Testing of a 4 K to 2 K Heat Exchanger with an Intermediate Pressure Drop

Pete Knudsen, Rao Ganni

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

Office of Science U.S. Department of Energy

Jefferson Lab Thomas Jefferson National Accelerator Facility

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. **Most efficient 2-K super-conducting applications accept a goal of 20 J/g as the useful latent heat, or enthalpy flux, to the load.** However, this leaves ~17% of the latent heat unutilized. 2-K systems are very energy intensive processes. The table to the right 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?**

#	Type	Approx. 2 K Inverse COP
1	Warm vacuum pumping using a 4.5 to 2K HX	4600
2	Warm vacuum pumping using 300 to 4.5 K and 4.5 to 2K HX's	1400
3	Partial cold compression (small system)	1200
3	Partial cold compression (large system)	750 - 950
4	Full cold compression	750 - 850

Theory and existing configurations

Arrangements 1 to 3 show typical cold-end configurations. *(h)* is the positive pressure supply stream (~3 bar 4.5 K from the refrigerator). *(l)* is the sub-atmospheric load return stream going back to the 2-K refrigerator and/or warm vacuum pumps. 'JT' is the Joule-Thompson throttling valve supplying the load. '$q_{TL,h}$', '$q_{TL,l}$' and 'q_L' are the supply transfer line heat in-leak, return transfer line heat in-leak and the (heat) load, respectively. 'HX', 'HX-A', 'HX-B' are heat exchangers.

Arrangement 1 is used in small systems. Arrangement 2, similar to CEBAF, uses a large 4.5 – 2 K HX, absorbing the transfer line heat in-leak at the load supply temperature (~2 K). Arrangement 3 is typically used on modern large systems (e.g., SNS, FRIB) and uses smaller 4.5 – 2K HX's at the load, absorbing the transfer line heat in-leak at the refrigerator return temperature (~4 K). Performance for these is summarized in the table below. Case 3(i) is typical of large systems, while case 3(ii) is typical of small systems.

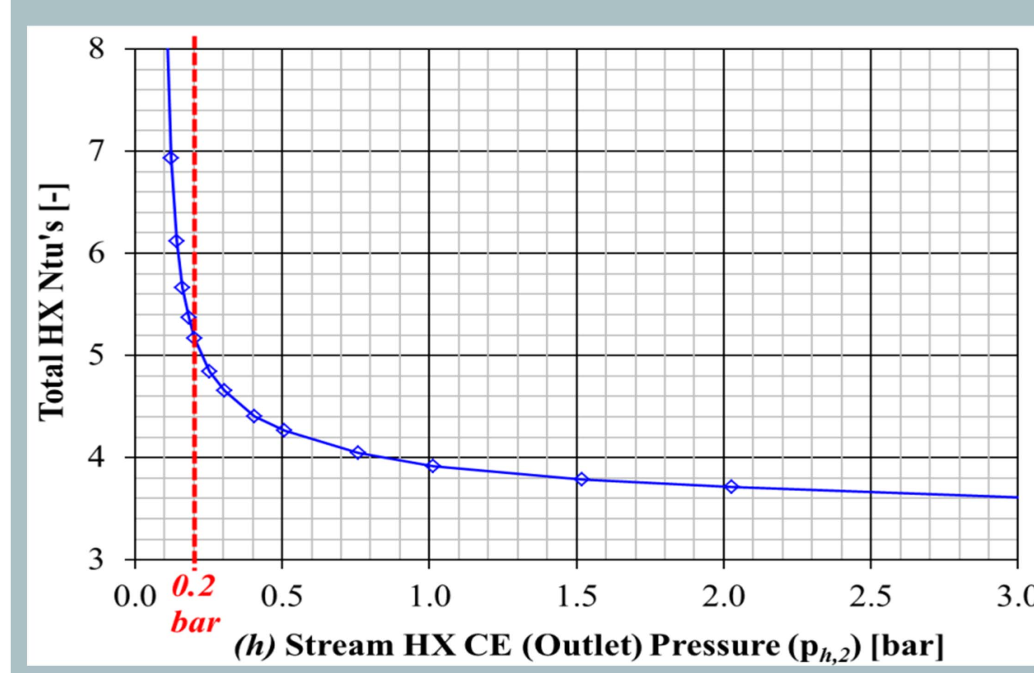
Arrangement 4 appears the same as 3. However, there is a large continuous pressure drop in the supply *(h)* stream through the HX. The figure below shows how the enthalpy flux (i.e., the difference in enthalpy between *(h)* and *(l)* streams at the load) varies with the *(h)* stream pressure exiting the HX. As can be seen there is a significant improvement in performance by using the pressure drop ***in*** the HX. The pressure-enthalpy chart to the right provides an explanation. *So, although there is no work extraction (between 4.5 to 2 K), additional cooling can be realized by expending the (h) stream availability (in this case, pressure) **with** heat exchange, rather than solely across the JT valve supplying the load.*

Arrangement 4(b) shows a practical implementation of 4(a), using an upper (HX-A) and lower (HX-B) heat exchanger with an intermediate JT valve. The valve 'V2' can be a passive component, such as a gravity controlled differential pressure check valve. *Note that the performance of 4(a) and 4(b) are identical*. The difference is that 4(b) requires a modestly longer HX (30%). Arrangement 4(c) show an alternate practical implementation that can require less space.

Arrangement 5 is identical to arrangement 3; ***not*** arrangement 4. In this, the duty of lower HX-B becomes part of the load.

Selection of intermediate pressure

The (h) stream out of the cold-end of the HX cannot go below the temperature at the corresponding saturation pressure; which for 2.2 K is 52.1 mbar. Additionally, as shown in the figure below the HX length (NTU's) rapidly increases below ~0.2 bar.

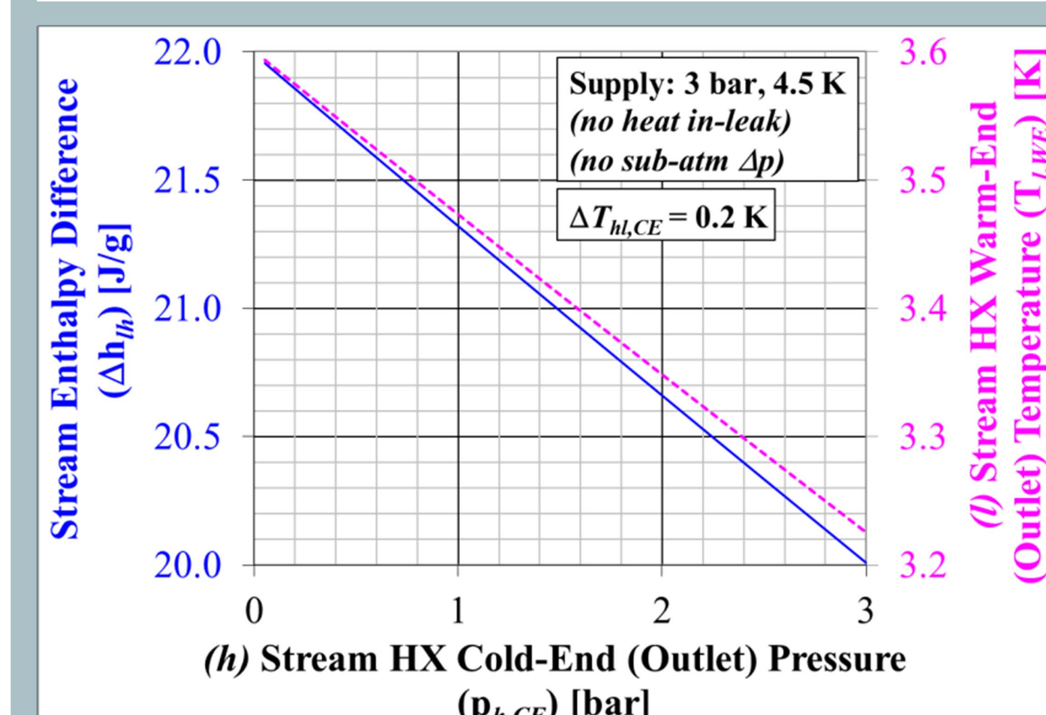


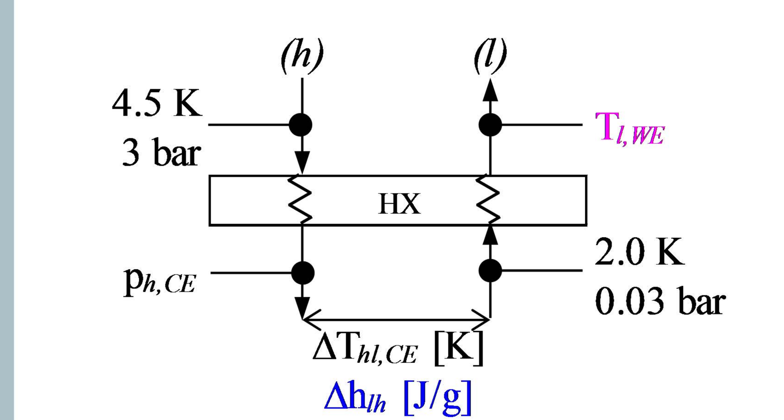
Arrangement performance

Symbol	Description	Units	1	2	3(i)	3(ii)	4(a)	4(b)
$\dot{m}$	Mass flow rate (§)	[g/s]	10	10	10	10	10	10
$p_{h,1}$	Supply pressure from refrigerator	[bar]	3	3	3	1.3	3	3
$T_{h,1}$	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	~1.3	0.2	0.2
$T_{l,1}$	Return temperature to refrigerator	[K]	2.3	3.4	3.7	3.9	4	4
$p_{l,5}$	Load return pressure	[bar]	0.03	0.03	0.03	0.03	0.03	0.03
$T_{l,5}$	Load return temperature (‡)	[K]	2	2	2	2	2	2
$\Delta T_{hl,3}$	Stream temperature diff. on HX cold end	[K]	N/A	0.2	0.2	0.2	0.2	0.2
$\Delta h_{lh,5}$	Load enthalpy flux	[J/g]	12.7	18.7	20	21.1	21.9	21.9
$q_{TL,h}$	*(h)* stream transfer line heat in-leak	[W]	5	5	5	5	5	5
$q_{TL,l}$	*(l)* stream transfer line heat in-leak	[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	5.2

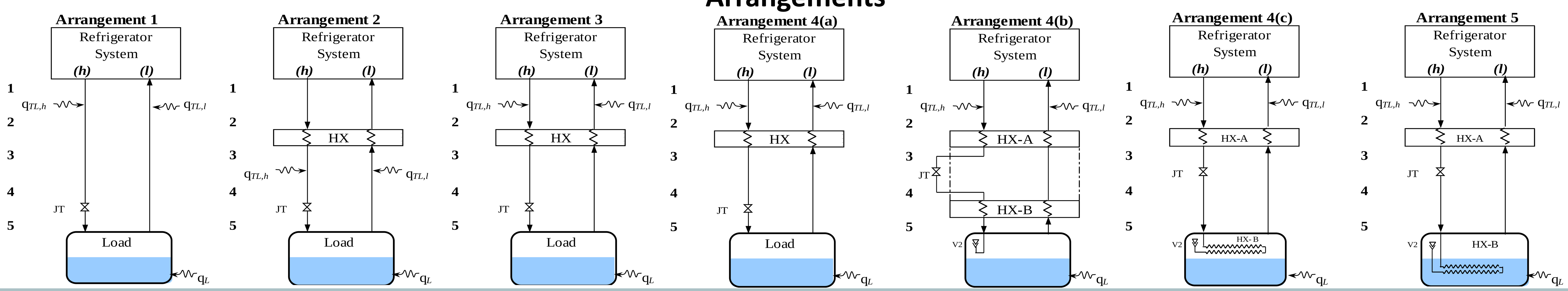
Notes: (‡) Load temperature is saturation temperature at load pressure
(§) *(h)* stream mass flow is equal to *(l)* stream mass flow

Heat exchange with pressure drop





Arrangements

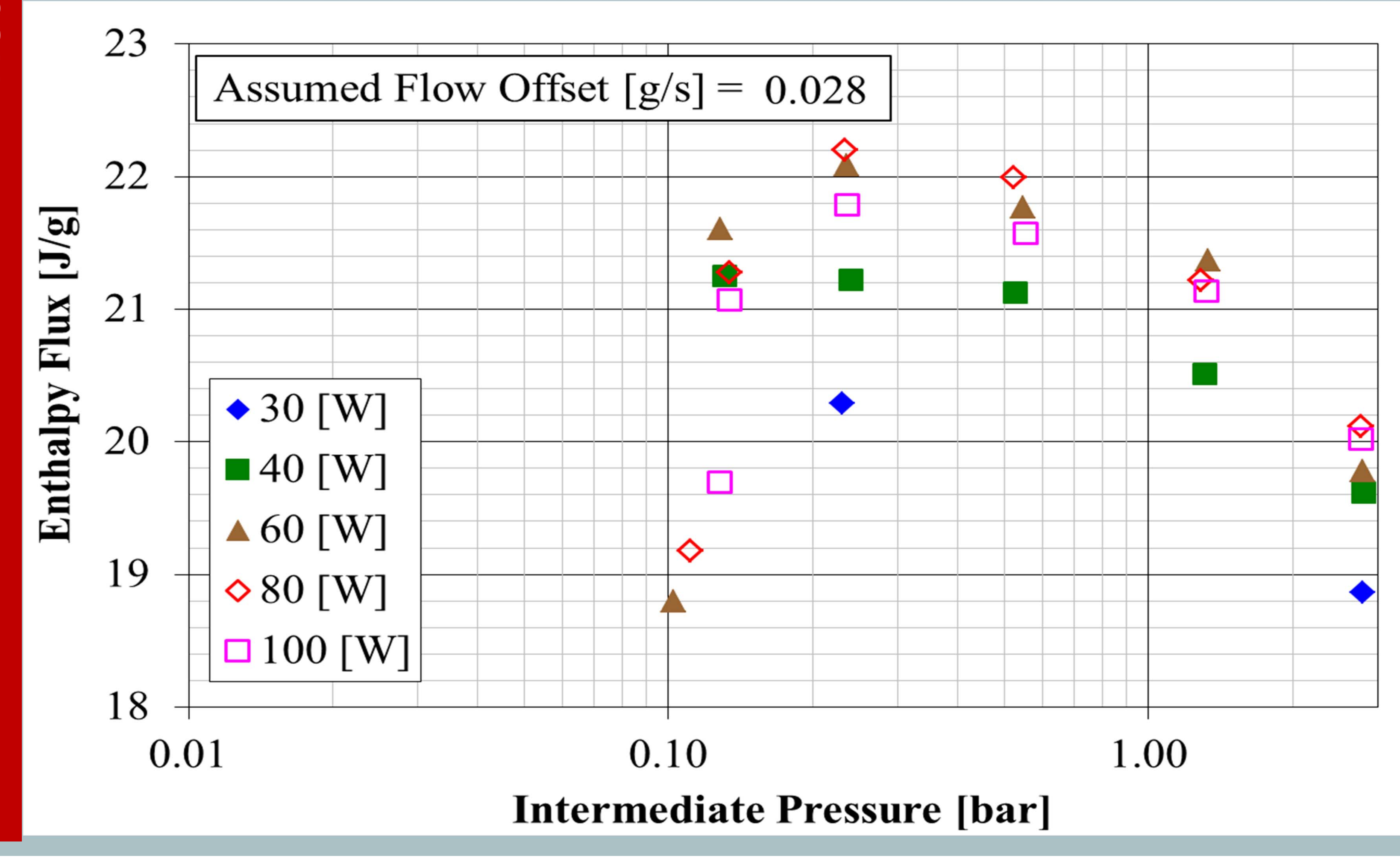


Results

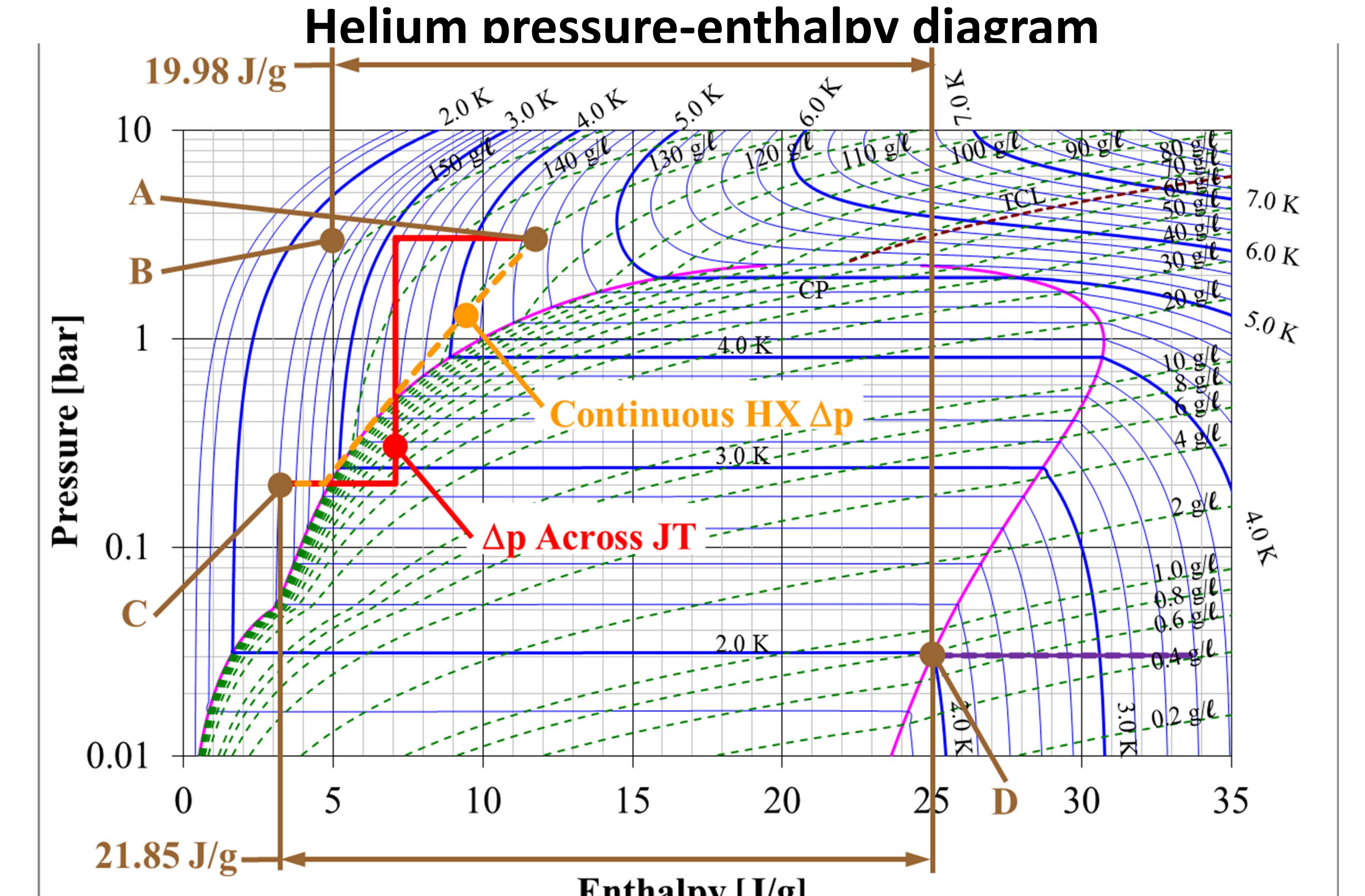
In order to validate a practical implementation of the theory, a HX test can was designed, fabricated and tested. JLab developed the design, the fabrication was done by MSU-FRIB and the HX test can was tested at JLab's Cryogenic Test Facility (CTF). The HX test can was designed for up to ~6 g/s.

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 Collins type 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. A standard cryogenic Coriolis meter (CMF025) located just upstream of the HX test can was used to measure the mass flow rate.

The figure below shows the results from testing in Dec. 2014 and Jan. 2015. This testing validated the practical implementation of arrangement 4(b) and demonstrated the theoretically predicted peak performance at ~0.2 bar, in addition to a capacity improvement of up to the theoretically predicted value of ~9%. Additional testing implementing a passive back-pressure device is planned.



Helium pressure-enthalpy diagram



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

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