Testing of a 4 K to 2 K heat exchanger with an intermediate pressure drop

P Knudsen, V Ganni

Cryogenics Group, Engineering Division, Thomas Jefferson National Accelerator Facility, 12000 Jefferson Ave., Newport News, VA 23606, USA

E-mail: knudsen@jlab.org

Abstract. Most large sub-atmospheric helium refrigeration systems incorporate a heat exchanger at the load, or in the distribution system, to counter-flow the sub-atmospheric return with the super-critical or liquid supply. A significant process improvement is theoretically obtainable by handling the exergy loss across the Joule-Thompson throttling valve supplying the flow to the load in a simple but different manner. As briefly outlined in previous publications, the exergy loss can be minimized by allowing the supply flow pressure to decrease to a sub-atmospheric pressure concurrent with heat exchange flow from the load. One practical implementation is to sub-divide the supply flow pressure drop between two heat exchanger sections, incorporating an intermediate pressure drop. Such a test is being performed at Jefferson Lab’s Cryogenic Test Facility (CTF). This paper will briefly discuss the theory, practical implementation and test results and analysis obtained to date.

1. Introduction

Refrigeration below 4.5 K typically involves sub-atmospheric helium at some point in the process. As most present day particle accelerators are designed to operate at 1.8 to 2.1 K (i.e., 16 to 42 mbar), these systems will be referred to as nominally 2-K or just 2-K. Since there is typically no work extraction below 4.5 K (i.e., refrigeration produced by non-isenthalpic expansion), it is critical to utilize the exergy flux below 4.5 K to the load in an efficient manner, with a minimum of losses [1]. Figure 1 shows the latent heat of helium vs. saturation temperature. Most efficient 2-K superconducting applications accept a goal of 20 J/g as the useful latent heat (to the load). However, as can be seen in Figure 1, this leaves ~17% of the latent heat unutilized. 2-K systems are very energy intensive processes [1-3]. Table 1 summarizes some ‘good’ reasonable system performance expectations. So the question is, can the enthalpy flux to the 2-K load be increased by any process changes between the 4.5 K to 2-K temperature levels?

2. Theory and existing configurations

Typically the supply pressure to the load for large systems is ~3 bar (i.e., super-critical) for thermo-hydraulic reasons. For small systems, saturated liquid at ~1.2 bar may be supplied to the load. Regardless, for a sub-atmospheric 2-K load, the supply pressure is throttled through a (Joule-Thompson, or JT) valve which results in a significant exergy loss. Although the enthalpy flux (between the return and supply streams) is the same both upstream and downstream of the JT valve to
the load, as we will see, this exergy loss can be reduced without extracting work, thereby increasing
the enthalpy flux to the load.

One might ask, what kind of performance benefit does even a typical HX between the 4.5 K
refrigerator supply and the 2-K load provide? Figure 2, arrangement 1 is such a configuration. As
seen in Table 2, the unrecovered refrigeration from the sub-atmospheric load results in an enthalpy
flux (to the load) of 12.7 J/g. Note that Table 2 is meant to be a comparative tool; a specific design
may have somewhat different values. Arrangement 2 in Figure 2 shows the configuration used for the
JLab/CEBAF design [4]. As can be observed, the distribution heat in-leak occurs between the HX and
the load. Per Table 2, this arrangement yields a load enthalpy flux of 18.7 J/g; an increase of ~50%.
In Table 2 the HX cold-end stream temperature difference of 0.20 K was chosen for practical reasons;
keeping the supply stream leaving the HX above lambda and providing a realistic HX design (i.e.,
reasonable NTU’s). Arrangement 3 in Figure 2 shows the configuration used at SNS and other more
modern installations. This arrangement uses a HX at the load, so that the heat in-leak is absorbed
between the refrigerator and the HX at a higher temperature level, rather than with a single large HX
adjacent to the refrigerator (which is at lower temperature level; as in the CEBAF design). Thus, there
are many more HX’s, but of much smaller size. Per Table 2, case 3(i), arrangement 3 with a supply
pressure of 3 bar yields a load enthalpy flux of 20.0 J/g; an increase from arrangement 2 of 7%. Of
course, this result may be intuitive, since from the second law of thermodynamics, the incurred losses
are lower in absorbing a fixed heat in-leak at a higher temperature. For a small system which provides

![Figure 1. Helium latent heat vs. saturation temperature](image)

**Table 1.** ‘Good’ reasonable system performance expectations

| #   | Type                                         | Approx. 2 K Inverse COP |
|-----|----------------------------------------------|-------------------------|
| 1   | Warm vacuum pumping using a 4.5 – 2K HX     | 4600                    |
| 2   | Warm vacuum pumping using 300 – 4.5 K and 4.5 – 2K HX’s | 1400                    |
| 3   | Partial cold compression (small system)     | 1200                    |
| 4   | Partial cold compression (large system)     | 750 - 950               |
| 5   | Full cold compression                       | 750 - 850               |
a supply pressure of ~1.3 bar, per Table 2 case 3(ii), the load enthalpy flux is 21.1 J/g; an increase from case 3(i) of 6%, or an increase from arrangement 2 of 13%.

Before passing by this result, we might ask if an imposed and significant supply pressure drop through the HX is more constructive than wasting it across the load JT valve. To investigate this question we can examine the effect on the enthalpy flux assuming a continuous supply pressure drop through the HX from 3 bar at the inlet to a given (lower) outlet pressure. The plot in Figure 3 is the enthalpy flux (to the load), $\Delta h_{\text{LE}}$, vs. the outlet supply pressure on the cold-end of the HX, $p_{\text{LE}}$, for an inlet supply pressure of 3 bar. This Figure shows that there is a very significant performance advantage in utilizing the pressure drop in the HX, rather than wasting across the JT valve at the load. This can also improve the heat transfer coefficient. Figure 3 also shows the sub-atmospheric stream warm-end outlet temperature, $T_{\text{WE}}$ vs. $p_{\text{LE}}$. Consideration of this temperature is important in the sizing of a cold compressor, if used. For this simple analysis, a cold-end stream temperature difference of 0.20 K is used, and zero heat in-leak and zero sub-atmospheric pressure drop has been assumed.

**Figure 2.** Cold end arrangements 1 to 3

**Table 2.** Process parameters for cold end arrangements 1 to 4

| Symbol | Description | Units | 1 | 2 | 3(i) | 3(ii) | 4(a) | 4(b) |
|--------|-------------|-------|---|---|------|-------|------|------|
| $m$    | Mass flow rate (§) | [g/s] | 10 | 10 | 10 | 10 | 10 | 10 |
| $p_{h,1}$ | Supply pressure from refrigerator | [bar] | 3.0 | 3.0 | 3.0 | 1.3 | 3.0 | 3.0 |
| $T_{h,1}$ | Supply temperature from refrigerator | [K] | 4.5 | 4.5 | 4.5 | 4.5 | 4.5 | 4.5 |
| $p_{h,4}$ | Intermediate (h) stream pressure | [bar] | ~3.0 | ~3.0 | ~3.0 | ~3.0 | ~3.0 | ~3.0 |
| $T_{l,1}$ | Return temperature to refrigerator | [K] | 2.3 | 3.4 | 3.7 | 3.9 | 4.0 | 4.0 |
| $p_{l,5}$ | Load return pressure | [bar] | 0.030 | 0.030 | 0.030 | 0.030 | 0.030 | 0.030 |
| $T_{l,5}$ | Load return temperature (‡) | [K] | 2.0 | 2.0 | 2.0 | 2.0 | 2.0 | 2.0 |
| $\Delta T_{\text{h,LE}}$ | Stream temperature diff. on HX cold end | [K] | N/A | 0.20 | 0.20 | 0.20 | 0.20 | 0.20 |
| $\Delta h_{\text{h,LE}}$ | Load enthalpy flux | [J/g] | 12.7 | 18.7 | 20.0 | 21.1 | 21.9 | 21.9 |
| $q_{TL,h}$ | (h) stream transfer line heat in-leaf | [W] | 5 | 5 | 5 | 5 | 5 | 5 |
| $q_{TL,l}$ | (l) stream transfer line heat in-leaf | [W] | 20 | 20 | 20 | 20 | 20 | 20 |
| $q_L$ | Load | [W] | 127 | 187 | 200 | 211 | 219 | 219 |
| $NTU$ | HX no. transfer units | [-] | N/A | 3.3 | 3.6 | 3.8 | 4.0 | 5.2 |

**Notes:**
- (‡) Load temperature is saturation temperature at load pressure
- (§) (h) stream mass flow is equal to (l) stream mass flow
Next, let’s consider using such a HX, which has a significant pressure drop in the supply stream, in a manner as shown in Figure 4 arrangement 4(a). We will fix the outlet supply pressure at 0.2 bar, for reasons that will be discussed shortly. The performance of this arrangement is shown in Table 2, yielding a load enthalpy flux of 21.9 J/g; an increase from arrangement 3(a) of 9%. However, having such a large continuous pressure drop is not a practical design. Even if provisions are made for the cool-down (or warm-up) of the HX and a design-manufacturing process is conceived to achieve close tolerance on the pressure drop, actual and varying operating conditions (process and/or equipment) may render the HX unusable. Arrangement 4(b) in Figure 4 shows a practical implementation of achieving the same performance result [5]. An intermediate JT valve is placed between the upper (warmer) and lower (colder) HX. The outlet pressure from the lower HX can be regulated by a passive device (V2), such as a gravity controlled differential pressure check valve. As can be seen in Table 2, arrangement 4(a) and 4(b) have identical performance, but 4(b) requires a modestly longer heat exchanger (i.e., more NTU’s).

So, why was an outlet supply pressure of 0.2 bar chosen? Why not take the entire pressure drop from 3 bar to the load pressure, even if arrangement 4(a) could somehow be made practical? First, the cold-end HX stream temperature difference of 0.2 K, which represents a practical design, imposes a fluid property limit of the saturation pressure at the cold-end temperature; i.e., 52.1 mbar at (2.0 + 0.2 =) 2.2 K. So, why not 0.0521 bar instead of 0.2 bar? As shown in Figure 5, this would require an impractically long HX. In fact, below ~0.2 bar, the HX NTU requirement begins to grow very quickly.

Figure 6 is a pressure-enthalpy diagram for helium, which is useful to visualize the fluid behavior which allows effective use of the pressure drop. Point “A” to “B” is the supply stream path for arrangements 2 and 3. Point “A” to “C” is the supply stream path for arrangement 4. It is evident that two-phase conditions are likely just downstream of the intermediate JT for arrangement 4(b).

Figure 7 shows two additional arrangements. Arrangement 4(c) is an alternate equivalent arrangement of 4(b), where the lower HX is placed in the load vapor space. This could offer an option where there are special constraints preventing the lower HX from being placed just outside the load. Arrangement 5 is not equivalent to arrangement 4 since a portion of the heat load is going into the liquid; i.e., the lower HX duty is now part of the load! The performance of this arrangement can be equivalent to arrangement 3.

![Figure 3. Effect of supply stream pressure drop through HX](image-url)
3. New concept validation
In order to validate a practical implementation of the theory [5], a HX test can as in Arrangement 4(b) was designed, fabricated and tested. JLab developed the design, the fabrication was done by MSU-FRIB and the HX test can was tested at JLab’s Cryogenic Test Facility (CTF). The HX test can was designed for up to ~6 g/s. Figure 8 shows the HX test can being fabricated at MSU-FRIB (with all the internal components tack welded in place).

The HX test can is comprised of two 178 mm diameter by 697 mm long Collins type HX’s (upper and lower), a 13 liter liquid vessel with a 200 W externally mounted heater (bands), two JT valves and a copper radiation shield. The HX supply stream is a 12.7 mm diameter copper tube with 8 mm height (external) copper fins and an internal twisted copper tape turbulator that is helically wound onto a 114 mm diameter mandrel with 16 wraps, fin-to-fin, using a solid braid Nylon rope to seal between the shell and mandrel gaps. The fin densities of 354 and 472 fins per meter are used for the upper and lower HXs.

![Diagram of Arrangements 4(a) and 4(b)](image)

Figure 4. Cold end arrangements 4(a) and 4(b)

![Graph showing total HX NTU vs. HX CE outlet pressure](image)

Figure 5. Supply pressure at HX cold end vs. HX NTU’s [5]
Process Points: “A” = 3 bar, 4.5 K; “B” = 3 bar, 2.2 K; “C” = 0.2 bar, 2.2 K; “D” = 0.03 bar, 2.0 K

Figure 6. Helium pressure-enthalpy diagram of HX process

Figure 7. Cold end arrangements 4(c) and 5
lower HX, respectively. The inside of the mandrel is a vacuum, common to the HX test can volume. Both JT valves are on the supply stream. One JT valve is in between the HX’s (the intermediate valve) and the other is between the lower HX and the test load vessel. A passive device has not been implemented yet in order to provide more flexibility in testing. The copper radiation shield was anchored to the sub-atmospheric flow leaving the HX test can. In addition, super-insulation is used between the vacuum shell and copper shield, the inside of the copper shield and on the internal components. Lakeshore DT-670 diodes, mounted instream with both a primary and a back-up, were located at the inlet and outlet of the HX’s for both supply and sub-atmospheric streams. Diodes are mounted into a copper bulb which is immersed into the process flow. This design prevents any possibility of the diodes being carried away by the process flow and does not require the wires to cross the process fluid boundary (i.e., an instrument wire feed-through is not required). However, diodes must be properly anchored to bulb (to ensure good thermal contact) and wires must be properly thermally anchored (i.e., ‘heat stationed’). The diode signals were sent to a CryoCon 18. An American Magnetics super-conducting liquid level probe was used to measure the liquid level in the vessel. Five pressure transducers were used. A GE-UNIK PTX-5072 on the supply stream at the inlet to the upper HX, another on the supply stream downstream of the intermediate JT and the third on the liquid vessel (vapor space). A MKS DMB13T (low range) and GE-UNIK PTX-5072 (high range) was used on the supply stream downstream of the intermediate JT. A MKS DMB12T (low range) and GE-UNIK PTX-5072 (high range) on the liquid vessel (vapor space). A Rosemount 1151DP3S differential pressure transducer was installed to measure the sub-atmospheric stream pressure drop across both HX’s. A Sorensen-Ametek DLM-150-4 provided power to the vessel heaters.

Although there is an existing venturi flow meter (which is part of the JLab CTF distribution) on the supply upstream of the HX test can, two other devices were implemented. The accuracy of this existing device was believed to be not accurate enough for the planned testing, since the maximum (fractional) change in mass flow is expected to be ~10%. One device was installed on the super-critical supply line between the CTF junction box and HX test can. The second device was installed on the sub-atmospheric return line between the HX test can and CTF junction box. A Micro Motion CMF025M Coriolis mass flow meter was installed on the supply line, modified for cryogenic temperatures. This flow meter is a commercial product and did not require any special modifications needed for earlier development versions [6] and is believed to have an accuracy of better than 0.5% over the intended test flow rate range. The device on the sub-atmospheric return line was a custom design (by JLab) thermal mass flowmeter. A 250 W cartridge heater was inserted into a 19 mm diameter copper tube with 19 mm high, 0.71 mm diameter copper wire (loop) fins; with a fin density of 60 loops per turn and 244 turns per meter. Downstream of the heater, a Lakeshore DT-670 diode was mounted, immersed in the process stream. In this case, the diode (a primary and back-up) was exposed directly to the process flow and encased in a perforated tube to prevent the possibility of being carried away by the flow. The diode signals were sent to a CryoCon 18. A Sorensen-Ametek DLM-150-4 provided power to the cartridge heater. The design allowed for the heater and diode to be replaced without cutting and (re-)welding.

Testing was conducted at the CTF using the two addition mass flow meters in December 2014 and January 2015. The key measurements required are the load (heat input), vessel (sub-atmospheric) pressure and the mass flow rate. The supply pressure and temperature, intermediate pressure and a reasonable estimate of the static heat in-leak (to the liquid vessel) are also required, but do not need the same accuracy as the key measurements. All measurements were taken at steady conditions with no mass accumulation. Figure 9 shows the preliminary testing results of the calculated load enthalpy flux, given the heat input and the mass flow rate, vs. the intermediate pressure (i.e., the pressure just downstream of the intermediate JT valve). Static heat in-leak to the vessel was measured using a boil-off test and is ~1.5 W. The supply pressure was typically 2.6 to 2.7 bar and the supply temperature 5.1 to 5.2 K. As anticipated in [5], we note that the performance improves on decreasing intermediate pressure down to 0.2 bar, below which the performance degrades. This is attributed to excess demand on the upper HX and two phase in the lower HX (supply stream) at a temperature that is much closer
to the sub-atmospheric vapor return than at higher intermediate pressures (refer to Figure 6). Measurements in Figure 9 were taken for the same Coriolis flowmeter zero calibration. This point may not have been precisely 0 g/s due to test constraints but is thought to be very close. A Coriolis flow meter offset of ~0.03 g/s was assumed to keep the outlet temperature above the lambda temperature.

Figure 8. HX test can under construction at MSU-FRIB

Figure 9. Preliminary HX testing results – load enthalpy flux vs. intermediate pressure
4. Conclusion
Preliminary testing has validated a practical implementation of constructively utilizing the supply pressure drop to substantially improve the performance of the cold end of a sub-atmospheric helium refrigeration system up to the theoretically predicted value of ~9%. Additional testing implementing a passive back-pressure (between the intermediate JT and the outlet of the lower HX) device is planned.

Acknowledgements
The authors would like to express their appreciation and thanks to the TJNAF management and to their colleagues at JLab and MSU for their support. This work was supported by the U.S. Department of Energy under contract no. DE-AC05-06OR23177.

References
[1] Ganni V, Knudsen P 2014 Helium refrigeration considerations for cryomodule design Adv. Cryo. Eng. 59 (New York: American Institute of Physics) pp 1814-1821
[2] Knudsen P, Ganni V 2012 Process options for nominal 2-K helium refrigeration system design Adv. Cryo. Eng. 57 (New York: American Institute of Physics) pp 800-813
[3] Claudet S, Ferlin G, Miller F, Tavian L 2004 1.8 K refrigeration units for the LHC: performance assessment of pre-series units Proc. 20th International Cryogenic Engineering Conference (Beijing, China, 11-14 May 2004)
[4] Chronis W, Arenius D, Devins B, Ganni V, Kashy D, Keesee M, Reid T, Wilson J 1996 Procurement and commissioning of the CHL refrigerator at CEBAF Adv. Cryo. Eng. 41 (New York: Plenum Press) pp 641-648
[5] Knudsen P, Ganni V 2011 Cold end process options for nominal efficiency improvements JLab Tech. Note 11-014
[6] Serio L A cryogenic helium mass flowmeter for the Large Hadron Collider Ph.D. Thesis (Cranfield University) 2007