addressing some fundamental questions. Namely,

1. Should all cold compression, all warm compression or some hybrid option be selected and why?
2. What are the dominate parameters affecting the overall system performance and efficiency?
3. What are some approximate performance expectations possible for various 2-K processes?
4. Since there is typically no work extraction between the 4.5-K refrigerator supply and the 2-K load, can the 2-K load capacity be increased by any process changes in between these temperature levels?

We will seek to answer the first question qualitatively and along with the second and third questions, by means of a parametric process study. The last question addresses the fact that 2-K processes typically provide an enthalpy flux of around 20 J/g to the load. However, in our range of interest, the latent heat is ~15% higher than this value. Additionally, the ratio of the supply to sub-atmospheric stream heat capacity is favorable for a small stream temperature difference at the 2-K load. As previously mentioned, since typical 2-K processes do not have work extraction below the refrigerator supply temperature (nominally 4.5 K), it is crucial to utilize the exergy supplied by the 4.5-K refrigerator in the most efficient and practical manner possible.

Cold or Warm Compression

We will note that [7] attempted to address this important question. Disregarding equipment sizes, neglecting pressure drop and assuming equal warm and cold compression efficiencies, using cold compressors will result in a lower 2-K process efficiency due to the necessarily increased mass flow inherent from the flow imbalanced created in the 4.5-K refrigerator. This can be visualized in an idealized process as depicted in FIGURE 1, where 'C' indicates warm positive pressure compression, 'VP' indicates warm sub-atmospheric compression (i.e., vacuum pumping), 'CC' indicates cold compression, 'X' indicates expansion from a high to lower pressure stream, '*x*' indicates expansion of the high pressure stream, 'B1' and 'B2' indicate balanced mass flow sections (with 'B2' between 4.5 and 2-K) and 'U1' indicates an unbalanced flow section. The following should be noted from FIGURE 1:

(a) For no cold compression, the entire process has balanced mass flow.
(b) The number of expansion steps in the balanced sections can be estimated from [8], where the temperature ratio for each expansion step is equal and minimizes the heat exchanger stream temperature difference losses. However, [8] will substantially under-predict the actual pressure ratio required as shown in [9,10] due to the work extraction required to gain the latent heat. [11] presents a method that is the same principal as [8] but allows a more practical pressure ratio (i.e., to better match real equipment).
(c) The number of expansion steps in the unbalanced section is estimated from [12], where the temperature ratio for each expansion step is equal and minimizes the mass flow. It must be emphasized that the fundamental reason for the equal temperature ratio being optimum for the balanced and unbalanced sections is not the same. Also, [13] properly understood and visualized the equal temperature ratio for each expansion step.

(d) Higher cold compression efficiency reduces the total temperature ratio of the unbalanced section ('U1') and therefore reduces the high pressure stream mass flow required.

(e) Although the input power required for the cold compression is proportional to the suction temperature, for a given cold compression (total) pressure ratio the suction temperature has negligible effect on the input power required for the 4.5-K refrigerator which is much greater than the input power required for the cold compression. So, the effect of lowering the cold compression suction temperature is to reduce the equipment size.

(f) In the no cold compression (and to a lesser extent in the partial cold compression) case, recovering the refrigeration from the sub-atmospheric stream involves a bit of a 'catch-22' situation. That is, to minimize the vacuum pump size and its input power, it is important to minimize the pressure drop (in the sub-atmospheric stream). However, this requires a larger heat exchanger (HX) to both meet the lower pressure drop requirement and ensure proper heat transfer (i.e., pressure drop is needed for adequate heat transfer). Note that as shown in FIGURE 1, it is assumed that the vacuum pump is 'warm' and operates at or above some minimum suction temperature (say, > 275 K).

Smaller capacity refrigerators will typically be less efficient than larger refrigerators since larger systems tend to have more expansion steps and the turbo-expanders (usually used in larger systems) have a higher efficiency as their size increases [9]. So, for small 2-K loads, not using cold compression can yield a more efficient system (assuming equal vacuum pump and cold compressor efficiencies). However, using no cold compression for larger 2-K loads will result in quite large equipment sizes (i.e., both vacuum pumps and HX's) and the portion of the total input power required for the vacuum pump will be much higher as a result of improved refrigerator efficiencies. By small loads we mean systems less than 200 to 300 W [14,15].

PARAMETRIC PROCESS STUDY

In order to investigate the dominate parameters for 2-K process that affect the overall system efficiency and to determine some approximate performance expectations possible for various 2-K processes, a parametric study was performed. FIGURE 2 shows the basic four types of processes examined. Type-1 is direct vacuum pumping, with or without a 4.5 to 2-K HX (i.e., HX-2). Type-2 utilizes an unbalanced refrigeration recovery HX and no cryogenic rotating machinery. Type-3 has several variants (which are not all shown in FIGURE 2), but each uses partial cold compression, each with a different number of cold

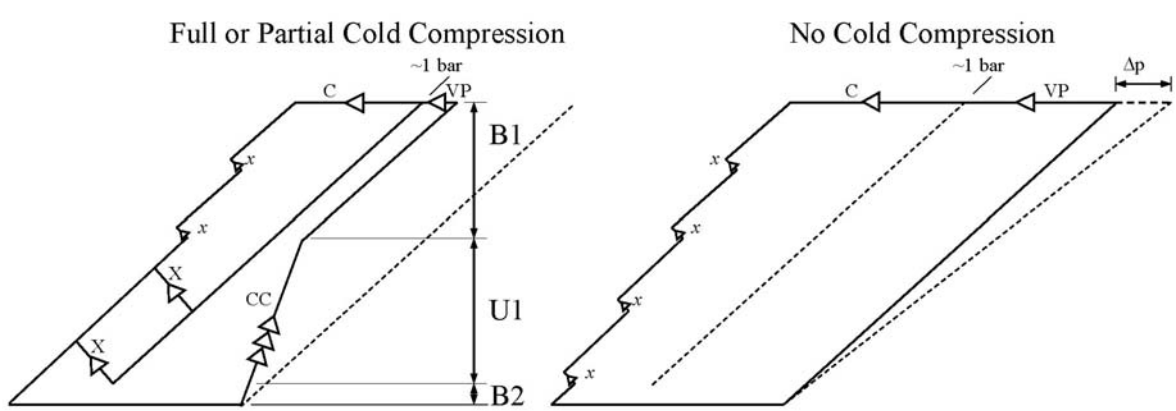

FIGURE 1. 2-K Process (Log T vs. s)

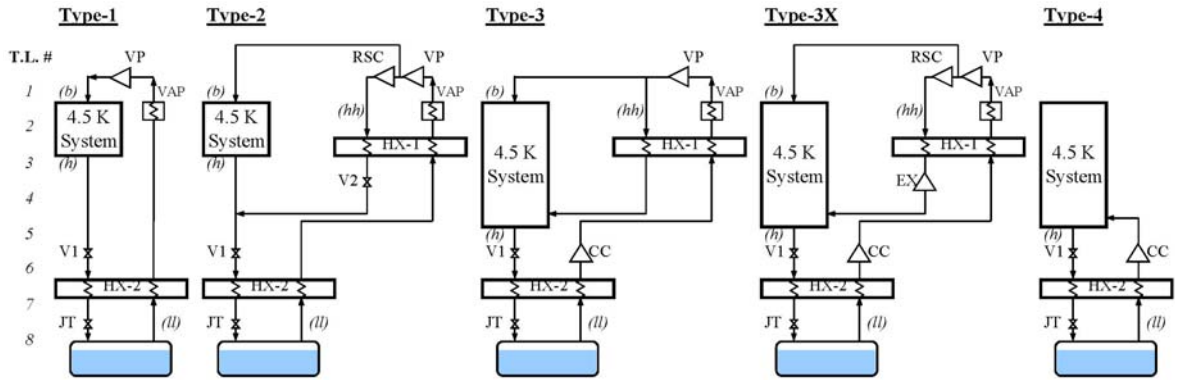

FIGURE 2. Process Configurations

compression stages and an unbalanced refrigeration recovery HX. Type-3X employs an expander to extract work from the higher pressure stream {labeled'*(hh)*'} before injecting into the 4.5-K cold box system. Both types-3(3C) and 3X use three stages of (equal pressure ratio) cold compression. Types-3(1C) and 3(2C) are the same as type-3(3C) except use one and two stages of cold compression, respectively. Type-4 uses all cold compression (four stages). The "4.5-K System" denotes the 4.5-K cold box and compressor system, which supports the 4.5-K cold box. In types 1 and 2, the 4.5-K system operates as a liquefier (i.e., unbalanced flow from 300 to 4.5 K), where in the other types it operates as only a partial liquefier (i.e., unbalanced from injection to 4.5 K). Note that neither types 2 or 3 are the same as the configuration in FIGURE 1 for no cold compression. Such a configuration was not examined because it is not a practical cycle for the presently available (commonly adopted) components as it requires a much higher pressure than typically used for helium systems [9,10]. TABLE 1 shows the parameters varied, as applicable for each configuration type; namely, HX-1 (300 to 4.5-K) Ntu's, HX-2 (4.5 to 2-K) Ntu's, cold compressor isentropic efficiency (per stage), expander adiabatic efficiency (type-3X only) and the high pressure stream supply pressure.

The following were used for the process study,

- 2.00 K (31.4 mbar) load with a mass flow rate of 100 g/s
- No heat in-leak for the HX's
- 3 bar supply pressure from the 4.5-K system
- 1.216 bar injection pressure to 4.5-K system {type-3(1C), 3(2C), 3(3C), 3X and 4}
- 1.085 bar vacuum pump discharge pressure
- 0.3 bar high pressure stream pressure drop in HX-1 (except for type-2, use 0.1 bar)
- 0.05 bar high pressure stream pressure drop in HX-2
- 0.4 bar pressure drop for compressor oil removal
- 0.02 bar pressure drop for vacuum pump oil removal
- 302 K discharge temperature for compressor and vacuum pump
- 292 K suction temperature to vacuum pump
- 2.515 maximum pressure ratio across any cold compressor stage
- 75% cold compressor motor efficiency
- 95% compressor and vacuum pump motor efficiency

The sub-atmospheric stream pressure drop was characterized by assuming that the ratio of the pressure drop to the upstream pressure scaled with the total sub-atmospheric stream HX Ntu's, and was the same for each HX (including the ambient vaporizer, 'VAP', in FIGURE 2). For this study, it was assumed that this was 25% (pressure drop to the upstream pressure) per 50 Ntu's.

TABLE 1. Varied Process Parameters

Parameter	Configuration Type					Range Varied
	-1	-2	-3	-3X	-4	
HX-1 Ntu's		■	■	■		25, 30, 35, 40, 45
HX-2 Ntu's	■				■	0, 0.5, 1, 3.5, 5
CC Isentropic Efficiency			■	■	■	60, 70, 80, 90, 100%
Expander Adiabatic Efficiency				■		0, 60, 70, 80, 90%
High Pressure		■		■		6, 9, 12, 15, 18 bar

Note: Type-3 configuration includes type-3(1C), type-3(2C) and type-3(3C) which have one, two and three stages of cold compression, respectively.

4.5-K System Characterization

Even for the same number of expansion steps, a liquefier (i.e., unbalanced mass flow from 300 to 4.5-K) will have a lower overall exergetic efficiency than a refrigerator (i.e., balanced mass flow) since the work extracted is usually not recovered (for helium systems) [9]. The ratio of the reversible work utilizing the extracted work (to reduce the required input power) to the reversible work where the extracted work is not utilized is, $\upsilon = \left(w_{rev} / \tilde{w}_{rev}\right) = 1 - \Delta h / \left(T_0 \cdot \Delta s\right)$; where Δh is the enthalpy flux [J/g], Δs is the entropy flux [J/g-K] and T_0 is the reference temperature (usually the ambient temperature; ~ 300 K). If we designate υ^* for the refrigerator and υ for the 4.5-K system handling a partial liquefaction load, then we can scale the overall exergetic efficiency (of the 4.5-K system) by, $\eta = \left(\upsilon / \upsilon^*\right)\eta^*$; where, η^* is the overall exergetic efficiency of the refrigerator (i.e., balanced mass flow). By 'overall exergetic efficiency', we mean to ratio of the load exergy flux to the total input power required. We will assume that the supply conditions for υ^* and υ are equal and that the return condition for υ^* is saturated vapor at 1.25 bar. FIGURE 3 provides an idea of what numerical values result from this scaling as the return (injection) temperature is varied, assuming a supply condition of 3 bar and 4.5 K from the 4.5-K system. Note that 'pR' is the return (injection) pressure to the 4.5-K system.

Vacuum Pump and Compressor Characterization

Although there is sufficient data [16] to adequately characterize the isothermal efficiency of screw compressors, there is limited data to do the same for various vacuum pumps [15]. Using the aforementioned, we developed a rough characterization, as shown in FIGURE 4, for the isothermal efficiency applicable for vacuum pumps and (positive pressure) compressors.

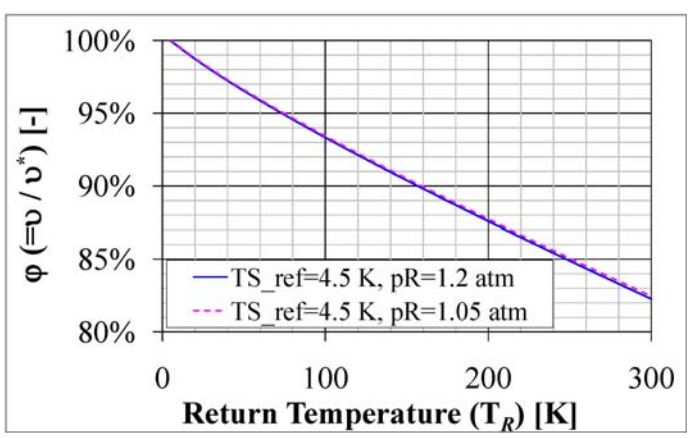

FIGURE 3. Scaling of 4.5-K System Exergetic Efficiency

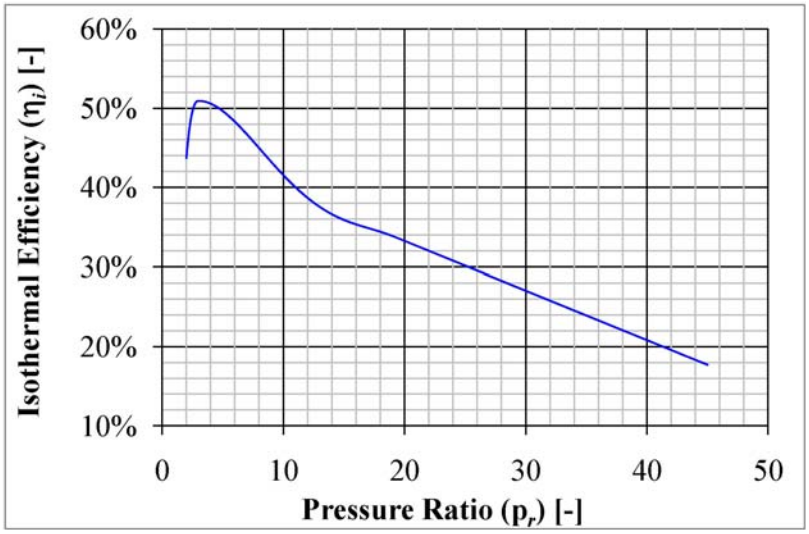

FIGURE 4. Vacuum Pump and Compressor Isothermal Efficiency

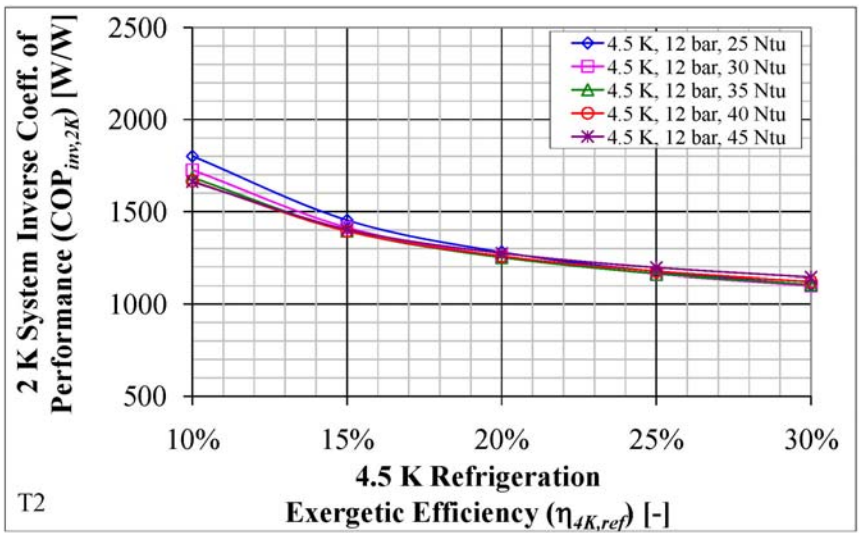

FIGURE 5. Type-2 Performance vs. $\eta_{4K,ref}$ for Various HX-1 Sizes

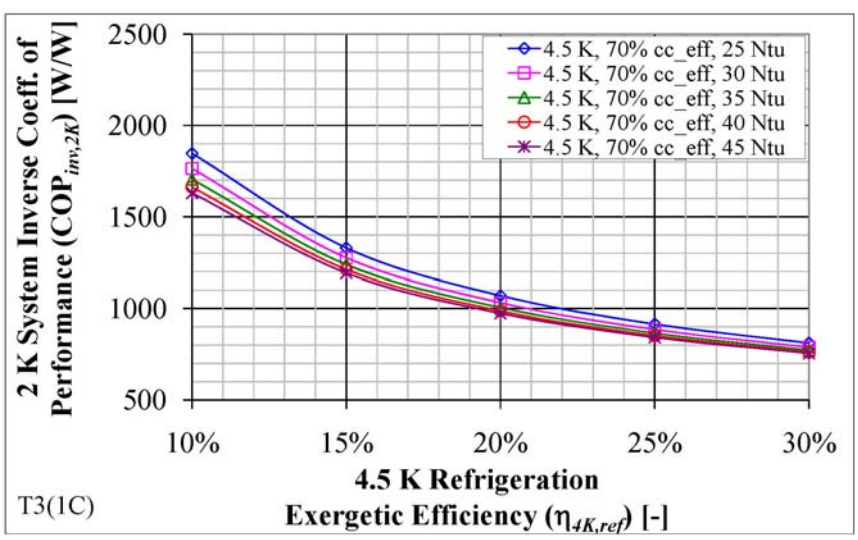

FIGURE 6. Type-3(1C) Performance vs. $\eta_{4K,ref}$ for Various HX-1 Sizes

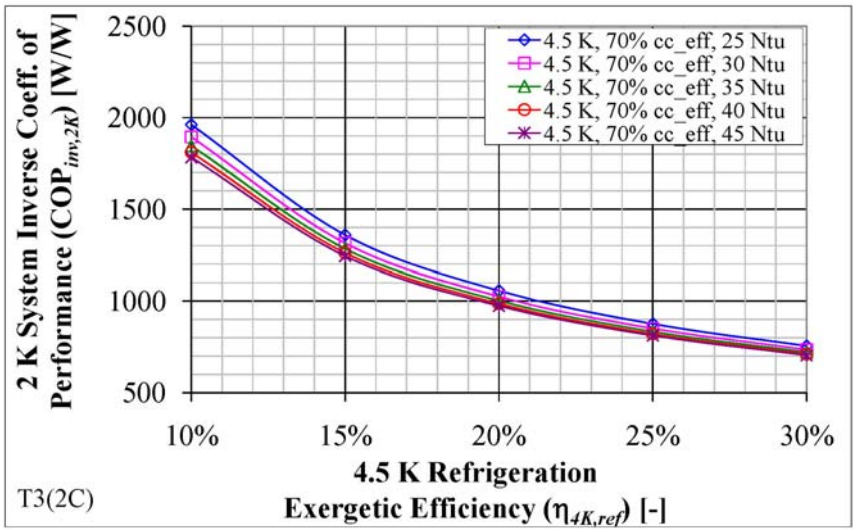

FIGURE 7. Type-3(2C) Performance vs. $\eta_{4K,ref}$ for Various HX-1 Sizes

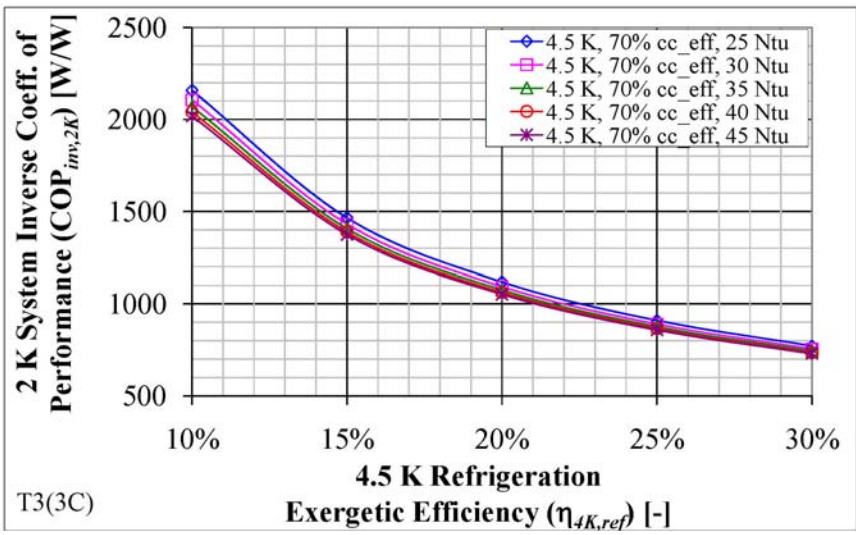

FIGURE 8. Type-3(3C) Performance vs. $\eta_{4K,ref}$ for Various HX-1 Sizes

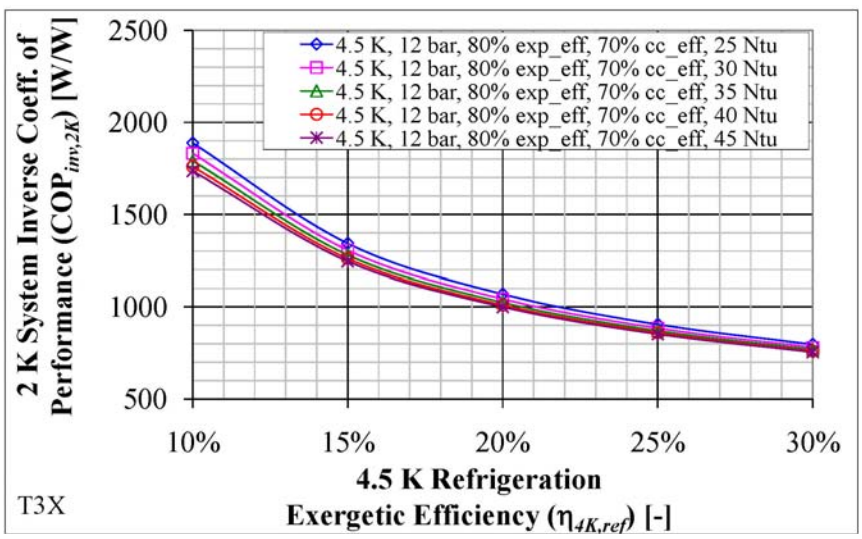

FIGURE 9. Type-3X Performance vs. $\eta_{4K,ref}$ for Various HX-1 Sizes

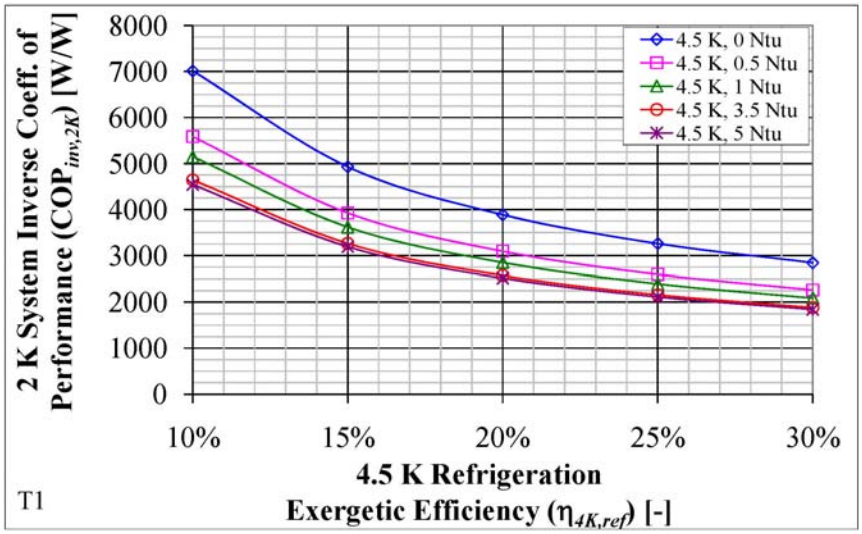

FIGURE 10. Type-1 Performance vs. $\eta_{4K,ref}$ for Various HX-2 Sizes

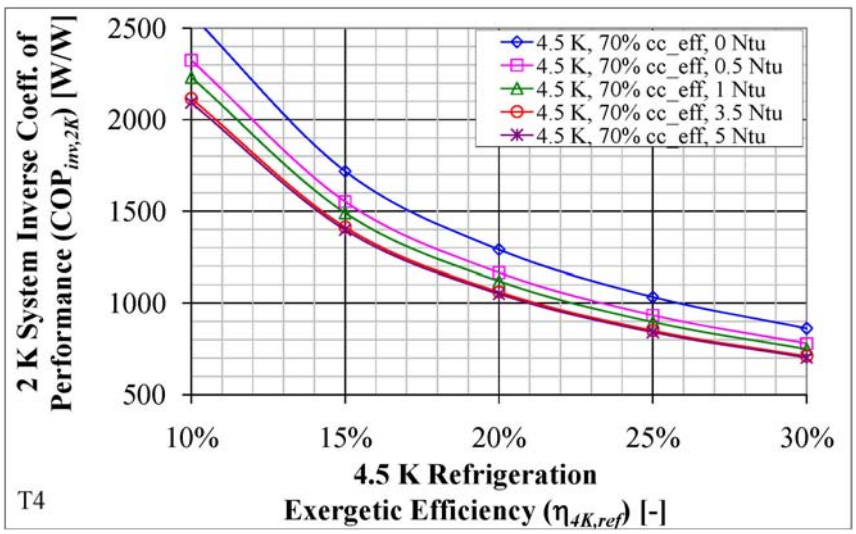

FIGURE 11. Type-4 Performance vs. $\eta_{4K,ref}$ for Various HX-2 Sizes

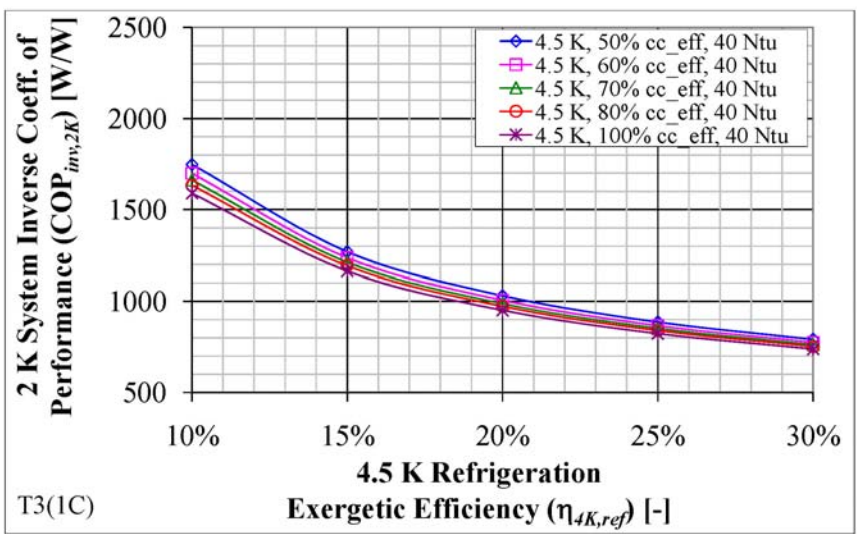

FIGURE 12. Type-3(1C) Performance vs. $\eta_{4K,ref}$ for Various Cold Compressor Isentropic Efficiencies

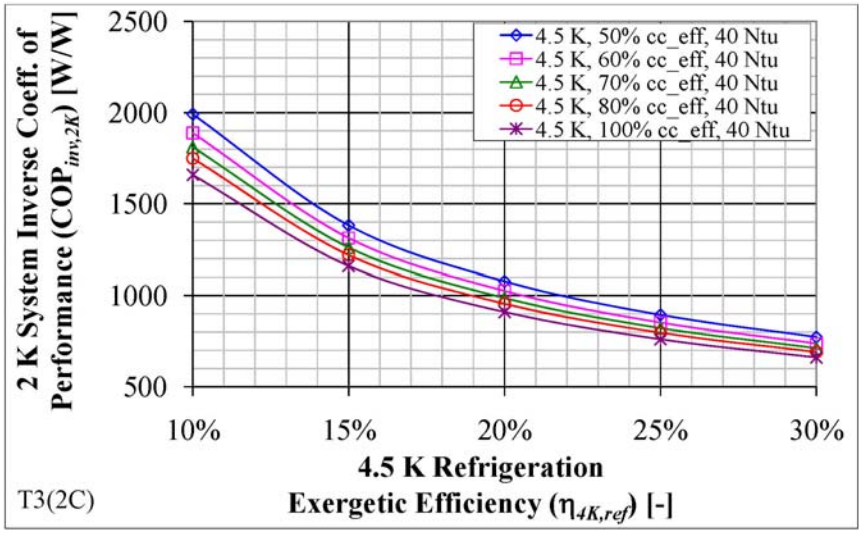

FIGURE 13. Type-3(2C) Performance vs. $\eta_{4K,ref}$ for Various Cold Compressor Isentropic Efficiencies

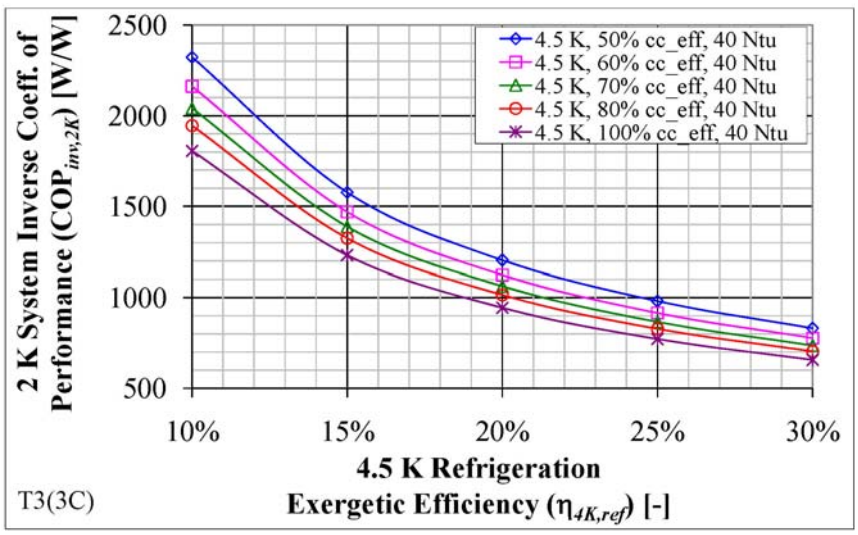

FIGURE 14. Type-3(3C) Performance vs. $\eta_{4K,ref}$ for Various Cold Compressor Isentropic Efficiencies

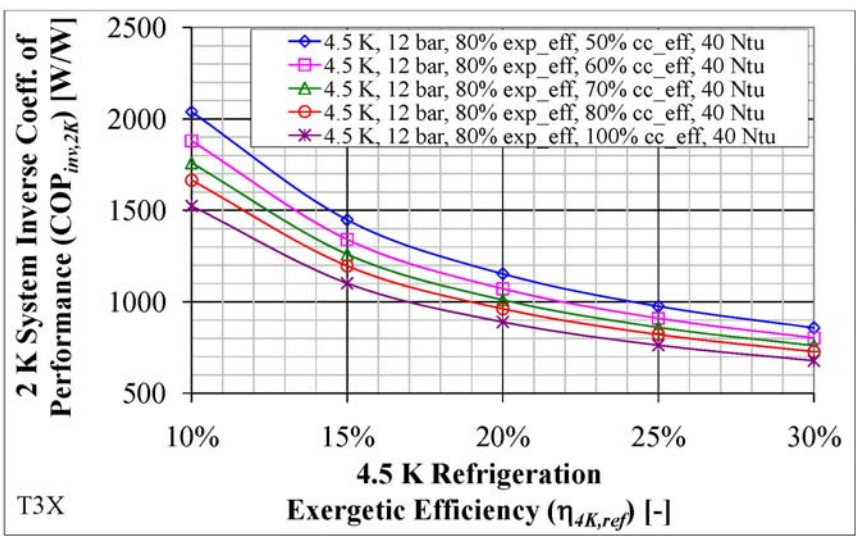

FIGURE 15. Type-3X Performance vs. $\eta_{4K,ref}$ for Various Cold Compressor Isentropic Efficiencies

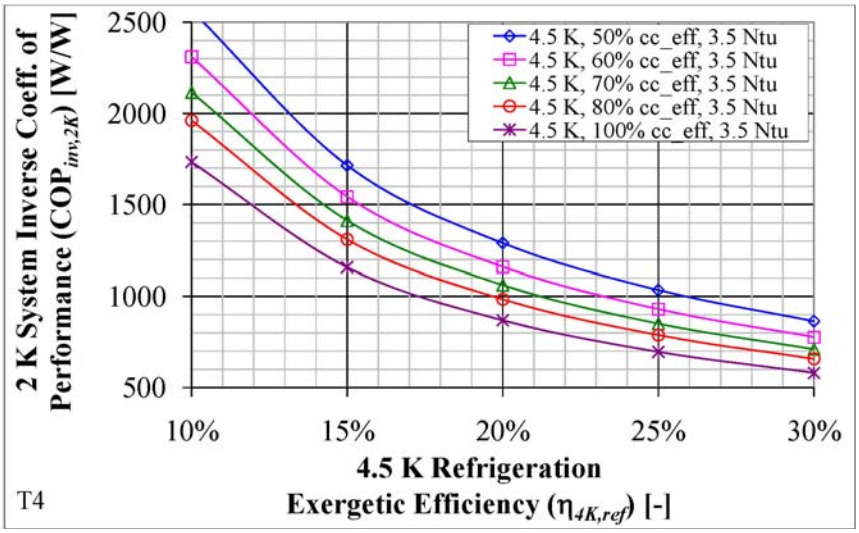

FIGURE 16. Type-4 Performance vs. $\eta_{4K,ref}$ for Various Cold Compressor Isentropic Efficiencies

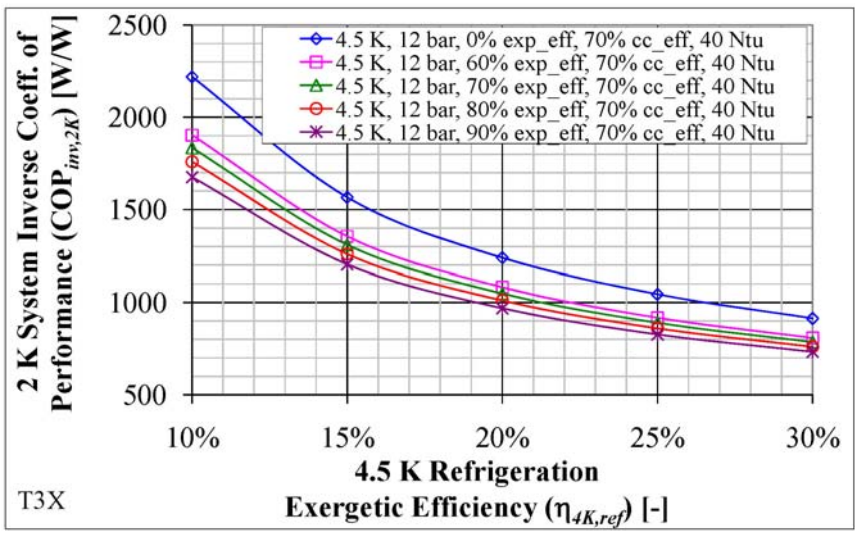

FIGURE 17. Type-3X Performance vs. $\eta_{4K,ref}$ for Various Expander Adiabatic Efficiencies

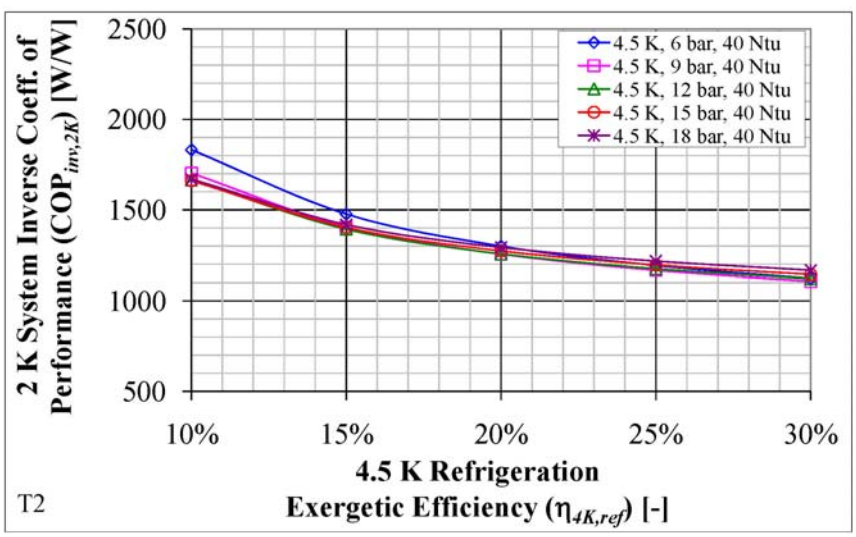

FIGURE 18. Type-2 Performance vs. $\eta_{4K,ref}$ for Various High Pressure Supply Pressures

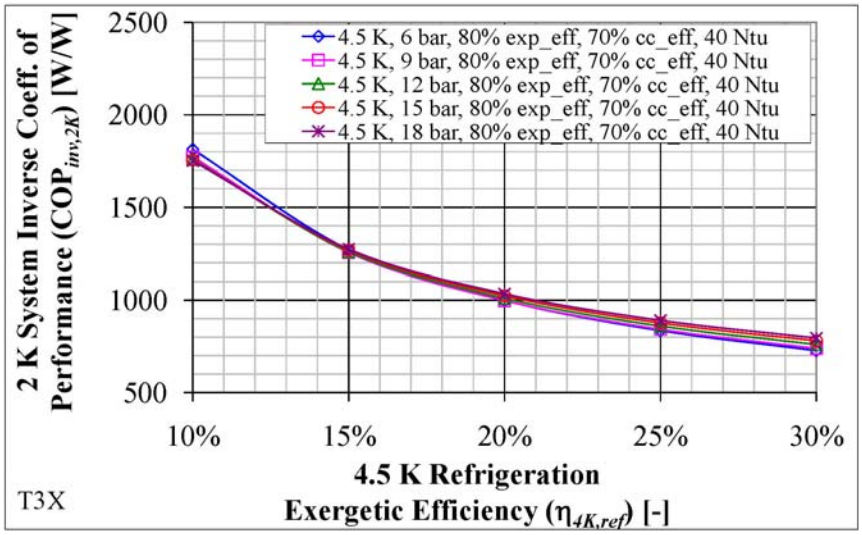

FIGURE 19. Type-3X Performance vs. $\eta_{4K,ref}$ for Various High Pressure Supply Pressures

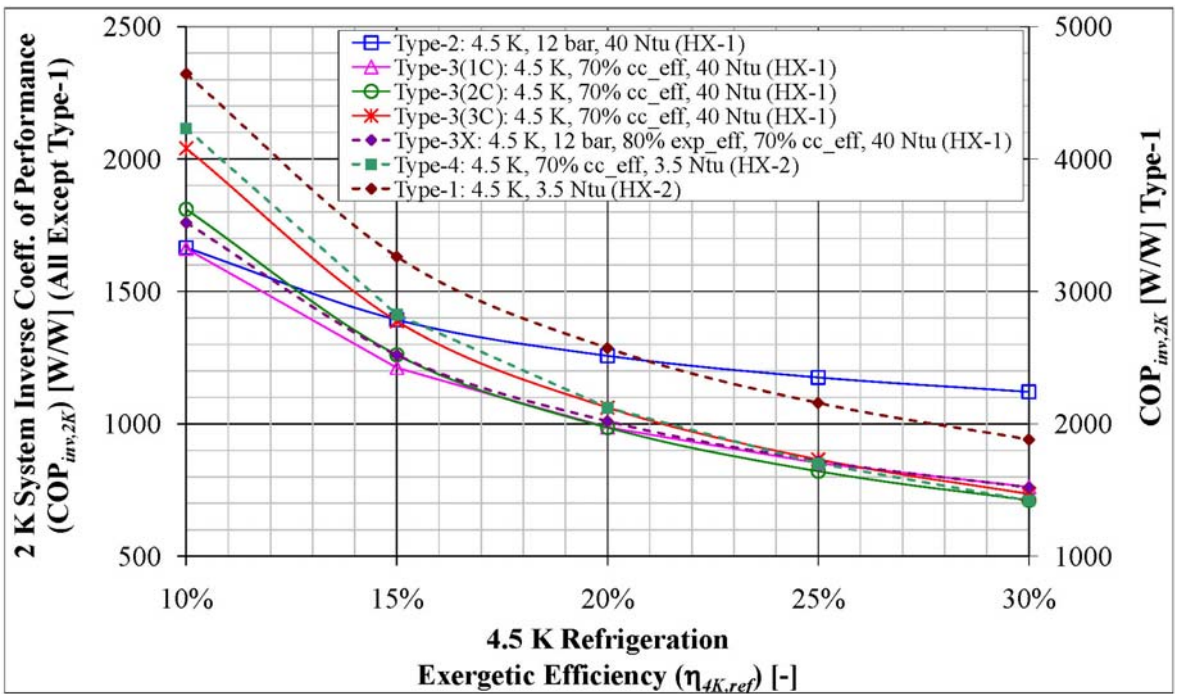

FIGURE 20. Performance vs. $\eta_{4K,ref}$ for Various 2-K Process Types at Nominal Parameters

RESULTS

As is evident from FIGURES 5 to 19, the overall exergetic efficiency of the 4.5-K system ($\eta^* = \eta_{4K,ref}$ in the figures) is by far the most influential parameter governing the performance of the 2-K process. In general, the next most influential parameter is the cold compressor (adiabatic) efficiency {for types-3(3C), 3(1C), 3(2C) and 4}. The expander efficiency for type-3X and the size of HX-2 for types-1 and 4 are also influential. However, as noted by comparing the performance amongst other types, this influence should be viewed as providing a minimum equipment requirement if these process options are to be selected. As seen from FIGURE 20, type-2 and type-3(1C) appear to be more efficient than the other types at low $\eta_{4K,ref}$ (i.e., small loads) and type-2 does not vary as much as

the other processes over the range of $\eta_{4K,ref}$ shown. Partial cold compression {types-3(2C) and 3X} can be almost as efficient as type-2 or type-3(1C) at very low $\eta_{4K,ref}$, but all types using any amount of cold compression tend to become close at high $\eta_{4K,ref}$ (i.e., large systems). It is obvious that type-1 is not an efficient process, requiring ~2 times more input power (for the same load) than the next most inefficient process. This process should be avoided for all but very small loads needed for a very short one time use [17].

COLD END PROCESS IMPROVEMENT

Since there is typically no work extraction below the 4.5 K level, the 4.5 to 2-K HX (HX-2 in FIGURE 2) is crucial in maximizing the exergy supplied to the load from the 4.5-K system. For thermo-hydraulic reasons, the supply pressure to the load is usually around 3 bar (super-critical) for large systems and around 1.2 bar (saturated liquid) for small systems. However, the load is sub-atmospheric, so this loss in availability (across the 'JT' valve) seems unavoidable. FIGURE 21 shows that for a cold (load) end temperature difference of 0.2 K (in order to provide the stream temperature difference necessary for heat transfer and to avoid super-fluid in the HX), the enthalpy flux (difference) supplied to the load is 20.0 J/g for a 3 bar pressure out of the HX, and 21.2 J/g for a 1.2 bar pressure out of the HX. Yet, the latent heat at 2.00 K is 23.4 J/g. If the supply stream {labeled '*(h)*' in FIGURE 22} pressure drop is used in the HX in such a way that the higher pressure stream reaches, say 0.2 bar, at the cold end of the HX, the enthalpy flux to the load will be 21.85 J/g. Rather than rely upon fine tuning of a continuous pressure drop through the HX (which is ideal for single point operation but not practical for capacity variation), of perhaps of more practical interest is to divide the HX into two sections with the pressure drop taken across a JT valve in between these sections, which is also used to control the liquid level at the load. FIGURE 22 shows both of these configurations. The device labeled 'V2', which could operate passively such as a gravity check valve, maintains the desired outlet pressure on the cold end of the HX, thereby indirectly providing a sufficient stream temperature difference necessary for heat transfer. This implementation results in a modest 3% load capacity increase at the same input power (or conversely, a 3% input power decrease for the same load) for the 1.2 bar (saturated liquid) supply pressure. However, this is a 9.3% load capacity increase for a 3 bar supply pressure, as would be common in larger systems! Theoretically, the outlet pressure could be allowed to decrease to the load pressure (resulting in the maximum enthalpy flux; 23.4 J/g for a 2.00 K load). However, as shown in [18], this would require an infinite length HX. The selection of 0.2 bar represents a practical value, below which the HX size rapidly grows.

CONCLUSION

The most influential parameter for 2-K processes is the 4.5-K system overall exergetic efficiency. However, both the vacuum pump and cold compressor efficiencies are also very influential, though to a lesser extent. Interestingly, a type-2 configuration may be the best option for smaller systems with lower 4.5-K system efficiencies, though the pressure drop characterization (for type-2) may not be sufficiently conservative for higher HX Ntu's. Some partial cold compression configurations {types-3X, -3(1C) and -3(2C)} also appear to be attractive at low 4.5-K system efficiencies. Type-4 and type-3(3C) are not as

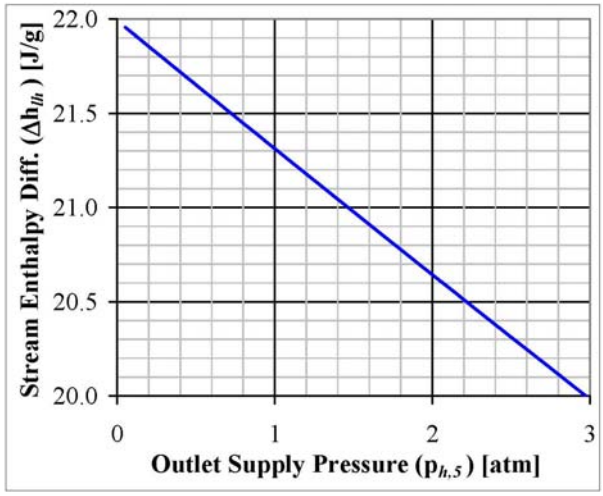

FIGURE 21. Cold End HX Higher Pressure Stream Outlet Condition vs. Enthalpy Flux to the Load

efficient as type-2 and the other type-3's at low 4.5-K system efficiencies since the vacuum pump contributes less to the required input power (for the other type 3's) than does the 4.5-K system for type-4. However, as the 4.5-K system efficiency increases (typically as the system size increases), the 2-K system efficiency using partial and all cold compression configurations {types-3X, 3(1C), 3(2C), 3(3C) and 4} are all close and better than type-2. For an existing 4.5-K system that has a low efficiency, type-3X {as compared to type-3(3C)} can improve its 2-K capacity. A type-4 configuration avoids sub-atmospheric conditions in the HX's (except for HX-2) and at the ambient temperature level, thereby eliminating the possibility of air in-leakage into the process and also providing a minimum equipment size. This has a strong influence on the long term reliable operation and the capital cost (mostly due to the smaller equipment foot-print without vacuum pumps). The pump-down for type-4 is a bit more interesting, since there are no warm vacuum pumps allowing a partial pump-down (to some intermediate sub-atmospheric pressure) prior to starting the cold compressors.

Further, a cold-end process configuration has been presented that could allow an input power reduction for the same load of close to 10% for a 3 bar load supply pressure common to large systems.

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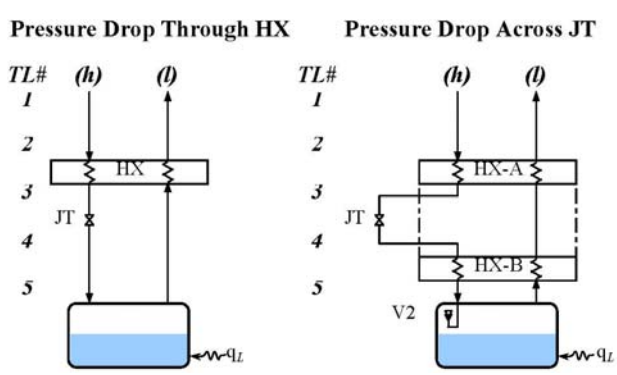

FIGURE 22. Possible 2-K Process Cold End Configurations

REFERENCES

1. Adler, E., et al, "Refrigeration at 1.8 K by Using a Combination of Warm Screw Compressors and Cold Two-Stage Turbo-Compressors," in *Proceedings of the 17th International Cryogenic Engineering Conference (ICEC17)*, July 14-17, Institute of Physics, Bristol, UK, 1998, pp. 101-108.
2. Bon Mardion, G., "Initial Operation of the 1.8 K Tore Supra Cryogenic System," ," in *Proceedings of the 12th International Cryogenic Engineering Conference (ICEC12)*, July 12-15, University of Southampton, Southampton, UK, 1988, pp. 511-518.
3. Gistau, G.M., et al, "The 300 W – 1.75 K Tore Supra Refrigerator – Cold Centrifugal Compressor Report," in *Advances in Cryogenic Engineering* 33, edited by R.W. Fast et al, Plenum, New York, 1988, pp. 675-681.
4. Gistau, G.M., et al, "The Tore Supra 300 W – 1.75 K Refrigerator," in *Advances in Cryogenic Engineering* 31, edited by R.W. Fast et al, Plenum, New York, 1986, pp. 607-615.
5. Roussel, P., et al, "The 400 W at 1.8 K Test Facility at CEA-Grenoble," in *Advances in Cryogenic Engineering* 51, edited by J.G. Weisend II et al, AIP, New York, 2005, pp. 1420-1427.
6. Arenius, D., Thomas Jefferson National Accelerator Facility, private communication, May 2008.
7. Quack, H., "Cold Compression of Helium for Refrigeration Below 4 K," in *Advances in Cryogenic Engineering* 33, edited by R.W. Fast et al, Plenum, New York, 1988, pp. 647-653.
8. Trepp, C., "Refrigeration Systems for Temperatures Below 25 K With Turboexpanders," in *Advances in Cryogenic Engineering* 7, edited by K.D. Timmerhaus et al, Plenum, New York, 1962, pp. 251-261.
9. Ganni, V., "Design of Optimal Helium Refrigeration and Liquefaction Systems: Simplified Concepts & Practical Viewpoints," Cryogenic Society of America Short Course Symposium, June 28, Tucson, AZ, 2009.
10. Ziegler, B., Quack, H., "Helium Refrigeration at 40 Percent Efficiency?," in *Advances in Cryogenic Engineering* 37, edited by R.W. Fast et al, Plenum, New York, 1992, pp. 645-651.
11. Quack, H., "Maximum Efficiency of Helium Refrigeration Cycles Using Non-Ideal Components," in *Advances in Cryogenic Engineering* 39, edited by P. Kittel et al, Plenum, New York, 1994, pp. 1209-1216.
12. Knudsen, P., Ganni, V., "Simplified Helium Refrigeration Cycle Analysis Using the 'Carnot Step', " in *Advances in Cryogenic Engineering* 51, American Institute of Physics, New York, 2006, 1977-1986.
13. Collins, S.C., Cannaday, R.L., "Expansion Machines for Low Temperature Processes," Oxford University Press, 1958, pp. 38-40.
14. Peterson, T., "Status: Large-Scale Subatmospheric Cryogenic Systems," in *Proceedings of the Particle Accelerator Conference* (PAC89), March 20-23, Chicago, IL, IEEE, 1989, pp. 1769-1773.
15. Knudsen, P., "Process Study for the Design of Small Scale 2 Kelvin Refrigeration Systems," MSc. Thesis, Old Dominion University, May 2008.
16. Ganni, V., et al, "Screw Compressor Characteristics for Helium Refrigeration Systems," in *Advances in Cryogenic Engineering* 53, AIP, New York, 2008.
17. Knudsen, P., "Small 2 K System Design," Jefferson Lab Professional Development Program: Cryogenic Seminars, April 5, 2011, http://conferences.jlab.org/cryo/, date accessed, June 1, 2011.
18. Knudsen, P., Ganni, V., "Cold End Process Options for Nominal Efficiency Improvements," JLab Technical Note 11-014, 2011.