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LARGE HELIUM REFRIGERATORS AND LIQUEFIERS

Author
Byrns, R.A.

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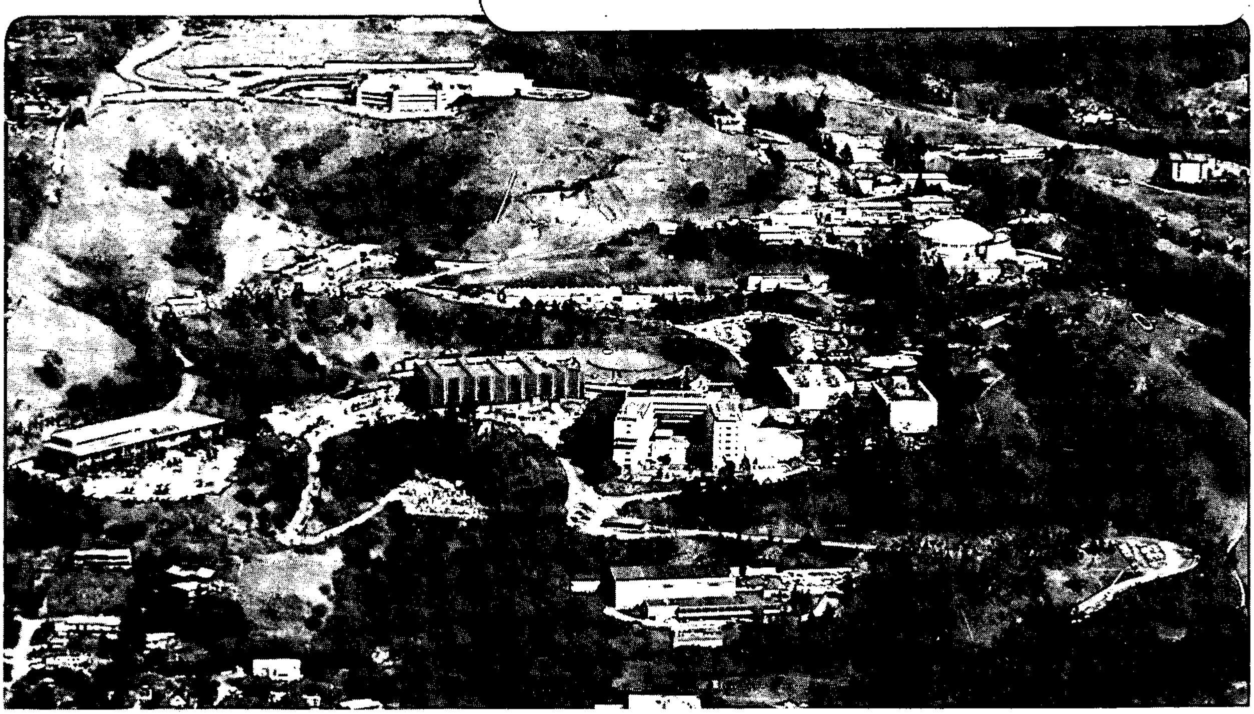

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LARGE HELIUM REFRIGERATORS AND LIQUEFIERS

R. A. Byrns

Lawrence Berkeley Laboratory
University of California
Berkeley, CA 94720

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Preface

This paper started when Brian Hands of Oxford University asked me to write a chapter on large helium refrigerators for his book on cryogenics. He found me in a weak moment at the 1981 Cryo Conference in San Diego and I said yes. Later it turned out he considered "large" > 10 L/hr LHe and I set "large" at > 1500 W.

This first draft edition is a starting point for a description of large equipment (probably only addressed to a handful of cryo-refrigeration fans throughout the world). Comments and corrections are welcome.

Don Brown, BNL, has made many helpful suggestions. I agree the BNL-CTi 4000 refrigerator, 1450 W, should probably be included, as a good example, with its four reciprocating expanders (2 redundant). Barrera, Petersen, and Watt of SLAC have made salty comments about the Second Law and operations. The section on entropy and optimization is doubtless fuzzy (but could require a whole book, written by someone who understands it).

Future editions should include the refrigeration for "Tore Supra" in France. This unique superconducting Tokamak will have 18 magnet coils at 1.8 K superfluid. The refrigerator will yield 300 W at 1.8 K and 750 W at 4.5 K. Operations are scheduled for early 1985. Both the HERA project at DESY, Hamburg, a 1000 magnet accelerator (similar to Fermilab) with four 8000 W machines (32 kW total), and the US Superconducting Super Collider (SSC) to be built within the next decade might also be described.

Rod Byrns

January 1984

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I a. Introduction[1-4]

Some of the most powerful methods of heat transfer involve change-of-state or phase. For cooling, evaporative or boiling heat transfer is most effective and ancient, e.g. perspiration. Another common cooling media involves the melting of ice. Various solutes, salt and ethylene glycol have been added to water to lower or raise its phase change points.

Now the more common commercial refrigerants are ammonia (NH_3) and the fluorinated hydrocarbons (Freon). Sulfur dioxide (SO_2) has become obsolete (toxic). These are all fluids that can be compressed, cooled, and expanded to make liquid and boiled or evaporated to provide cooling in a closed-loop continuous process.

The history of mechanical refrigeration begins in the 1800's and its motivaton was economic, initially for food preservation. Later oxygen production and air distillation products; Ar, Ne, Kr, Xe became important.

There are two primary methods for the large scale liquefaction of air, hydrogen and helium. The first uses the Joule-Thomson (J-T) effect which creates cooling by expansion of a gas through a valve. "Internal" work is performed and the process is considered isenthalpic and non-reversible, and thus, efficiency is not optimal. The higher density, more complex molecules (air, O_2, N_2, NH_3, SO_2, Freon) yield the most cooling at ambient or higher pre-cooling temperatures. The low density, more ideal molecules, H_2 and He, do not cool when expanded through a J-T valve (but rather heat!) unless the inlet temperature is very low; 65-80 K for H_2 and 15-20 K for He. Linde, in

Munich, Germany, used the high pressure J-T cycle to liquefy air in 1895, and later developed a commercial air separation plant in 1902.

The other major cooling method uses the reversible expansion of the gas to perform "external" work. For this process the gas molecular structure is less important. Cooling occurs from any inlet temperature, and because of its reversibility this isentropic process can achieve very high efficiencies. Claude, in France, first used a piston expander in 1902 to liquefy large amounts of air.

Dewar, in Scotland developed the vacuum insulated vessel in 1892 and liquefied hydrogen in 1898 using liquid air as pre-coolant in a cascade J-T cycle. Helium was first liquefied by Kamerlingh Onnes at Leiden, Holland in 1908 using LH_2 as pre-coolant in a J-T cycle. Until 1923, Onnes Laboratory was the only place in the world where measurements could be made in the temperature range of 1.5 - 4.2K. By Onnes death in 1926, only two labs other than Leiden (Toronto and Berlin) were able to liquefy helium. At the start of World War II there were about 12 laboratories with LHe facilities. By 1960, as a result of the Collins liquefier (1947) using two expander pistons in a Claude cycle, there were about 150 LHe installations.

In 1952, the National Bureau of Standards built the Cryogenic Engineering Lab at Boulder, Colorado equipped with a 350 L/hr LH_2 liquefier. A similar plant was built in Eniewetok in the Pacific for the Atomic Energy Commission. Both plants used the high pressure Linde J-T cycle. By 1956, similar LH_2 systems were installed at Berkeley and later Brookhaven for hydrogen bubble chambers. Sulzer in Switzerland built a large H_2

plant in Ems for D_2 distillation and other large H_2 plants at CERN, Geneva for bubble chambers. The Sulzer plants used turboexpanders in a Claude cycle.

Sputnik and the U.S. space program under NASA circa 1960 gave impetus to the installation of large LH_2 plants in Ohio, Florida and California. Capacity varied from 3000 L/hr (5.5 ton/day) up to 60 ton/day. The smaller plants used J-T expansion and LN pre-cooling; the larger, H_2 turboexpanders. LHe plants up to 1500 W, were installed for NASA for cryopumping large space chambers.

The decade from 1970 through 1980 was typified by the increase of large (100-1500 W or 30-350 L/hr) LHe plants, installed throughout the U.S., Europe and Asia. In the mid-west helium wells, 600-800 L/hr plants were common. In mid 1970, a 1000 L/hr LHe liquefier for Poland was built in England. Recent additions in Kansas include plants of 1200-1400 L/hr LHe.

Now in the 80's we have the worlds largest He liquefier (4000 L/hr) installed at Fermilab, Batavia, Illinois together with its 24 satellite refrigerators of 1000 W each. At Brookhaven National Laboratory, Long Island, New York assembly procedes on another large He refrigerator of 25 kW capacity at 3.8 K plus 55 kW at 50 K.

A chronological time line is shown in Table I showing some salient events occurring in the history of low temperature liquefaction-refrigeration.

TABLE I[1-4]

1816	Stirling Hot Air Engine patented.
1823	Faraday liquefied chlorine (-110 C). Postulated all gases condensible if temperature low enough.
1825-73	Many developments in engines and air refrigerators.
1873	Brayton cycle invented in Massachusetts.
1877	Cailletet produced a liquid air fog, Pictet a jet of liquid oxygen.
1884	Wroblewski and Olszewski liquefied O_2 and N_2 at Cracow University.
1886	Solvay described a cryogenic refrigerator with an expander incorporating a regenerator.
1892	Dewar developed a vacuum-insulated vessel for cryogens.
1895	Linde granted basic patent on air liquefaction.
1898	Dewar liquefied hydrogen (20 K) in bulk at Royal Institute, London.
1899-02	Claude invents engine and cycle for liquid air.
1902	First commercial plant for separating oxygen by Linde AG in Germany. l'Air Liquide Company formed in France to use Claude process.
1905	Other Linde plants established in Germany and France.
1906	British Linde founded, later then became British Oxygen Co. (BOC)
1907	Linde Air Products in U. S. established. Claude separated neon as air plant by-product.
1908	Kamerlingh Onnes liquefied helium at Leiden, reached 0.7 K.
1911	Kamerlingh Onnes discovered superconductivity.
1920	Claude constructs a hydrogen expander engine.
1926	Giauque and Debye proposed adiabatic demagnetization.
1932	Linde AG builds first industrial He liquefier.
1933	Giauque and MacDougall reached 0.25 K, adiabatic demag.
1934	Kapitza liquefied helium using one engine.
1946	Collins introduces helium liquefier with two expander engines; by 1964, 260 units installed, 400 by 1969.
1952	NBS-CEL Boulder, 350 L/hr H_2 liquefier installed.
1955-60	Four large LH_2 plants in U. S. for NASA, 3000 L/hr to 60 ton/day. Large H_2 plants in France and Switzerland for deuterium separation. Large H_2 plants up to 7 kW at 20 K in New York and Geneva for Bubble Chambers.
1962	Linde Div., Union Carbide Corp. installs three of worlds largest He liquefiers, 100 L/hr, two at NASA, Cleveland, Ohio; one at Amarillo, Texas. Amarillo plant increased to 250 L/hr with H_2 pre-cooling in 1965.
1963	Air Products and Chem., Inc. installs 1300W He refrigerator, 140 L/hr; Langley, VA, CVi, Columbus, Ohio builds many 20 K refrigerators for space chamber cryopumps.
1964	Div 500, A. D. Little, Inc. installs two large He liquefiers, 120 L/hr, Navajo, Arizona and Richmond, California.
1965	Sulzer Bros installs 800 L/hr LHe plant at Otis, Kansas.
1968	CTi (Cryogenic Technology, Inc.) introduces new third generation Collins liquefier Model 1400 (25 L/hr), 260 sold by 1983.
1965-75	Large H_2 bubble chambers installed at Brookhaven Nat. Lab. (BNL), Argonne (ANL), Fermilab and CERN, all requiring 20 K refrigerators and all with superconducting magnets and 4.5 K refrigerators. 300 W 1.8 K He II refrigerators installed at Stanford (CTi, 1967), Karlsruhe KFA (Linde AG 1970, Messer-Griesheim 1972) and CERN (BOC 1976). Large LHe liquefier built for Poland 1000 L/hr (BOC).
1975-82	Many LHe systems installed world-wide, with steadily increasing capacity, culminating in the worlds largest liquefier at Fermilab (4000 L/hr), and a 23 kW system at 4.5 K.

I b. Definition of the Heat Load and Capacity

The first and most important step in design is the selection of plant capacity. For a laboratory or commercial liquefier this may be reasonably straight-forward, based on present and projected use, economics or marketing. The available funding also has strong influence, but usually a number is selected, e.g. 100 L/hr or 1500 L/hr. The plant is then designed for that capacity, minimizing capital and/or operating costs.

A system designed as a refrigerator is not optimal as a liquefier, because of the liquefiers unbalanced flow. In the liquefier, mass flow is extracted at 4.5 K, whereas the refrigerator returns all boil-off flow through the heat exchangers (HX) to 300 K, extracting the available refrigeration in the gas.

Many of the modern high-technology LHe cryogenic systems operate in a mixed mode of liquefaction-refrigeration. This includes large superconducting (SC) systems, cryopumps and liquefiers for accelerators, fusion, MHD, energy storage and electrical apparatus. Extreme care must be applied in the analysis of the projected equipment heat loads and cycles, because this sets the refrigerator capacity. Usually at the start of a research project, the heat load numbers are uncertain because the design of the apparatus is incomplete, and many items such as magnet correction coils, power leads and transfer line losses are overlooked. It is not uncommon for the novice to size the plant capacity very close to initial design load, thinking refrigerator capacity is linear with price. Some years later heroic steps may be needed to cool and fill the system.

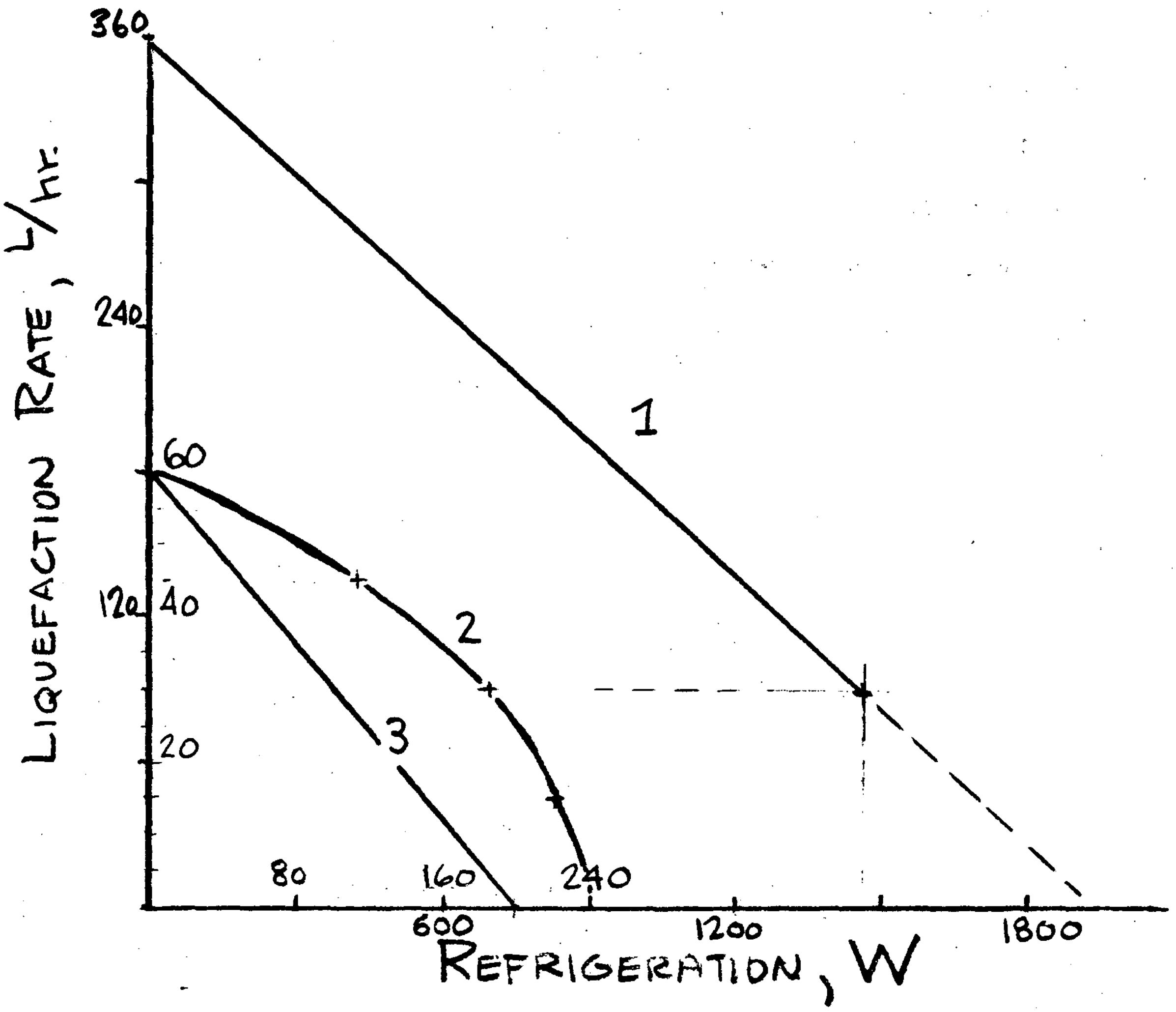


Figure 1.
1. LBL 1500 W produces 1450 W plus 90 L/hr or 360 L/hr.
2. CTi Model 2000 produces 240 W at 4.5 K or 60 L/hr. Measured curve shown (Ref. 5) with 30 g/s at 15 atm, 106 HP, 30 L/hr LN input. (1967) (two recips).
3. CTi Model 2800 produces 200 W or 60 L/hr with 36 g/s. (Two turbines) produces 260 W or 90 L/Hr with 54 g/s.
4. CVI-LLNL 3 KW produces 3300 W or 600 L/hr at 4.35 K.

The relationship between liquefier and refrigerator capacities is shown in Figure 1.[5] The ratio between these numbers usually varies between three and five depending on the specific design. The first approximation is usually linear, although it too will vary for different machines.

Major items to consider in sizing the refrigerator include:

1. Static heat load; consists of all radiation, convection and conduction to the apparatus.
2. Dynamic or pulse heat load; eddy current loss in magnets or electrical gear, efficiency loss in expansion-recompression of bubble chamber liquids or pumping losses.
3. Liquefier loads; provide mass flow from 4.5 K to 300 K to cool magnet current leads or to reduce static conduction leaks.
4. Distribution system; heat loads can exceed primary loads by three, four or more with 100 m or more of transfer line. Include valve boxes; bayonets (preferably vertical) and watch out for thermal-acoustic oscillations (TAO or Taconis effect).
5. Higher temperature loads; 20 K to 80 K which may include auxilliary heat shields or other fluids, e.g. hydrogen.
6. Availability of liquid nitrogen (LN) for pre-cooling in the cycle. Depending on logistics and price, LN can reduce capital costs, "LN equals another compressor" or provide for peak cool down loads.

7. Transient operations as well as steady state; special cool down rates including pressure drop (ΔP) and heat transfer as well as filling with LHe (liquefier mode). This is usually the maximum load condition because it requires holding all the static loads plus making liquid for fill. Recovery time after equipment failure.

8. Some allowance can be made for predicted future changes, usually in the direction of increased loads. i.e., magnet correction coils, current and diagnostic leads, stainless steel vs. titanium alloy conduction loss, distribution system growth, insulation malfunction or vacuum leaks, refrigerator degrade or other Murphy's Law events.

9. Selection of safety factor. After all loads are defined as accurately and realistically as possible, a multiplier is selected. For Isabelle (25 kW at 3.8 K), BNL selected a factor of 1.5. Both smaller and larger factors have been used. Some engineers would recommend (only partially in jest) e or pi, as in cost estimating.

Those who select design safety factors close to one may be later required to purchase large amounts of LHe to cool and fill their apparatus. Then they can never let it warm up until they have funds for more LHe. On the other hand, too large a factor can produce excess capital investment as well as excessive operating costs if the system can't be turned down for low steady state loads.

The wise selection of refrigerator size involves a careful design compromise between many factors, including heat load estimates, operating modes, LN logistics, power and manpower economics, non-linearity of refrigerator price vs. capacity, cost of capital and possibly even some politics. Heat load budgets must be fixed early and then rigorously policed through the project life.

I c. Specifications and Creative Contracts

Most large helium liquefiers-refrigerators are contracted out by both industrial companies and large research laboratories. Design and construction is highly specialized and it is usually more efficient to obtain competitive quotations. Sometimes inquiries are made initially to establish vendor capability and budgetary price range, often called a "request for proposal" (RFP). Later, firm prices are asked for with a request for quotation (RFQ). In order to be fair a well-defined specification (SPEC) is necessary. Simplicity should be the keynote, and will also tend to produce lower quotes. The fewer words the better and ideally a few pages should suffice for a simple performance SPEC.

Evaluation factors, penalty and incentive clauses need clear definition. Some past RFP/RFQ packages have been 200-300 pages, required three to five months and 250-300 K$ for vendor preparation. This is excessive.

A good SPEC may include:

1. Heat load guaranteed performance at given temperature level, simultaneous or separate with 2, below.
2. Liquefier capacity (either dewar fill or remote delivery, a possible difference of 14 percent dependent on cold gas return).
3. Any mixed mode operation.

4. Minimum pressure drop for certain flow conditions, cool down or increased mass flow, determines valve, pipe and heat exchanger sizing, as well as future up-grade limits.
5. Possibility of future capacity increase, usually limited by return side ΔP in HX.
6. Availability of LN for pre-cooling or rapid cool-down.
7. Rapid cool-down and warm-up capabilities if required.
8. Expander equipment, whether rotary or reciprocating, redundancy or back-up, auxiliary systems and parts.
9. Compressors, rotary or recip, stages, lubrication if any, oil removal and purifier, motor and starters, local control.
10. Purifiers for what contaminant and at what temperature, capacity and redundancy, re-cycle and activation time, mean time between maintenance (MTBM).
11. Plant control preferably a single occasional operator, local-remote, relay logic, microprocessor, computer data logging.
12. Heat exchangers and internal pressure vessels, construction, size, and materials.
13. Guaranteed maximum electric power use (compressor and cycle efficiency), LN use, maintenance and manpower needs, reliability and redundancies.
14. Penalty clauses for failure to meet performance both in delivered capacity and input requirements; for late delivery date and delay.

15. Incentive clause for early delivery, excess capacity over minimum spec., etc.
16. Progress payments, acceptance testing, method of evaluation and factors used in RFP and RFQ selection, warranty clause.

The contract implies a certain trust between buyer and seller. Prior to contract the buyer must be assured of the sellers competance and ethics to avoid costly problems later, like obtaining delivery and performance. It is also important to have several sources of supply to assure fair price competition.

The enforcement of the SPEC can sometimes be difficult and lead to tenuous negotiation. In general, it is best to avoid lawyers and litigation. A philosphy of "make peace not war" can be best.

A classic example involved the joint purchase of two 1500 W helium refrigerators by Fermilab/LBL in early 1975. The purchase became entangled in the Federal Court in Chicago and was delayed almost two years. Both buyers and sellers suffered and the attorneys profited most.

A more positive case is that of the Fermilab purchase of the Central Helium Liquefier, the world's largest. The minimum capacity specified was 3900 L/hr, with incentive payment for excess production. It is fully expected to deliver well over 4500 L/hr.

Karlsruhe's Kernforschungs Anlage, FRG put out an RFQ for a 300 W, 1.8 K helium refrigerator. Linde AG was the first successful seller. Later the German Government purchased a second unit from Messer-Griesheim, and KFA obtained two units instead of one as originally planned.

II a. Cycles, Design and Optimization

1.0. Process Description

The major difference between a liquefier and a refrigerator is the unbalanced flow of the liquefier. Make-up gas is added at the warm end continuously, while liquid is withdrawn at the cold end. The process can be thought of as a separate stream of gas being cooled and liquefied by a closed cycle refrigerator. This independent stream must be cooled from ambient to its liquefaction temperature, and the heat of vaporization removed. Cooling must be applied at all intermediate temperature levels. Because of the unbalanced flow, liquefier cooling requirements are more severe than those of a refrigerator.

For the closed loop refrigerator, both supply and return mass flows are equal. The refrigeration is supplied at the cold end by the evaporating liquid, and the temperature is constant as long as the pressure over the liquid surface remains constant. If the compressor suction pressure is fixed at one atmosphere then the minimum liquid boiling pressure is one atmosphere plus the pressure drop up the HX return side.

There are three cooling methods for helium liquefiers. The cascade system uses liquid pre-cooling baths at different temperature levels, with counter-flow HX to exchange the sensible heat of the boil-off gas to ambient. The first LHe systems used LN(77 K) and LH_2(20 K) fore-cooling. In some cases these liquids were vacuum pumped to lower their temperature. LN freezes at 65 K, but liquid air forms a eutectic in the 50 K region. Because of the complexity and hazard, LH_2 has been eliminated in most LHe systems. LN pre-cooling is powerful and still used in many systems. For a fixed output, or for cool-down and LHe fill transient cycles, LN pre-cool can reduce compressor capital costs. It can double liquefaction rates. The use of LN depends greatly on availability, logistics and delivered costs, as well as local power costs. In some cases it may be more economic to add an LN refrigerator.

In the second method, expander engines or turbines remove work from the gas and hence provide strong cooling to the process flow. The efficiency is very high because the external work removed from the system approaches reversibility. On the T-S diagram, ideal expansion occurs isentropically, along a vertical constant entropy line.

The third cooling method is the J-T (Joule-Thomson) process. Figure 2[6] shows the elements of a J-T refrigerator with state points on a T-S diagram. Compresssed gas enters the counter-flow heat exchanger (HX) at I, is cooled to 2 where it expands at the J-T valve. The temperature drops due to "internal work" done by the gas against molecular attraction. The cold gas enters the HX at 4 and cools the input gas until eventually liquid forms at 3. The expansion process is isenthalpic and occurs along a constant enthalpy line, where it enters the liquid dome. The enthalpy, H, remains constant before and after expansion. The process occurs adiabatically with no external heat entering the system. Such a process is non-reversible and therefore not efficient, but the J-T valve expansion has been used for many years because of its inherent simplicity and reliability. The absence

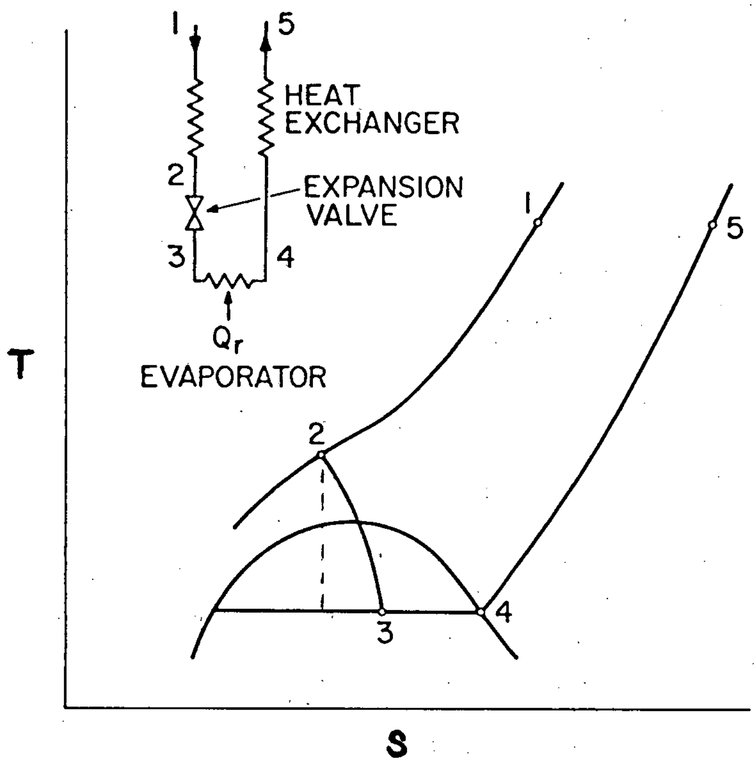


Figure 2. Schematic of Joule–Thomson Refrigeration System.

FIGURE 2 Ref. 6

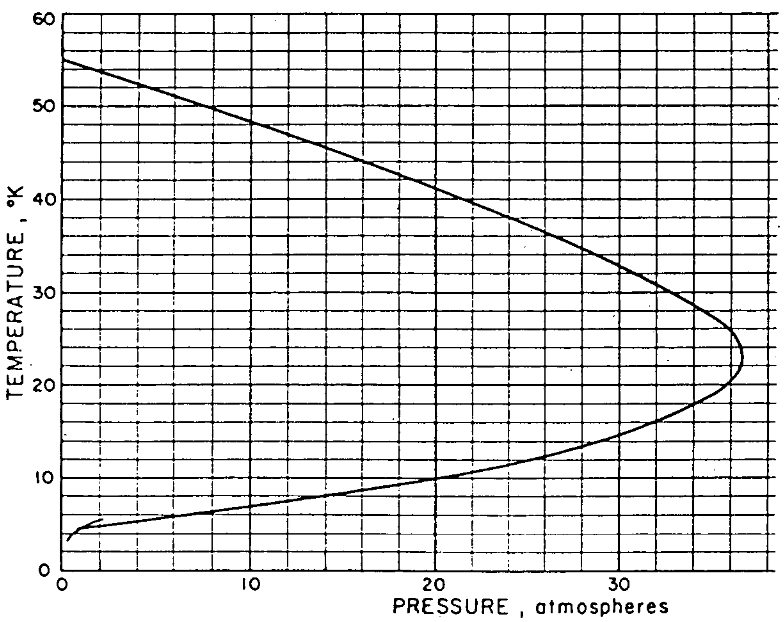


Figure 3a. Helium inversion curve.

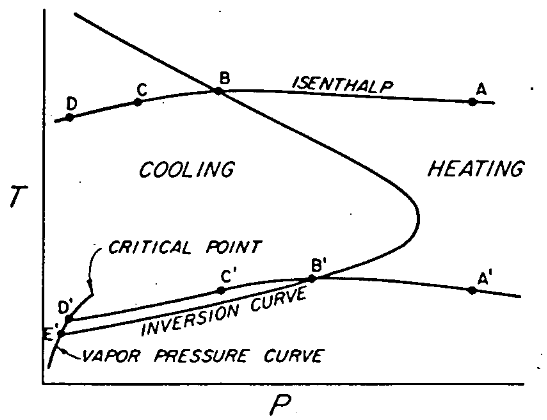


Figure 3b.

Joule–Thomson expansion process.

FIGURE 3 Ref. 6

of moving parts at 10 K is a great advantage. Ice formation is easily broken off when necessary.

The T-S diagram of Figure 2 shows the process traced along an isobar (constant pressure) in the HX from 1 to 2, expansion along an isenthalp (constant H) where it enters the liquid dome, and intersects the low pressure line of boiling liquid, usually one atmosphere pressure, 4.2 K for LHe. Actually it may be 1.1 atm and 4.5K to allow for pressure drop up the low pressure return side of the HX. This pressure drop sets the limit for the minimum temperature boiling point and may limit future increased mass flow capacity for plant up-grade.

The position of 3 along the liquid line indicates the quality (amount of gas) of the two-phase flow from the J-T valve. At 4, the quality is 100 percent saturated vapor and no liquid, a maximum S under the liquid dome. 100 percent liquid, zero quality, is at the left side of the dome, a minimum S. For a refrigerator the heat, Q_r, is applied to boil the liquid, and the mass flow is equal in supply and return streams. A liquefier will have make-up gas added at 1 or 5 (compressor discharge or suction) and liquid removed at 3. For a liquefier, 0.10 or less, of total compressor flow is the common liquefied fraction.

The J-T process was used by Linde (high pressure, 200 atm, Linde cycle) to liquefy air, and Kamerlingh Onnes with cascade LN and LH_2 pre-cool to liquefy helium. While fine for O_2 and N_2, the J-T process requires auxiliary cooling for gases (H_2 and He) which heat when expanded from ambient. Helium, for a practical system requires cooling below 20K (LH_2) and design temperature is usually 7-15 K.

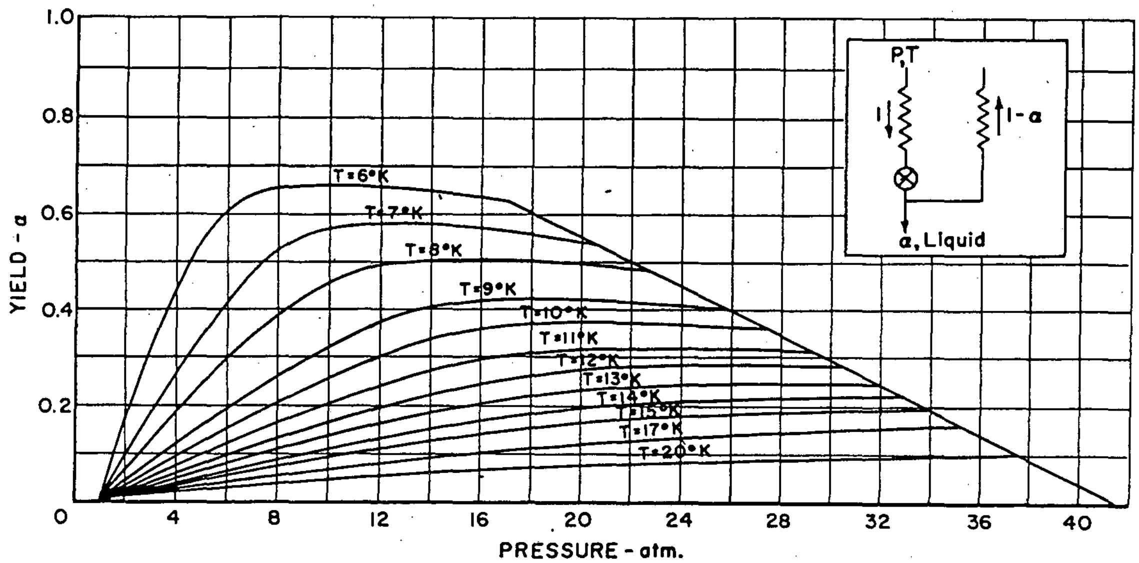


Figure 4. Yield α with J-T heat exchanger efficiency η = 100% as a function of HP inlet pressure and temperature.

FIGURE 4 Ref. 7

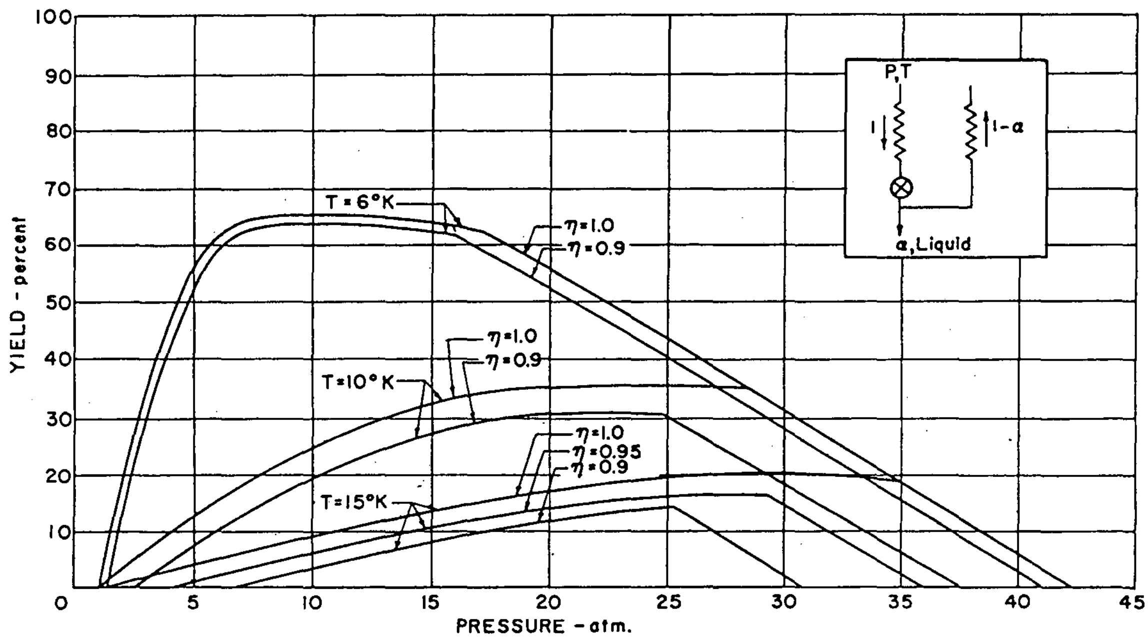


Figure 5. Yield α with J-T heat exchanger efficiencies η = 100% and η = 90% as a function of HP inlet temperature and pressure.

FIGURE 5 Ref. 7

Figure 3[6] shows the Joule-Thomson inversion curve for helium. Expansion from P and T-values inside the curve produces cooling, expansion from outside the curve heats. A curve of constant h has positive slope (cooling) inside the curve and negative slope (heating) outside the curve. The curve locus where the J-T coefficient equals zero, (slope change) is the optimum point for maximum cooling.

Figure 4 and 5[7] show the liquid yield as percent of total flow thru the J-T HX for varied P and T and HX effectiveness. The curves are based on a heat balance around the J-T HX boundary limits (Fig. 2).

$$mh_1 = (m-a)h_5 + ah_3$$

Heat balance for a liquefier.

$$mh_1 - mh_5 = a(h_3 - h_5)$$

m = mass flow entering J-T HX.
a = mass flow liquefied
h = enthalpy

$$\frac{a}{m} = \frac{h_1 - h_5}{h_3 - h_5}$$

For a refrigerator the heat balance is:

$$Q_r = m\,(h_5 - h_1)$$

where: Q_r = refrigerator load (h_5-h_1)

If the temperature at 5 is 20 K, the temperature difference T_1-T_5 must be less than 2 K or there will be no refrigeration at all. Typical temperature differences of 0.5 or less for the J-T HX are used.

The J-T HX is the most important part of the system. Small losses at the cold end create large loss on yield. Heat leak, HX effectiveness or cycle inefficiency can all contribute. From Figure 2 we can see that if we replace the J-T valve with a "wet" engine, the expansion from 2 to 3' is isentropic and intersects the liquid line at lower S (or more liquid). This is the reason for the growth of "wet" expanders since 1970. Liquid yields can be increased by 0.33 and refrigerator yields by 0.40.[8]

2.0. The Brayton and Claude Cycles

Low temperature expander engines are used extensively in cryogenic cycles. Both turbines and pistons are used, and the power produced is usually wasted. The primary function is to reduce the temperature of the gas expanding thru the engine so that the cold exhaust gas can be used for cooling.

The Brayton cycle consists of a simple closed loop with a warm compressor, a counterflow HX and an expander at the cold end. Were the cycle considered 100 percent efficient, completely reversible, then on a T-S diagram it would be an isothermal compression, cooling along an isobar in the HX, an isentropic expansion and heating in the HX return side along a lower isobar.

With He as the working fluid, the Brayton cycle can produce low temperatures of 15-20K. Claude in early attempts to use an engine to liquefy air, found heat entering the cylinder and liquid forming on the wall to be a major problem. The heat load was moved external and downstream of a J-T valve. Thus the Claude cycle Figure 6 combines the Brayton and the Joule-Thomson cycle for high efficiency. Expander engines are used to take some of the flow and use it to pre-cool He into an efficient range for J-T expansion.

The Collins He liquefier cycle is shown in Figure 7[9] as an example of the Claude cycle. Where Claude used only one engine, Collins used two for pre-cooling before the J-T valve. The more expanders in the system the closer the approach to reversibility and the greater cycle efficiency. A comparison of relative mass flow required for the

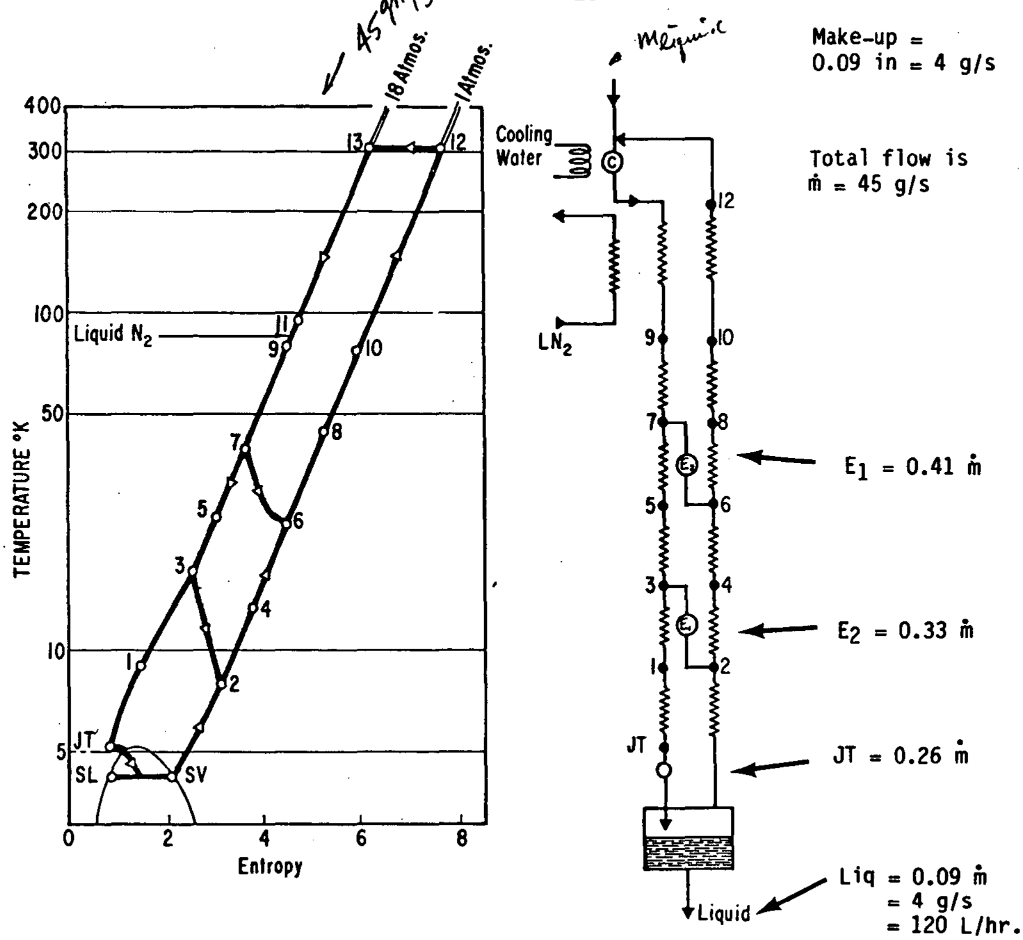


Figure 7. TS diagram of liquefaction cycle with liquid nitrogen precooling, ADL Model 60 Helium Liquefier.

FIGURE 7 Ref. 9

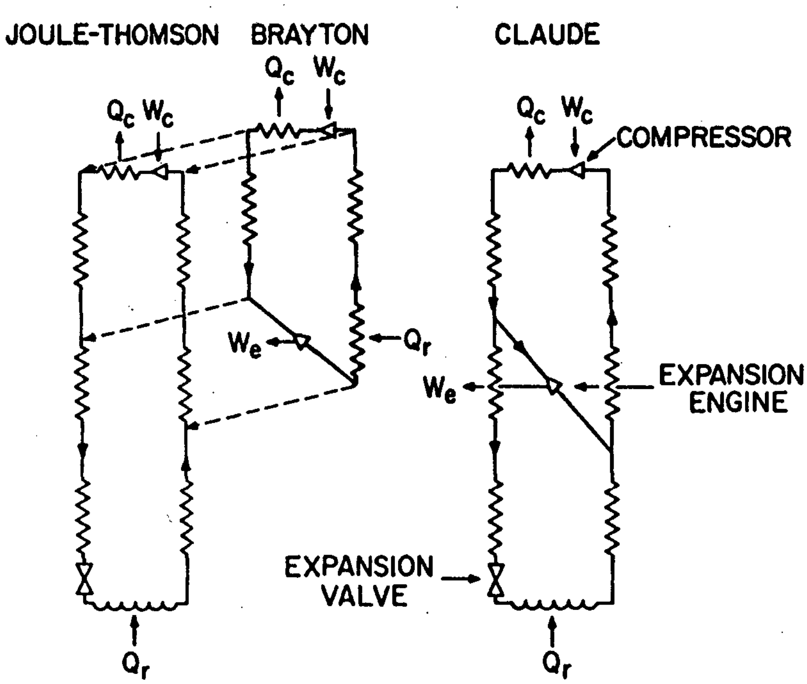


Figure 6. Schematic of Claude Cycle Refrigerator composed of superimposed Joule-Thomson and Brayton Cycles.

FIGURE 6 Ref. 6

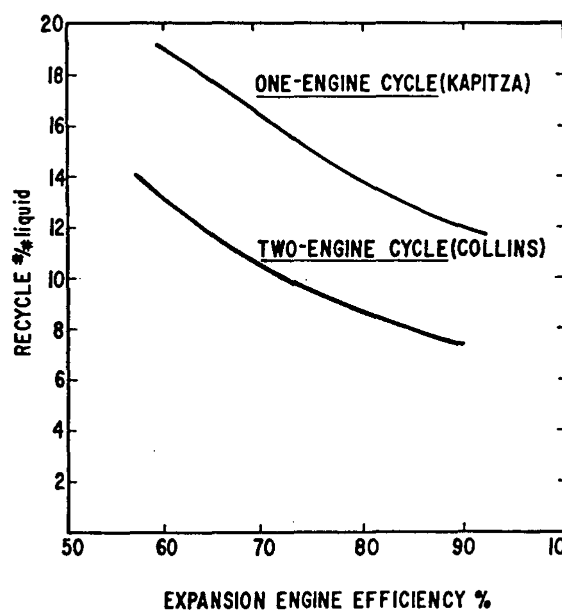


Figure 8. Comparison of Kapitza and Collins Cycles.

FIGURE 8 Ref. 9

Kapitza (1934) He liquefier and the Collins (1946) two engine machine is shown in Figure 8.[9]

3.0. Process Efficiency

Helium system design is governed by the first and second laws of thermodynamics and conservation of mass. With the first law, "The energy of the world remains constant" heat and mass balances can describe the result of a process. The first law is valid for the result occurring in either direction. The second law, "The entropy of the world increases towards a maximum" can predict allowable directions for a process to occur. It involves entropy, thermodynamic temperature scales, Carnot cycle and reversibility. More recent terms include "lost work", availability (which refers to energy available during and after the process) and exergy. How close to reversibility or how little available energy is lost gives a measure of process efficiency.

In general, a process of many small steps is more efficient. A ramp is easier to climb than a stair and many little steps are easier than one big leap. Thus the reason for many expander engines and many compressor stages for efficient systems. For HX, small temperature differences are more efficient than large, at the ends as well as thru-out the length.

It is convenient to compare real system performance with ideal cycles as defined by Carnot. For a refrigerator operating between two temperature limits, Carnot cycle efficiency is given by:[10]

$$\frac{W}{Q} = \frac{T_o - T}{T}$$

$$= \frac{300-4.2}{4.2} = 70.2$$

where W = net power input
Q = refrigerator output
T = absolute temperature at which Q is produced
T_o = absolute temperature at which heat is rejected

For a refrigerator operating between 4.2 K and 300 K, 70.4 watts is the minimum power required for a watt of cooling at 4.2 K. A real machine of one watt at 4.2 K might have Carnot efficiency of one percent and require 7 kW of input power. For 1 to 10 kW output, efficiencies improve to 20-25 percent or 280-350 W per W.

Strobridge[10] further defines Carnot cycle efficiencies for a refrigerator operating between T_H and T_L (high and low temperatures) and where there is no phase change as:

$$\frac{W}{Q} = \frac{T_o}{T_H - T_L / \ln \frac{T_H}{T_L}} - 1$$

Let $T_H = 8$ K
$T_L = 6$ K

$$= \frac{300}{(8-6)/\ln \frac{8}{6}} - 1 = 42.2$$

and for a liquefier:

$$\frac{W}{m} = T_o(S_o - S) - h_o + h$$

$$= 236 \frac{\text{W-hr}}{\text{liter}} \text{ at } 4.2\text{K.}$$

where:
m = mass liquefied/unit time
S = specific entropy
h = specific enthalpy
o subscript refers to ambient,
no subscript refers to liquid

Consider the simple Brayton cycle, power is fed to the compressor and heat is added to the expander exhaust, with power removed at the expander. If the compressor, expander and heat exchanger are all 100 percent efficient the cycle is reversible and defined by Carnot. Real compressor efficiencies are about 60 percent, expanders about 75 percent. These two items comprise the major system losses. Other losses, HX, ΔP and ambient heat leak are of lesser effect, say 10 percent of total.

System losses consist of:

1. Compressor, mechanical and thermal.

2. Expander, mechanical and thermal.

3. HX effectiveness loss, consists of three parts:

   a. Area for HX and warm end dT,
   b. Differences in mass rate and specific heat product in the HX streams.
   c. Fouling by impurities.

4. Pressure drop thru HX, valves, pipes, etc.

5. Heat in-leak from ambient.

Compressor efficiency is compared to an isothermal process which appears as a horizontal line on the T-S diagram. Real compression appears with rising temperature, or if multi-stage as a saw-tooth of temperature rising and falling as intercooling takes effect.

Expander efficiency of 100 percent appears as a vertical line between two isobars. Lower efficiency of a real process finishes at a higher temperature than ideal. The ratio of real vs. ideal enthalpy change gives expander efficiency and real expanded temperature.

4.0 Design and Optimization[11-18]

Design of a helium plant is a complex economic and thermodynamic problem. Like many engineering problems, it requires considerable pre-knowledge of proposed components. Expander performance and mechanical design, (bearing loads) are important. A small increase in

HX effectiveness might require doubling the size and be prohibitive in cost. Compressor type, performance, stages and costs, both capital and operating, are all critical factors. With components selected, the circuit arrangement can then be planned. This includes staging of compressors, single or dual pressure cycle and series or parallel use of expanders.

The non-ideal properties of helium, especially in the low temperature region, require detailed HX analysis to assure no Second Law Violations. Real gas specific heat values may require stages of J-T valves for full HX use. Variable gas properties also require optimization of expander pressure ratios and inlet temperatures.

With the design heat load or liquefaction rate set, then a heat balance is made around the J-T HX. With selected fixed entrance temperature and pressure, the mass flow required is established for the given yield at the fixed temperature and pressure. Initial studies can be made assuming 100 percent efficient compressor, expander and HX. Heat and mass flow balance equations are written for each HX and expander sub-loop. Comparison basis can be Figure of Merit:

$$\frac{W_c}{Q} = \frac{RT \ln P_1/P_2}{m(h_1-h_2)} = \frac{\text{isothermal compressor work}}{\text{refrigeration delivered}}$$

or: W/m_1, work per mass liquefied,

or: m_c, total circulating mass required,

or: t, temperature reached.

Expander inlet (or outlet) temperatures and pressures can be examined. Minimum input W_c (or m_c) can be found as a function of compressor discharge pressure, number of expanders and mass flow split thru the expanders. A Claude cycle optimized as a liquefier might have 0.20–0.35 m_c flow thru the J–T lower HX, or 0.50–0.65 m_c as a refrigerator. To further refine the study, expander and HX efficiency can be added as parameters. For a given set of efficiences and a fixed inlet pressure, there is an optimum engine inlet temperature. The J–T part of the cycle becomes more efficient the lower its inlet temperature. This temperature is a function of the expander exit temperature and its efficiency. As the pre-cooling expander temperature is lowered, its refrigeration decreases and it requires more flow and input power. Optimum engine inlet temperatures and flow split is a balance between these effects. The heat transfer limits of the J–T HX must not be exceeded. (HX pinch effect.)

While hand calculations can be adequate for optimization, large complex system analysis now uses the computer. An algorithm is written for the heat balance of each HX. Efficiencies, pressure drops and heat leak can be included. Thermophysical properties of helium from the National Bureau of Standards can be used as sub-routines. Parameters are varied and iterations of the simultaneous equations are made until the solution is closed. Many variables can be thus quickly examined.

II b. Compressors

The compressor provides the foundation or heartbeat for the system. The greatest system energy loss stems from the compressor, and with the high helium specific heat ratio;

$$k = c_p/c_v = 1.66 = 5/3$$

there is much potential for heating. If we plot:

$$PV^n = NRT = K \text{ (a constant)}$$

on a P-V diagram the maximum work occurs when $n = 1.66$ and the minimum when $n = 1.0$ (isothermal) with real compression somewhere in between. On the T-S diagram isothermal compression is a horizontal, constant temperature process from one isobar, P_1 to another. The isothermal work done in compressing a gas from p_1 to p_2 equals $\dot{m}\, RT \ln [p_2/p_1]$ and it is useful to use the isothermal efficiency;

$$n_{iso} = \frac{0.61\, \dot{m} \ln (p_2/p_1)}{W_{real}}$$

where
W = input power, kW
$\dot{m}$ = mass flow, g/s

R = helium gas constant = 2.08 J/gm °K
T = absolute suction temperature = 293 °K

for comparison of real machines. For reciprocating compressors using piston rings, 0.50 to 0.60 is common, ranging up to 0.70 for large machines.[18] The oil flooded screw compressor efficiency can range from 0.45 up to 0.60 for large units.[19] The reciprocating machine can use oil lubricated piston rings, plastic rings, combinations of both called mini-lube, or labyrinth seals. For helium, the stages should limit compression ratios to about 2.5 to control temperature rise. Provision must be made for oil removal, and plastic ring wear fragments must be filtered. The non-contaminating labyrinth sealed compressor provides clean gas and little wear for reliable service. Plastic ring machines require low speeds, temperature control and ring service every 3000 hours.

A quest for greater reliability and longer service intervals led to development in the early 70's of the oil flooded helical screw compressor for helium service. With other gases this machine has had up to 50,000 hours MTBM. The compressor has many advantages; it is positive displacement, it has no rings or valves to wear, initial capital investment is low, oil injection provides a good approach to isothermal compression and He loss due to leakage is minimal. High compression ratios per stage are possible and the mass flow throughput is easily reduced for power saving. Thoroughly degassed (air) and dewatered synthetic oils are used. Oil removal systems for impurity levels of less than 1 ppm_w (10^{-6}) are required. Screw compressors have now been installed in many helium plants, and some units have accrued operating times well in excess of 30,000 hours.

The present trend has been to install screw compressors for all new commercial as well as research helium plants. The low initial capital costs are attractive, but the greatly reduced service and maintenance needs are the major reason, far out-weighing any advantage the reciprocating machine may have in isothermal efficiency and power costs.

II c. Heat Exchangers (HX)

The counterflow heat exchanger serves to isolate the 300 K level from the 4.2 K cold end and provides a handling method for the process helium gas. Design challenge comes from maximizing the effectiveness, or efficiency of the HX and minimizing the costs and pressure drop losses. The modern brazed aluminum plate-fin HX best answers the need for the large systems, and can approach 0.99 effectiveness. Typical warm-end temperature differences (ΔT) might be 4-5 K at 300 K, 1-2 K at 80 K and 0.1-0.3 K at 7-15 K. Reducing the ΔT's by half might improve overall performance by 4-8 percent. Cycle losses at the cold end have severe impact on performance and not only must ΔT's be minimized but Second Law violations must be avoided. These occur due to thermal property variations in cold helium gas, and can sometimes be cured by changing the high side pressure levels with multiple J-T valves.

In general, system HX are placed with the cold end down to prevent convection effects and thermal mal-distribution. If this position is reversed, or HX are horizontal, or design requires a future large mass-flow increase or low flow mode, then the HX must be carefully examined for effective transfer area loss due mal-distribution.

In fabrication, quality aluminum to stainless steel transition joints must be assured for tolerance of thermal cycles, shocks and impurity build-up. Provision for cleaning and de-riming is required. Sometimes water, oil, air and teflon ring dust needs purging. Severe corrosion has sometimes occurred in Al HX due to water and/or possible acids from industrial air. Another failure mode may be ice-wedging in brazed HX (analogous to granite fracturing in the Sierra). To help solve these problems, operations must avoid air exposure of cold HX. Off-cycle or rapid warm-up modes need to avoid freezing, melting and trapping LN, which can create enough volume change to burst the HX.

II d. Expanders

There are two general types of expander engines for external work removal in a Claude or Brayton cycle. The first is the reciprocating single piston engine with inlet and exhaust valves. Engine loads are usually electric braking or oil pumping and cooling. The energy in most helium systems is too small to recover and is wasted.

The single piston engine must be well isolated from 300 K heat gain and can have cold piston rings, seals at 300 K or labyrinth cold seals. Valves can be arranged as overhead, "L" head or slide types. It is desirable to keep cylinder clearance volume at a minimum. Slow speeds are also preferred for low wear on valves and seals, and thus reduced maintenance and outage. Minimal leakage thru valves and rings is also necessary for high efficiency.

Recips can be used in the 40-20 K region in plants up to 1200 W or 300 L/hr capacity. "Wet" recip expanders in the two-phase region are now commonly used in large LHe plants of 800-1300 L/hr. Typical efficiencies for warm or "dry" expanders are 0.70-0.80; and 0.80 for "wet" engines. These engines are low flow, high pressure devices and as the plant size and $\dot{m}$ increase, the reciprocator becomes flow limited. (Compressor characteristics are similar; centrifugals supercede recips at high flows.) Recips are usually arranged in the system in parallel, expanding the high side flow in one step to the low side or return. Pre-expansion pressures up to 25 atm are used.

The recip, in its working range, in general has better efficiency than a turbine and rather constant high efficiency thru a wide mass flow range. This permits the plant to operate well over many modes,

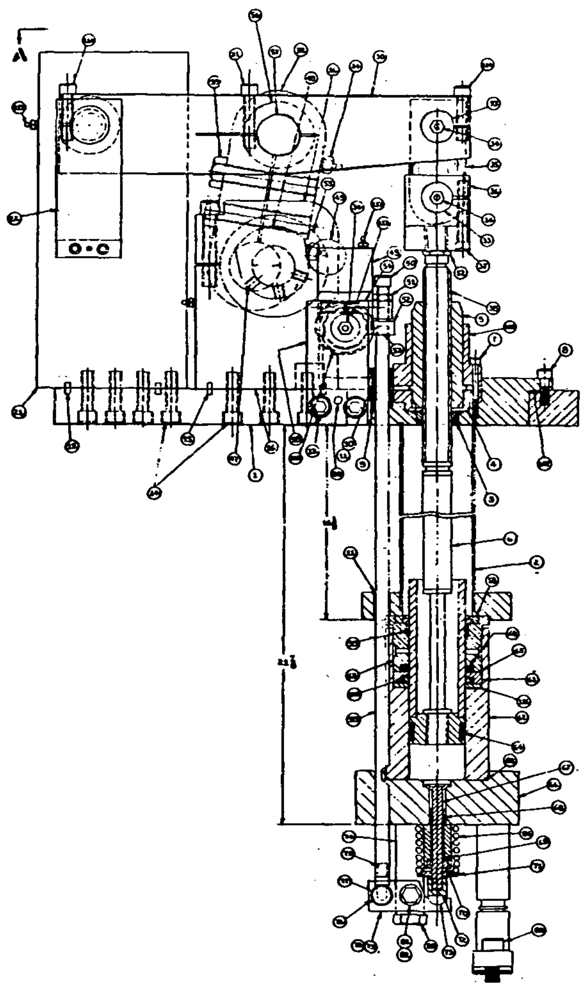


Figure 9. Fermilab reciprocating expansion engine.

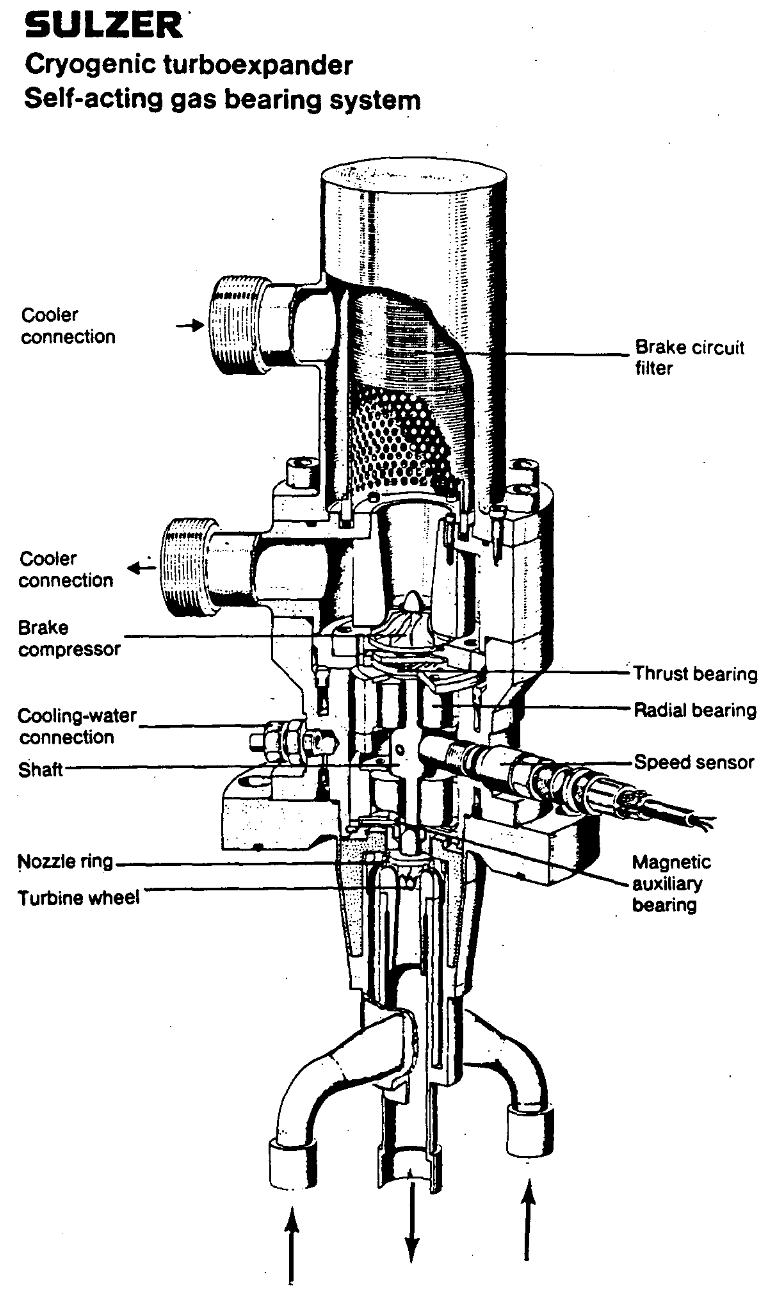


Figure 10. Sulzer Turboexpander (gas-bearing).

i.e. cool-down, LHe fill, and steady state. Efficient plant turn-down is feasible. A two engine recip plant in the 300 W range might require 0.30-0.40 less $\dot{m}$ and power input than a similar two-stage turbine plant. Another advantage is that operators can perform required maintenance with little outage or down-time. Teflon ring expanders presently achieve 3000-8500 hours MTBM. For near absolute reliability some install redundant engines.

The turbo expander is a high flow, low pressure device. Turbines might operate in the 300 to 4000 rps region. High speed is necessary to keep efficiency up, otherwise the leakage degrades it greatly.

Turbines are usually arranged in the plant system in series, so that pressure drops can be efficiently handled. For small systems expander wheels are small, clearance to diameter ratio is high, leakage is therefore high and efficiency is low. As wheel size increases, clearance is constant and efficiency increases. For very small plants it is almost physically impossible to build a small enough turbine and still maintain efficiency. For the LBL 1500 W refrigerator the upper turbine has a wheel diameter of 20 mm, flow is 110 g/s, p_1 = 17 atm, T_1 = 23 K and p_2 = 3 atm, T_2 = 15 K. The lower turbine is 12 mm dia. for 17 atm and 7 K inlet, 80 g/s.

Another problem is plant turn down, or variable $\dot{m}$. For this to be efficient, the inlet turbine nozzle area should be varied. This is very difficult if not impossible in small turbines. The BNL refrigerator has Rotoflow turbines with some wheel diameters of ~117 mm and uses variable inlet nozzles. An effective method for plant turndown is to reduce high and low pressures, holding turbine volume flow and pressure ratios constant, or reduce only the high side pressure.[15]

Turbine types can be classified by bearing/brake systems. The oil bearing turbine is rugged and has high stiffness. Work is removed by hydraulic friction in the oil brake. The cold end is mounted up so oil falls by gravity and doesn't contaminate the helium process stream. The oil sub-system with filters, regulators, etc. is an expensive part of the turbo-system. Manufacturers include Air Products and Chemicals, Inc. (APCI), Linde A.G., Rotoflow and Sulzer. In the smaller turbines, the oil bearing hydraulic losses would be too great to allow the wheel to turn at efficient speed. Thus, the gas bearing, with almost no friction, is needed for small turbines. It also is replacing oil systems because of their sub-system costs.

The gas bearing turbine can be sub-divided into two general systems, the hydrostatic (or gas injected) and the hydrodynamic or self-pumping groups. Both systems use brake wheels to compress gas on the warm end and water HX to remove heat energy. Cold end is usually mounted down (but sometimes horizontal in the case of L'Air Liquide).

The gas-injected or static bearing is robust with high bearing stiffness. It can take high shock loads, but can use 0.5 to 3 gm/sec of compressor output. This is not too much to pay for the reliability of the turbine. This turbine type is often used with expansion ratios of up to 8-10. Sixsmith at Oxford developed this turbine type early (1960) and BOC and L'Air Liquide followed with manufacture and development. The French firm uses a shrouded impeller wheel claiming more efficiency due to less leakage.

The hydrodynamic bearing pumps its own pressurized gas to support the loads. This is done in an enclosed area by wedge action against floating shoes. Stiffness is low and the thrust bearing cannot take much shock or impact. Thrust bearings are usually supplemented with permanent gas injection for start-up and shut-down or more recently with magnetic bearings ($SmCo_5$). This type turbine prefers to operate at lower pressure ratios of 2-5. Sulzer has built many units of this type with shaft diameters of 16, 22, 32, 45 mm. Many thousands of operating hours have been accumulated world-wide by the Sulzer turbine. Hitachi, more recently, for their 300 W refrigerator, builds a helium turbine with radial dynamic floating shoes, a static injected gas thrust bearing and shrouded impeller. Other forms of dynamic bearings have been used. Airesearch has used foil bearings for many years in aircraft Brayton cycle coolers. Early foil bearings were developed in California for computer high speed tape drives.

Gas bearing turbo expanders have typical efficiencies of 65-75 percent. They are compact, have very high speeds with little friction and no wear. Hence they can be very reliable with long operating times, say in excess of 20,000 hours MTBM. For high flow and steady loads requiring continuous on-stream time, turbines are the only solution.

II e. Control, Instrumentation, Purity and Gas Management

Most refrigerators can be operated with very simple controls. A transducer on suction pressure responds readily to heat loads and a few automatic valves actuated from an industrial controller usually suffice. Such a system can operate for long unattended periods under steady-state conditions. Of course, fail-safe design is required for all fault modes including power failure. For the transient modes such as cool-down, LHe fill and warm-up, manual control is the most direct method.

Computer control and micro-processors are important new developments, but can become very expensive in software and hardware development for single systems if not limited in scope. Close coordination between programmer and process designer is necessary. For monitoring, computer data-logging and archiving is very useful. With multiple systems scattered over wide areas or remote equipment, computer control and monitor is invaluable and necessary for minimizing man-power costs.

Gas purity is important for long term reliability. Neon build-up at the cold end and air, water, oil and plastic compressor ring dust at the warm end can lead to gradual performance decay or sudden stops. On-line detectors using optical, spectrographic and conductivity methods are used for system observation and purifier activation.

Gas management is important and resolves into the old engineering problem of operating vs. capital costs. Helium gas can be costly, and systems with large LHe inventory need methods to recover and store gas. Installed costs for medium pressure (20 atm) vessels, high pressure (200 atm) vessels, compressors and purifiers can be expensive. With developmental magnets subject to sudden quench, gas holders or balloons are used. Even with a simple system some ballast volume, gas or liquid, is needed for compressor/cold box operating stability. Rare gases, Ne, D_2, as well as He are sometimes stored or transferred to cold dewars as liquid under fault conditions.

II f. Distribution and Cooling Methods

Transfer and handling losses of single phase, two-phase, subcooled, supercritical, or superfluid helium can comprise major system losses. After great effort to achieve high efficiency in the liquefier, the distribution system is sometimes neglected and high loss or even non-delivery can result.

The large helium extraction plants still distribute high pressure gas (> 150 atm.) in cylinders; small bottles, semi-trailers and railway tank cars. A more economic method, since the early 1960's, has used vacuum-insulated LHe semi-trailers, of 5000-10,000-gal capacity. These trailers are driven to local distribution centers, ~1500 miles distant or air freighted overseas, where they are set up for liquid or gas local delivery.

The extraction plant liquefiers work into large LHe dewars, up to 32,000-gal. Efficient design and procedures are especially required in cool-down and filling of warm returned LHe semi-trailers. Stratification and liquid behavior during vessel depressurization must be considered.

Pool boiling is the oldest method for cooling apparatus. For an open system, where dewars of LHe are used to supply equipment, the heat of vaporization (latent heat) is used and the available refrigeration (5 → 300 K, sensible heat) is usually wasted, altho sometimes it is used very effectively for shield or power lead cooling. Equipment is designed for heat transfer in the nucleate boiling region. Large systems are only economic for a few days operation with open loop LHe supply. Because of gas and refrigeration loss a closed loop refrigerator becomes mandatory.

Heat loads with magnets or electrical apparatus require the cold box to operate in the mixed mode. The refrigerator mode cools the load at 4.5 K, but for electrical current leads of 2000-10,000 amps, flow is required from 4.5 K to 300 K to isolate the heat load. This flow is usually 0.1-0.3 g/s and constitutes a liquefier load.

LHe can be transferred by pressure, gravity and cold vapor or liquid pumps. The standard method is pressure and bottom liquid withdrawal. For laboratory dewars or large semi-trailers no external work or warm gas is needed if the initial pressure is high enough. For refrigeration closed-loop systems the pressure can be supplied at the compressor (300 K) via regulator valves. Thermodynamically, it is more efficient to add the pump energy at 300 K, rather than with cold liquid or vapor pumps. The lower the temperature the more expensive are energy inputs.

For large systems the pressure transfer system has the cold box supply a large LHe dewar 4000-10,000 liters, at a slight positive pressure. LHe is then transfered to the heat loads via demand valves. BNL operated their four-magnet string in a high energy unseparated beam line this way, and also their 2 m bubble chamber SC magnet. Early BNL Isabelle accelerator design planned to use this technique. Fermilab's

1500 W 4.5 K magnet test stand uses this method as well as their 15 ft. bubble chamber magnet.

Gravity systems are simple, usually an overhead LHe dewar feeds the heat loads below. The LHe storage can act as a buffer volume and emergency refrigeration source. LBL uses gravity on two systems, a large DC magnet (HISS) of 2 m dia x 1 m gap, and a cryopump system of 30 m^2 for fusion development. LLNL (Lawrence Livermore National Laboratory) uses a pressurized 25 kL dewar to feed two 4.4 kL dewars which gravity cool a 1400 W cryopanel heat load and a 500 W magnet load for the Mirror Fusion Test Facility (MFTF).

Two-phase forced flow is a powerful cooling method. The stream leaving the J-T valve consists of liquid and vapor. The fraction is referred to as quality, and as in steam, zero quality is all liquid, and 100 percent quality is saturated gas. Only the liquid component is used for cooling, and when all the liquid has boiled, the cold gas returns to the J.T. HX return side. The forced flow is very effective for long strings of high impedance magnets, or long circuits of cooling tubes. Some caution is needed in design for minimizing pressure drop. The drop thru the heat load plus the drop thru the cold box HX to compressor suction determines the temperature at the equipment due vapor pressure effect. Carefully examine two-phase flow phenomena which can create serious fluid oscillations. Flow regimes in the plug, slug and mist regions are important.

Fermilab uses two-phase flow in cooling the long magnet strings of the new Tevatron accelerator. LBL uses it for the Time Projection Chamber (TPC) Solenoid DC Magnet 1.5 Tesla, 2.0 m dia by 3.4 m long.[20,22]

At DESY, Hamburg two-phase forced flow is also used for the "CELLO" magnet 1.5 m dia by 3.5 m long, 1.4 Tesla.[21] For the "ESCAR" project, planned as a small pilot project superconducting accelerator, LBL planned to cool 56 magnets by pool-boiling LHe, fed by 100 g/s of two-phase low pressure drop flow. A twelve magnet string, with 100 m of transfer line and load simulating heaters was tested satisfactorily in 1977.[23]

Above 2.3 atm helium is supercritical, the phase is undefined but is not liquid. Density can be greater than liquid, but heat transfer may be limited, dependent on density. No phase change occurs on heating. System design is greatly dependent on First Law considerations:

$$Q = m\ c_p dt.$$

For low dt, needed for uniformity and maximum SC magnet performance, rather high mass flow, $\dot{m}$ is necessary.

The BNL power transmission line (PTL) project uses supercritcal flow on a test section of high voltage transmission line. Expander turbines are located at the end of the line heat load and prior to entry on the return of the J-T HX. The BNL Isabelle colliding beam accelerator (CBA) planned to cool 1000 magnets with supercritical helium, 4000 g/s at 5 atm and 2.5 K will rise to 3.6 K. Test operations with four and eight magnet strings have been satisfactory.

An advanced cooling method employs superfluid He II at 1.8 K. Lowered temperature should increase magnet critical current and field, but early tests at the equilibrium pressure, 15 to 30 Torr, had poor results. In the early 1970's, Claudet of CENG France noted that He II pressurized to 1 atm could remove more heat. Since 1978-79 1.8 K,

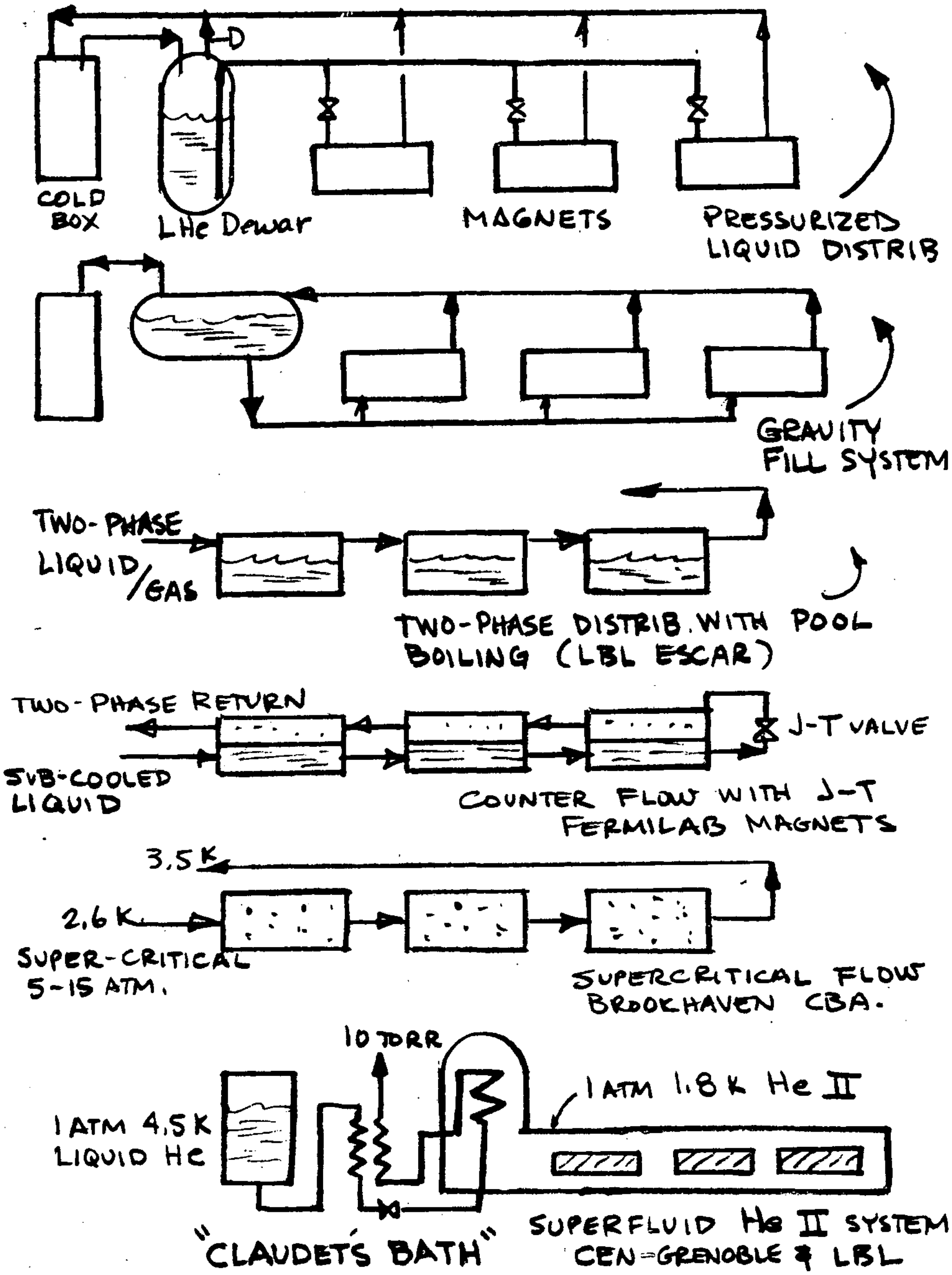


Figure 11.

Refrigeration distribution methods

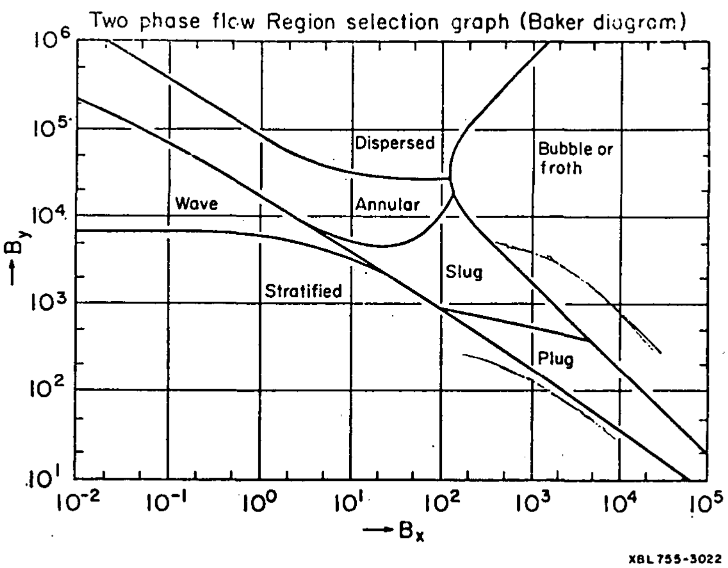


Figure 12. Two-phase flow regions.

1 atm test facilities at Saclay, Grenoble[25,26] and Berkeley[24] have been installed and used. Significant magnet improvements are reported. The use of He II not only improves the critical SC properties thru lowered temperature, but the mass and thermal transport properties are enhanced thru the remarkable nature of the superfluid. More system complexity and Carnot cycle inefficiencies are the price paid for these benefits. Design is always a compromise.

III a Large Helium Plants

Since the mid-50's the growth of cryogenic research and applications at 4.5 K has been steady and expansive. Now there are many refrigerators in the 300-1000 W range at research centers thru-out the world. Correspondingly the bulk LHe plants for extraction from natural gas have also multiplied and expanded.

This section surveys some of the worlds largest helium plants and comments on salient features. In describing a system a schematic, a T-S diagram, equipment lists and performance should suffice. The plants described all use the Claude cycle with expander turbines and Al brazed plate-fin HX. Most of the new plants use oil flooded screw compressors.

III b Large Liquefiers

Most of the large commercial helium extraction plants have capacity of about 1000 L/hr ($\pm$400). The natural gas wells located in the mid-western U.S. are the major helium source. The helium plant investment decision is severely governed by the amount of nitrogen and other impurities in the methane. If the gas is "lean" and cryogenic processes are used for N_2 removal, then the feed helium gas can be delivered at 80 K and 20 atm. This helps the economics considerably. Other factors are location, transportation and demand.

Because of the rising demand in Europe a 1000 L/hr (150×10^6 std. cu. ft./yr at 0.72 availability) LHe plant for Poland was designed (1971) and built (1974) by Petrocarbon, Ltd. and BOC in England.[27] Plant design is closed circuit Claude cycle refrigerator with high efficiency feed gas purifiers (charcoal and molecular sieve)

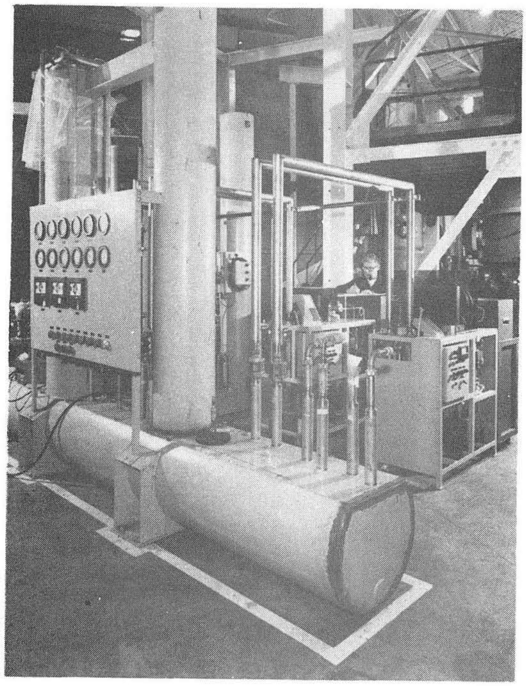


Photo. 1. (CBB 781-14311). The first one of three 400 watt refrigerators at LBL by Cryogenic Consultants Inc. (CCI). Construction is modular, based on the Fermilab Satellite Units. HX I, II are in the left hand vertical column, HX III in the horizontal and HX IV and V in the right hand vertical column. In the rear, from the left, are the charcoal adsorber (80 K) vertical column, the wet expander engine (7 K) and the dry expander engine (20 K). LBL photo.

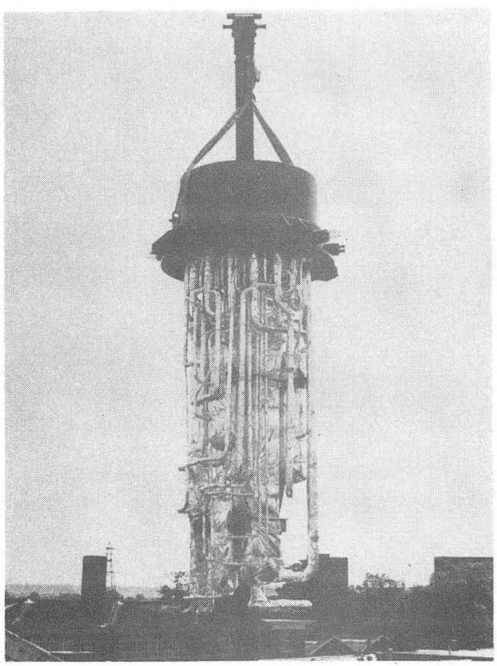
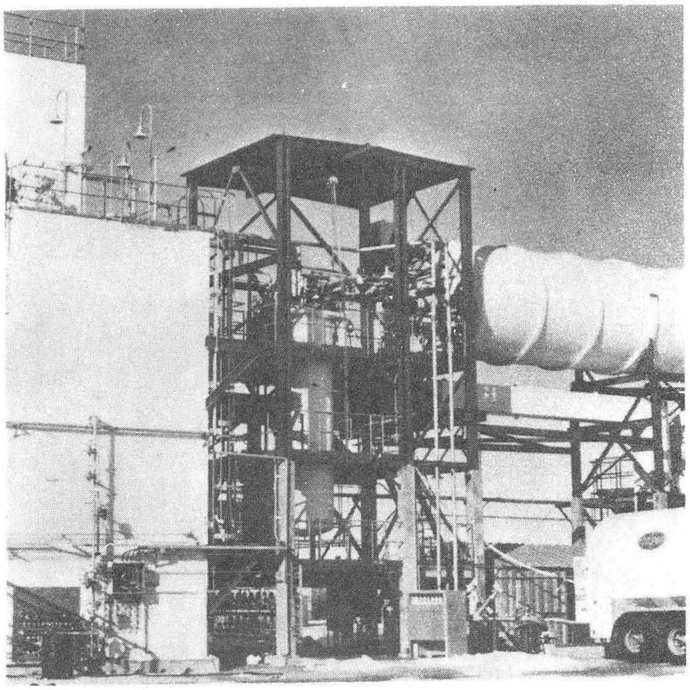
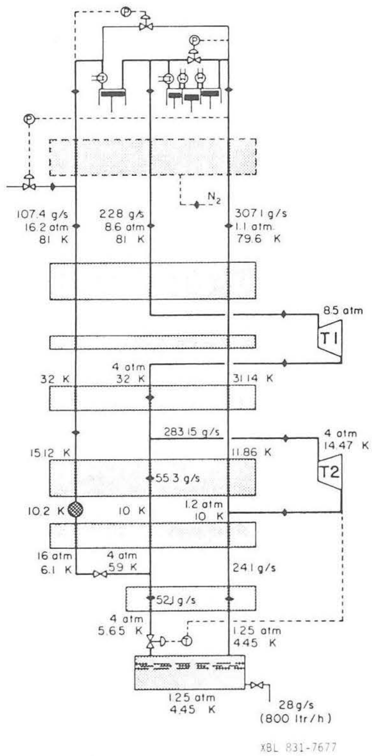


Photo. 2. (XBB 831-449). The 1000 L/hr liquefier cold box for Poland. Photo courtesy of BOC.



<u>Photo.</u> 3. (XBB 831-450). The 800 L/hr liquefier plant at Otis, Kansas. The large 30,000 gal, dewar at right is used to gravity fill trailers (in foreground). Photo courtesy Sulzer Bros.



<u>Photo.</u> 4. (XBL 831-7677). Schematic of the Otis plant. Feed-gas gas is added at make-up valve (upper left) at 16 atm, 80 K. 28 g/s is added to compensate for 800 L/hr withdrawn at bottom right. Photo courtesy Sulzer Bros.

and brazed Al plate-fin HX. The non-lube reciprocating compressor limits compression ratios to 2:1 and discharge temperature to 160°C because of the PTFE piston rings. Lucas gas-bearing turbo expanders are used. The LHe product goes to a 30,000 gal. dewar 30 ft above ground, and thence to trailers by gravity transfer. Until the construction of this plant there was little foreign helium production. A plant built in Canada in 1963 had a potential of 12 x 10^6 SCF/yr. Two plants started in 1968 were expected to recover 15 x 10^6 SCF/yr of GHe from Dutch North Sea gas. Estimates for European helium use in 1971 were 65 x 10^6SCF/yr. Helium has doubtless been produced in the USSR but few details are available.

There are two large recent LHe plants built in the U. S. by Air Products and Chemicals, Inc. The first, for Cities Service Helix, Inc. at Ulysses, Kansas in 1968 has capacity of 850 L/hr. The second APCI plant (ca 1981) on the Texas-Oklahoma Panhandle with capacity of 250 x 10^6SCF/yr or 1200 L/hr has a reciprocating compressor of 1500 HP (1125kW) and a very high transfer efficiency for trailer filling (0.93 instead of the usual 0.70).[28]

There are four large bulk LHe plants in the midwest using Sulzer oil bearing turbines. The Kansas Refined Helium Corp. Plant at Otis, Kansas went into operation in 1966 with a production of 800 L/hr.[29] LN cooling is used from 300 K to 81 K, two turbines in series from 81 K to 10 K and two stages of J-T valve cooling to 4.5 K. Turbine characteristics are:

	Temp.	Wheel	Speed	Efficiency	Pressure
	outlet	diam.	RPM		Ratio
1st Turbine	32 K	45 mm	95,000	0.78	8.6:4
2nd Turbine	10 K	50 mm	61,000	0.81	4:1.2

Plant efficiency was estimated by Trepp using an exergy method, at about 21-24 percent. Power input was 720 kW. In 1971 CTi-Helix added a wet expander reciprocating engine that increased capacity by 30 percent to 1040 L/hr.

In 1972, a Helix-Sulzer Plant was built at Ulysses, Kansas (Cities Service) for an initial production of 800 L/hr. However the design and construction of the cold box and HX was for an eventual rate of 2400 L/hr. Calculated thermodynamic efficiency was 0.17 of Carnot. In 1978 an increase to 1200 L/hr was made by adding a rotary screw compressor system (Mycom, 250 g/s, 1 atm to 20 atm) and adapting the turbo expanders to the higher flow. The up-graded plant went operational in early 1980 and up to that time had more than 50,000 hours operation.

A third Helix-Sulzer bulk plant was built for the U. S. Bureau of Mines, at Excell, Texas in 1978. Capacity was 500 L/hr with LN pre-cooling and 700 kW power input.

The fourth and largest, most efficient plant is the Helix-Sulzer unit built in 1978-79 at Bushton, Kansas for the Linde Division of Union Carbide Corp.[30,31] With 1300 kW power input, capacity is 1400 L/hr without LN pre-cooling. Three oil-bearing turbines, Sulzer T33,

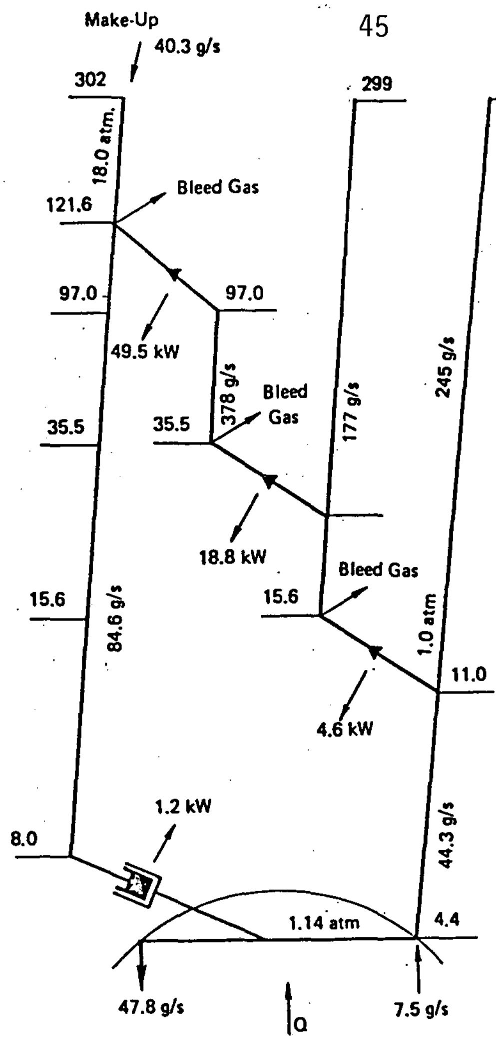


Figure 13. 1400 L/hr Liquefier T-S diagram. Ref. 31

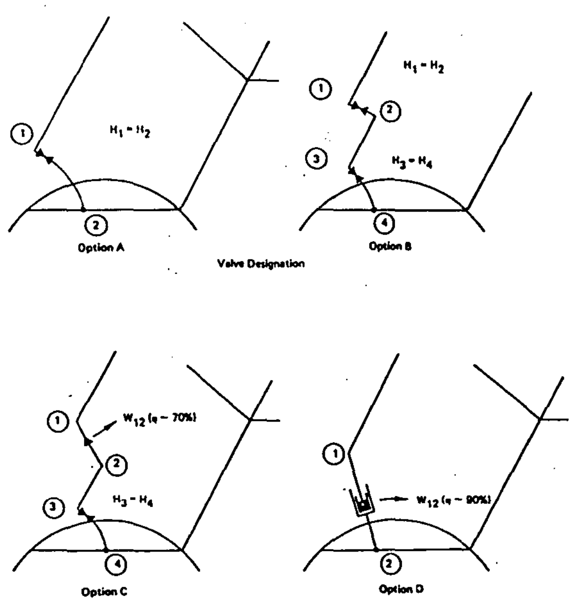


Figure 14. Options for entry to 2ϕ region.

Ref. 31

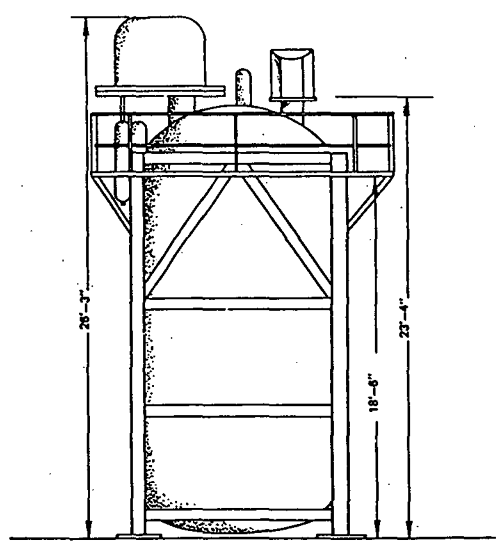


Figure 15. Cold box

Ref. 31

T22 and T11, operate in series across three pressure levels with ratios of 2.6 from 18 to 1 atm. A fourth and final stage of refrigeration is provided by a reciprocating "wet" expander operating in the two-phase region with about 90 percent efficiency. This engine has two 3.6 inch diameter phenolic plastic pistons with stroke adjustable to 3 inch, sealed with lubricated O-rings in cylinders at the warm end (300 K). Engine design cut-off is approximate 60 percent and pistons operate in parallel at 180° phase angle.

The thermal cycle is designed with dual pressure loops to increase efficiency and match the optimum pressure ratios of the turbines. Expander pressures are such that the first stage compressor mass flow is low and thus the compressor is of reasonable size and reduced capital investment. The bulk of the refrigeration is thus produced by the second and third stage compressors at the higher pressure level.

Six Sullair oil-flooded rotary screw compressors raise the pressure from 1.0 atm to 18 atm in three stages, two compressors in parallel per stage. For reduced loss, water inter and after-cooling is installed, together with multi-stage final oil removal mist separators, charcoal adsorbers and final filter.

The Sullair compressors are listed.[30] The first number 32, 25, 20 defines the rotor diameter in centimeters and the LB and SB refer to short and long booster or first stage. The mass flow and pressure level in each stage is variable with compressor slide valves.

Stage	Model	Qty	Displacement CFM	Outlet atm. (est.)
First	B32LB-500 HP	1	3354	2.6
	C20SB-100 HP	1	597	
Second	C25LA-500 HP	1	1710	6.8
	C20LA-250 HP	1	881	
Third	C20SA-400 HP	2	597	18

This plant has a thermodynamic efficiency approaching 25 percent of Carnot, and is one of the most efficient bulk helium liquefiers built. A comprehensive report by Hubbell describes the cycle optimizaton for steady state mode; reduced capacities when demand is down and trailer fill mode when cold gas return helps HX imbalance. Economics required the traditional LN pre-cooling be replaced with a turbine.

The largest single helium liquefier in the world is the Fermilab Central Helium Liquefier (CHL) with capacity of 4875 L/hr and Carnot efficiency of 22 percent. Calculated inputs are 2500 kW compressor power and 2700 L/hr LN pre-cooling. The cold box was delivered in 1978 and first operation was in early 1980. This machine, while designed, optimized and operated as a liquefier is really part of the Fermilab LHe refrigerator system and is described in more detail in the section following.

III c. Large Refrigerators

Many helium systems exist world-wide in Asia, Europe and the Americas. The small units usually employ the Gifford-MacMahon or Stirling cycle. Above a few liters per hour or 50 W the Claude cycle is usually employed. The University of California Low Temperature Laboratory in Berkeley, which initially used LN, LH_2 and J-T cascade, has for many years used a two-stage Phillips Stirling cycle refrigerator to produce LH_2 or LHe (60 L/hr). Presently plants of 100 L/hr plus or 200-1000 W are common. But even "large", defined here as above 1500 W, is small compared to industrial plants for LNG or air which can range into 100,000 HP.

1.0. The 1500W Refrigerator

In early 1974, a meeting was held at LBL with fabricators, APCI, Helix, Sulzer, Rotoflow and potential users; BNL, EPRI, Fermilab, LBL, LLNL and ORNL. The purpose was to standardize an optimum refrigerator in the 1500 W range for the next generation of research apparatus.

Fermilab had preliminary plans to use 20 such units around their ring accelerator with a 1000 magnets. LBL needed one for a small prototype accelerator "ESCAR" (Experimental Superconducting Accelerator Ring) with 56 magnets. ORNL had need for their fusion energy magnet test program.

Fermilab (1975) initiated the purchase of a turn-key system and LBL joined in the purchase of a cold box only. LBL purchased and developed their screw compressors separately. The specifications were stringent and included:

I. Performance

a. LBL - 1450 W plus 3 g/s liquefaction at 4.5 K

3 g/s ≃ 90 L/hr = additional 400 W. This flow is used for magnet power lead cooling.

b. Fermilab - 950W plus 1.7 g/s at 4.5 K plus 1250 W at 21 K for shield cooling.

c. 360 L/hr as liquefier.

II. Rotary equipment only for expansion and compression.

III. Complete control by a single operator from a remote location.

IV. Minimum and fixed limits for electrical power, LN feed, maintenance and manpower.

V. Rigid time limits on cold-box cool-down and warm-up for maximum availability.

The Helix-Sulzer design used six Trane Al HX (one for rapid cool-down only) and Sulzer gas bearing turbines. The Claude cycle is dual pressure, which again reduces the size of the first stage compressor. The cycle is selected for best match of the turbo-expander ratios and the compressor ratios. LN pre-cooling is used with a final stage of J-T cooling. The design is optimized as a refrigerator and the lower turbine mass flow sizing is a compromise for the various modes. The LBL mode required a J-T valve in parallel to handle 20 percent of the flow.

The two compressors are Sullair 250 mm dia. oil-lubed screws with first stage 1 to 3 atm, 110 g/s and 200 HP, second stage 3 to 15-20 atm, 250 g/s and 800 HP (LBL) and 1000 HP (FNAL).

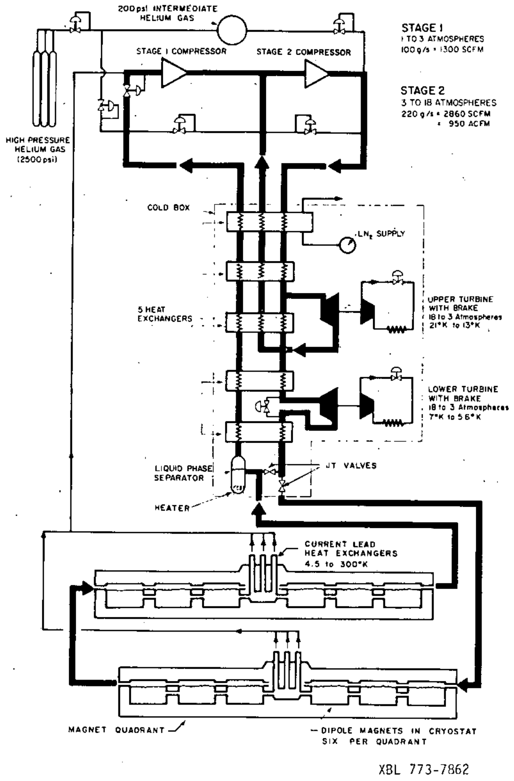


Photo. 5. (XBL 773-7862). Schematic of the 1500-W refrigerator connected to 12 dipole magnets in series two-phase flow with pool-boiling. Refrigerator design is dual pressure cycle. LBL photo.

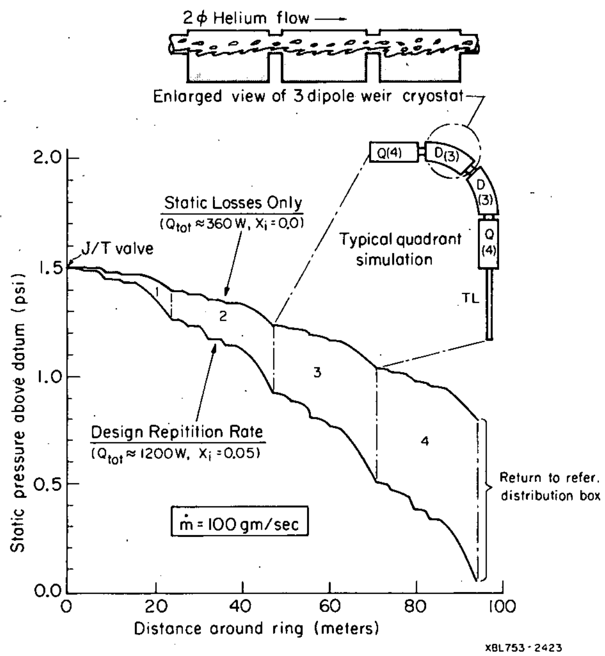


Photo. 6. (XBL 753-2423). Predicted pressure drop for two-phase 100 g/s mass flow in 56 magnet accelerator ring. LBL ESCAR photo.

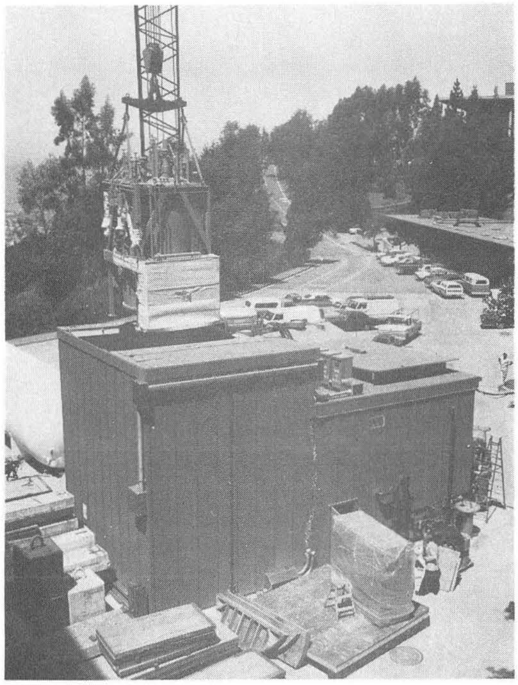


Photo. 7. (CBB 776-5485). Installing 1500-W cold box in LBL Bldg. 56. Compressors are in right hand section of building, control console and room in foreground. LBL photo.

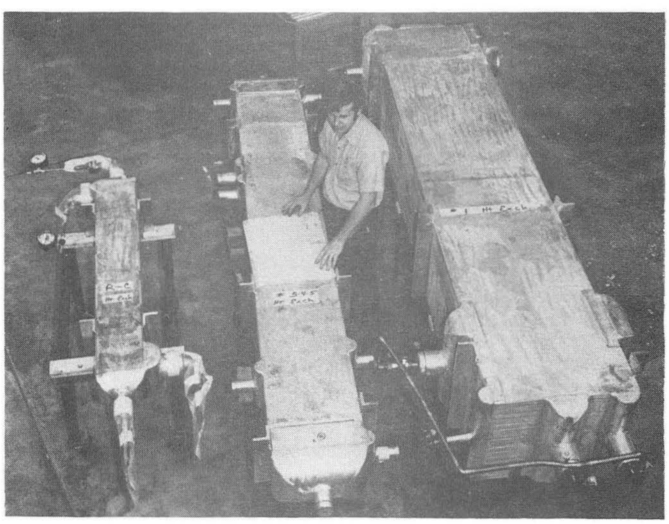


Photo. 8. (XBB 781-404). Brazed aluminum plate-fin HX for 1500-W cold box. L to R; rapid cooldown HX; No. 3,4,5 HX and No. 1 HX. Photo courtesy of CTI, now Koch Process Systems. (KPS)

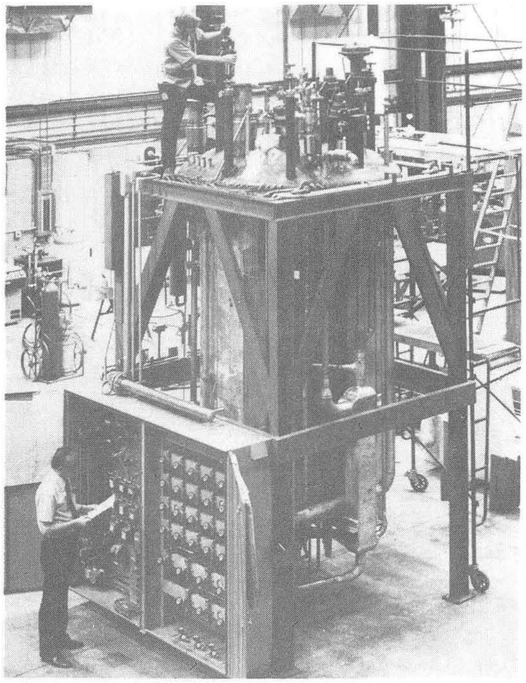


Photo. 9. (XBB 781-408). 1500-W cold box during assembly. Rapid cooldown HX in lower right foreground, 25 kW rapid warm-up electric heater on top of valve and transmitter control box. Photo courtesy of CTI, now Koch Process Systems. (KPS)

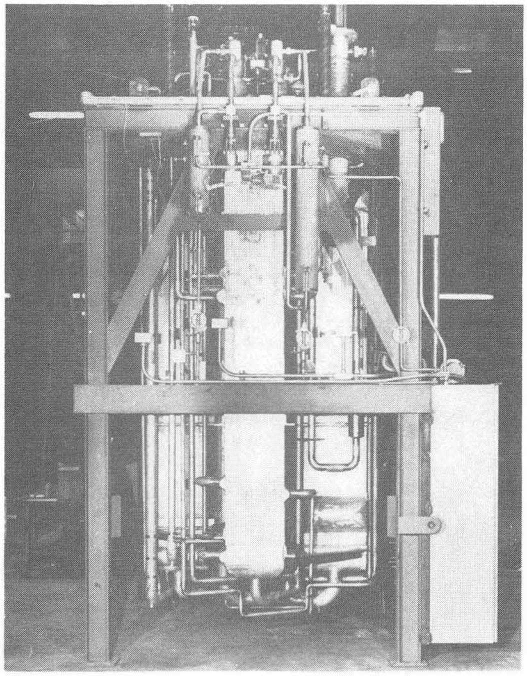


Photo. 10. (XBB 781-407). Another view of 1500-W cold box. Water HX for two turbo-expanders are shown at top center with brake control valves.
Photo courtesy CTI, now KPS.

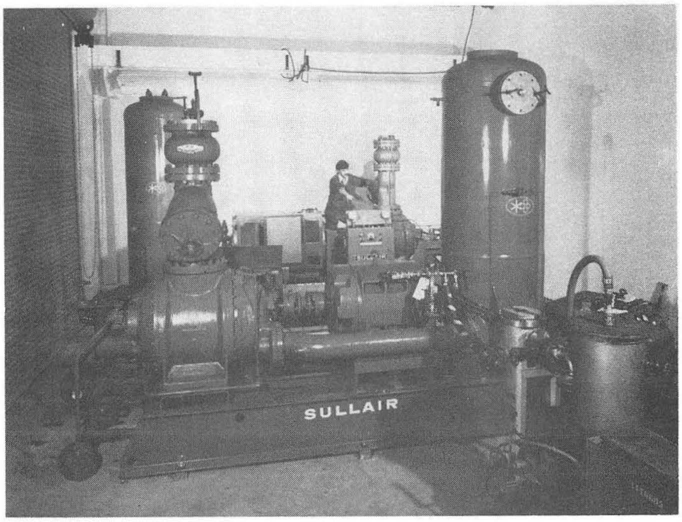
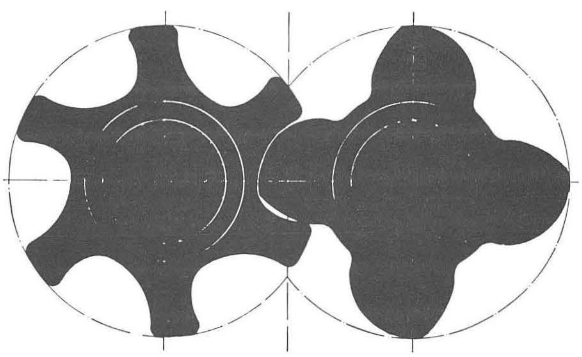


Photo. 11. (CBB 769-8182). Oil-flooded screw compressors for 1500-W plant. First stage 250 mm dia., 1700 SCFM 100 g/s, one to three atm., 200 HP unit in foreground. Second stage 250 mm, 250 g/s 2.5 atm to 16 atm, 800 HP in rear. LBL photo.

Schraubenverdichter
Screw Compressors
Compresseurs Helicoïdaux
Compresores Helicoidales



XBL 781-6864

Photo. 12. (XBL 781-6864). Screw compressor profile shows 4-lobe male driver and 6 lobe female rotor with high efficiency asymmetrical profile. Photo courtesy of GHH-FRG.

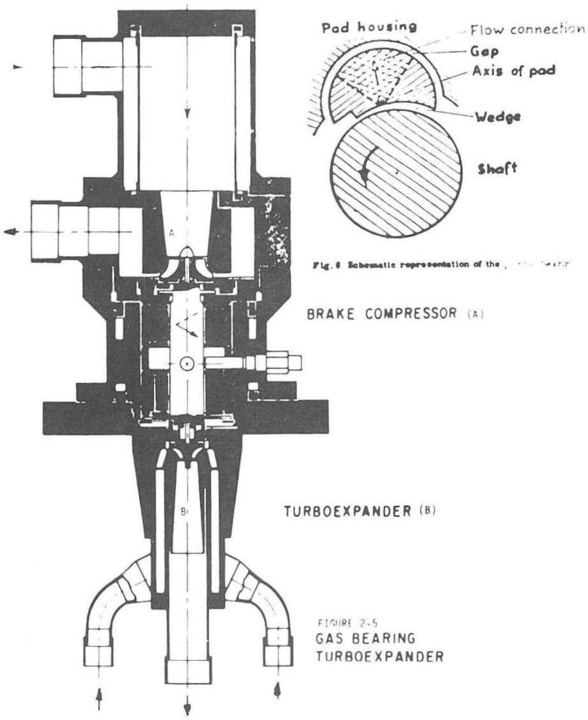


Photo. 13. (XBL 781-6863). Gas bearing turbo expander used on 1500-W. Cold circuit below, gas brake above. Hydro-dynamic or self-pumping gas radial and thrust bearings. Photo courtesy Sulzer Bros.

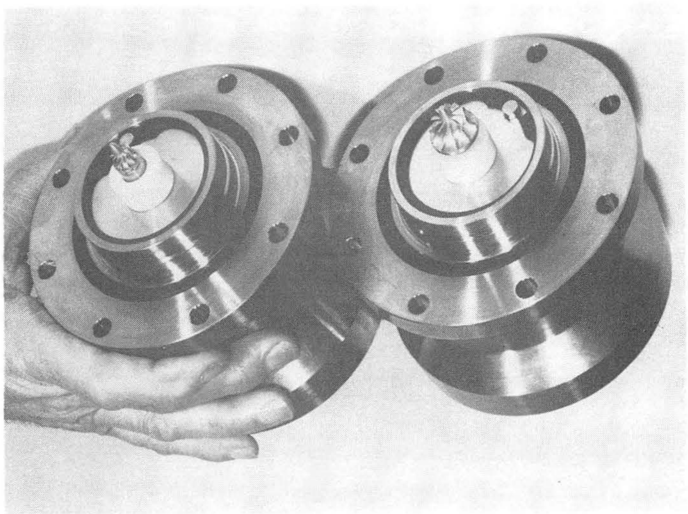


Photo. 14. (CBB 770-10437). Cold expander end of two turbines, 12 mm dia. and 20 mm dia., used on 1500-W. LBL photo.

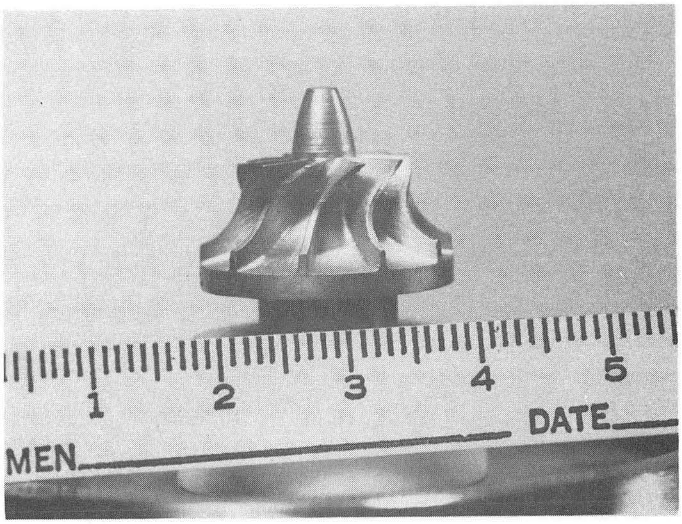


Photo. 15. (XBB 790-13993). Close-up of 20 mm dia. cold expander wheel. LBL photo.

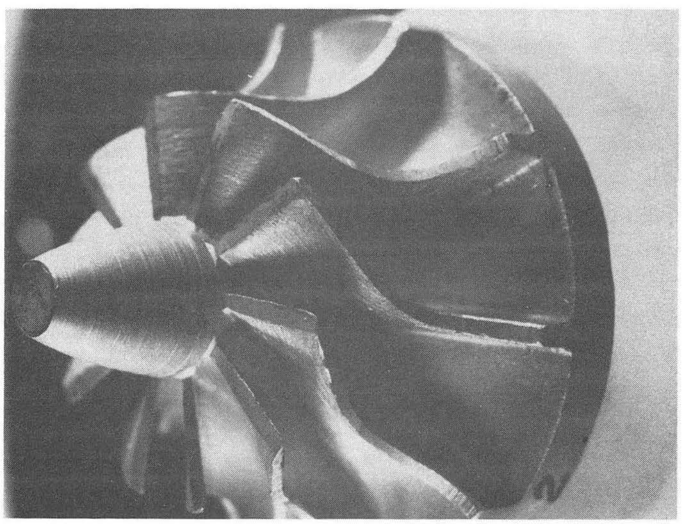


Photo. 16. (CBB 801-343). Closer-up of 20 mm wheel. LBL photo.

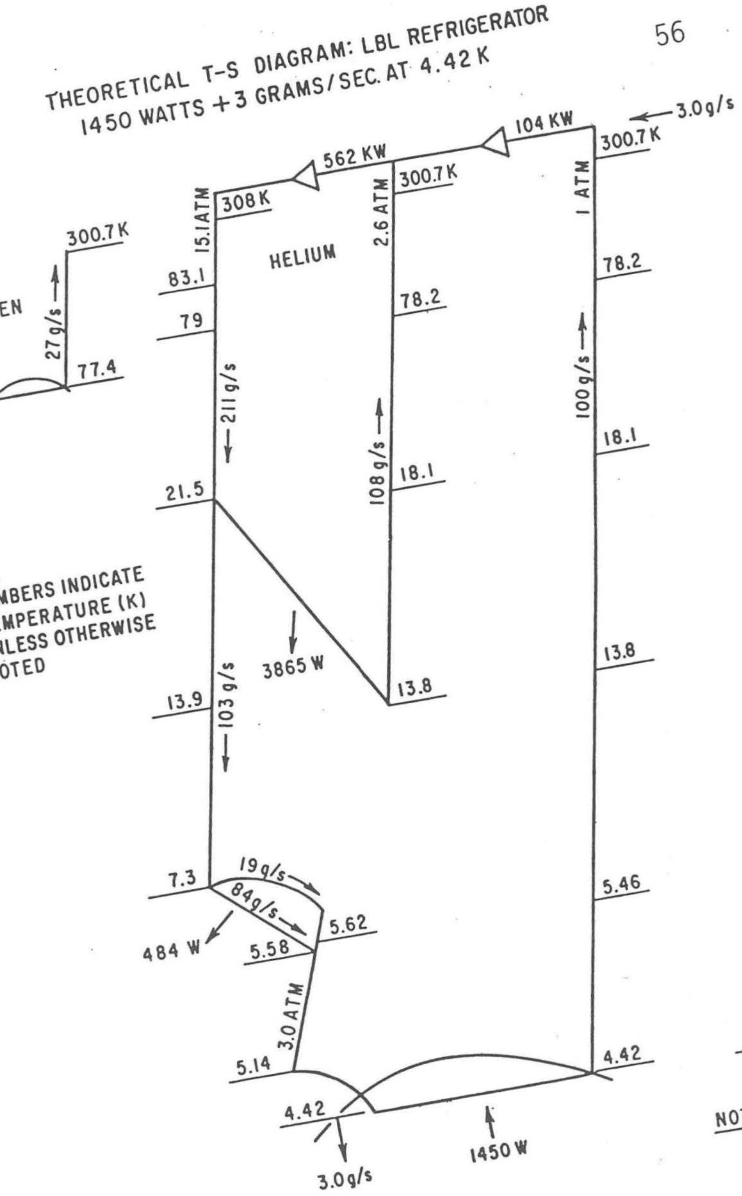


Figure 16. T-S Diagram LBL Refrigerator.

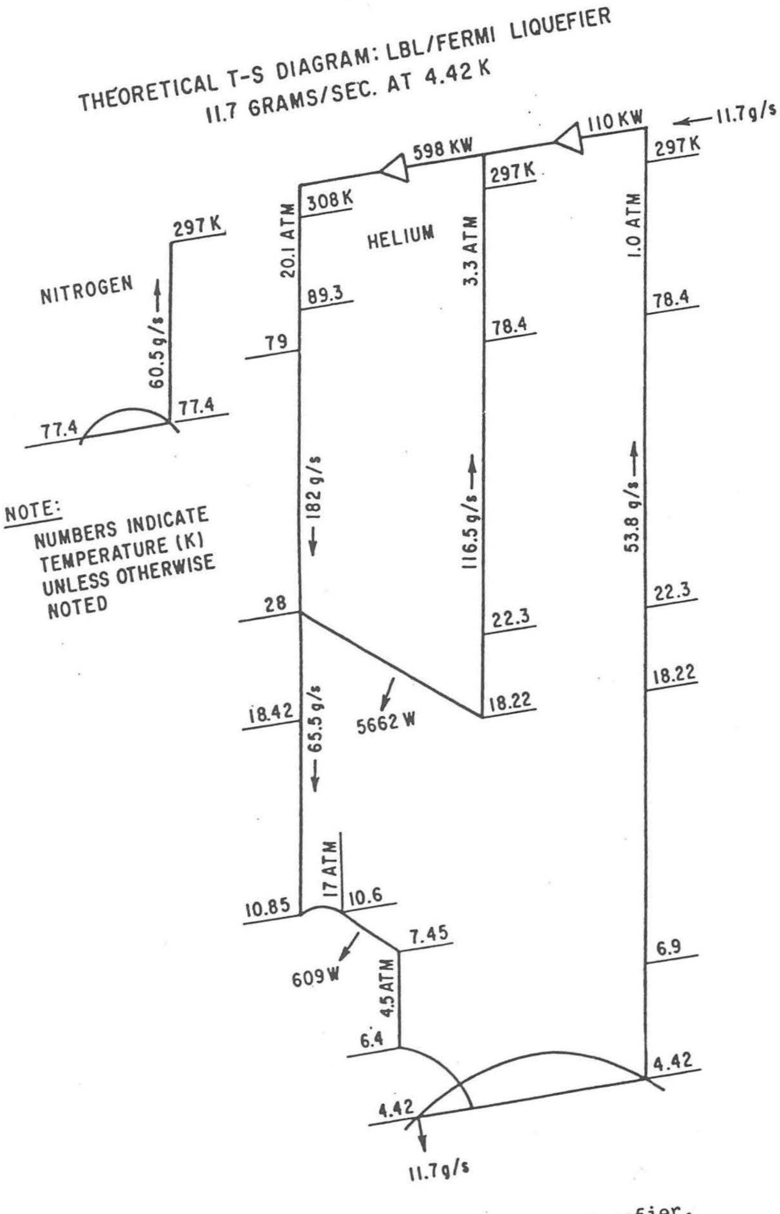


Figure 17. LBL-Fermi Liquefier.

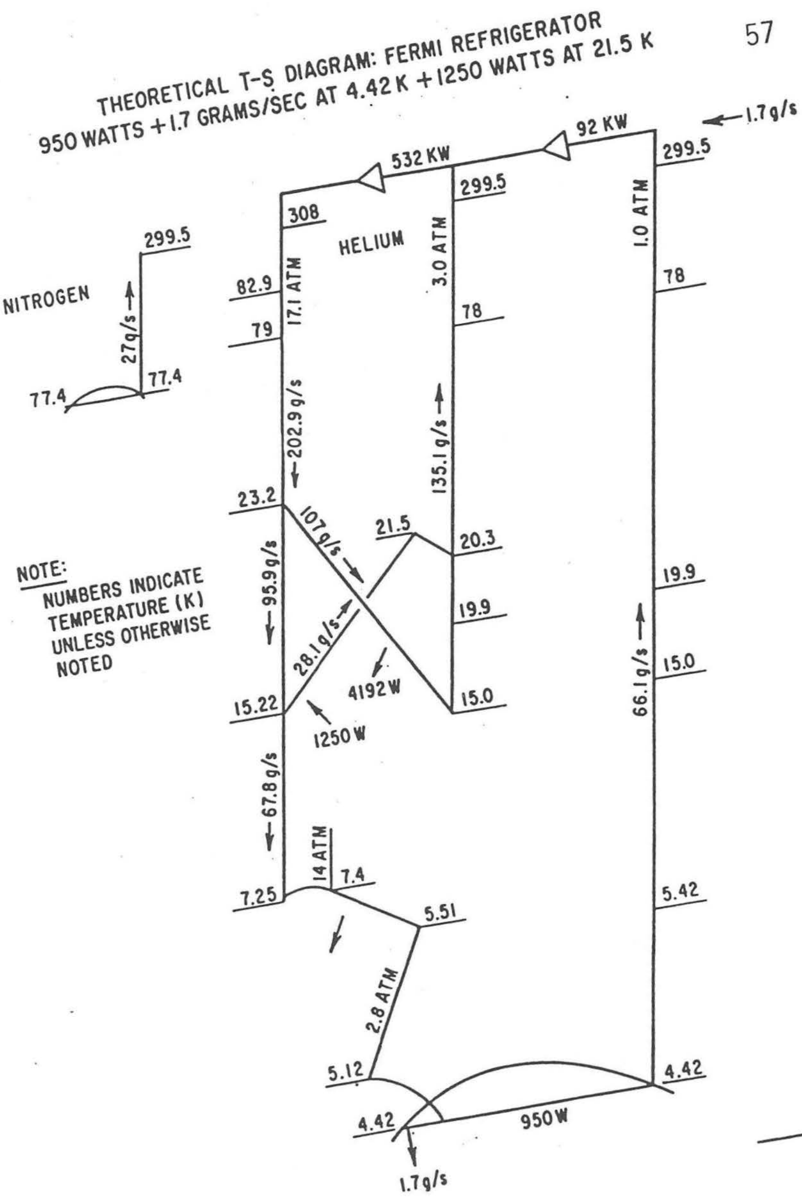


Figure 18. T-S Diagram–Fermi Refrigerator

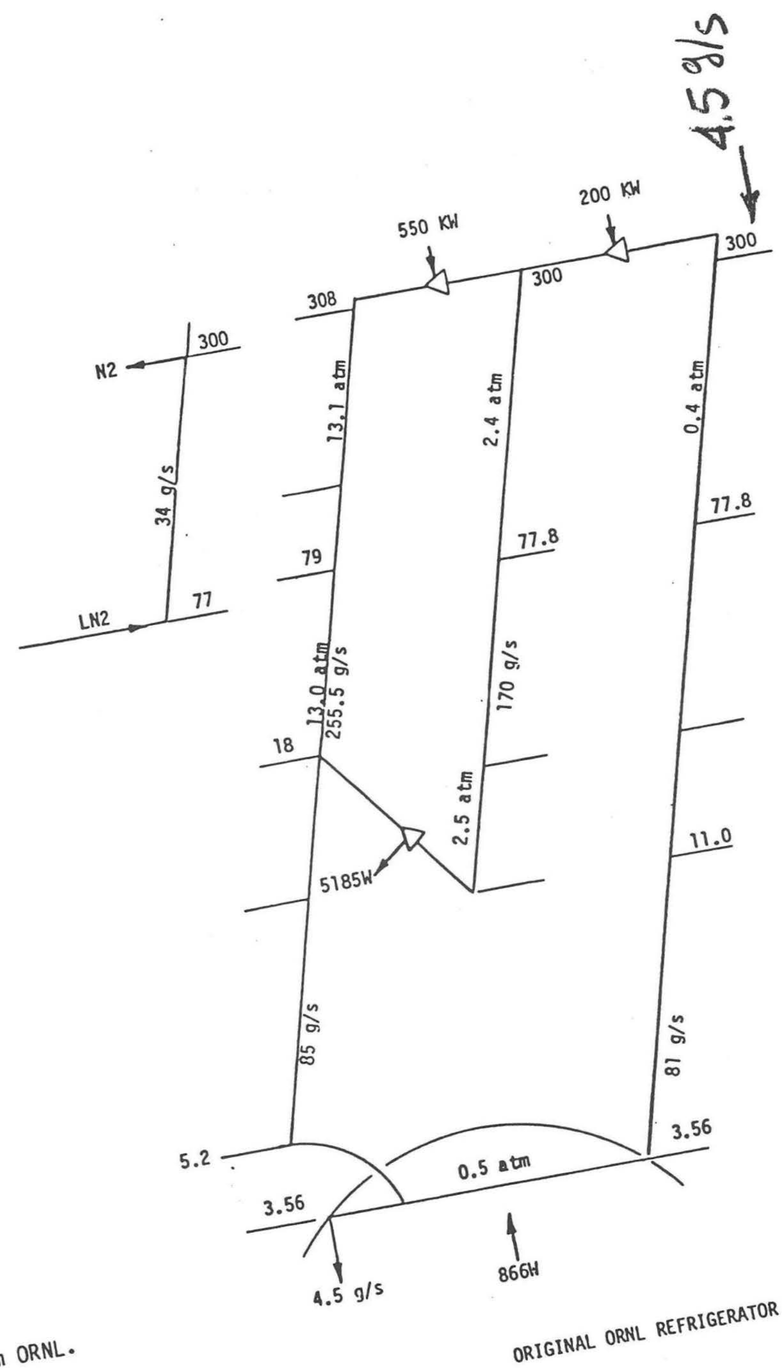


Figure 19. Original T-S diagram ORNL.

Ref. 33

1 a. Lawrence Berkeley Laboratory 1500 W[23]

The two compressors were delivered in September, 1976, and their system tests were completed in May 1977. The LBL cold box was delivered June 1977 and acceptance testing completed October 1977 with loads up to 1900 W. Testing with the ESCAR system components continued through 1978. The heat loads comprised twelve dipole magnets and 400 ft (120 m) of 2 inch (5cm) dia. transfer line. This line had a 2 inch inner (LHe) and a 3.5 inch outer diameter (vacuum) single pass with 40 layers of aluminized, perforated, crinkled 1/4 mil mylar. Measured heat loads on short 10 and 20 ft lengths were 0.25 W/m.

The magnets were cooled by pool-boiling in series connected cryostats. The cryostats are filled by a gravitational separation of liquid from the coolant stream (mist flow regime). The large flow cross-section of the magnet cryostats gives low-velocity and enhanced liquid-vapor separation. Location of the 2" dia. transfer line just above the magnet at the cryostat top limits the liquid fill level. The magnets cool down evenly with the series flow and fill sequentially. Very little liquid collected until the adjacent upstream magnet reached the spill over point. This simple weir system demonstrated the stability and effectiveness of the liquid-vapor separation.

1 b. Fermilab 1500 W Delivered Fall 1977[32]

The main ring magnet cooling design developed in another direction, so the 1500 W refer was installed for the magnet test facility. Each of the Fermi magnets, some 1000 dipoles and quadrupoles, plus spool piece correction coils and external beam line magnets requires complete cold testing, both for magnetic field accuracy and vacuum integrity. For the test facility the 1500 W cold box feeds a 10,000 L LHe dewar at slight pressure, which in turn feeds two distribution manifolds with LHe. These manifolds have valving to permit warm magnets to be connected, cooled and tested at 4.5 K under steady-state; then warmed-up and removed. Each SC magnet is tested before installation in the tunnel ring. By late 1981, the plant had accumulated 22,000 hours and in late 1982 the final magnets had been tested and were being installed in the accelerator tunnel.[32]

1 c. <u>Oak Ridge National Laboratory 1500 W</u>[33]

The ORNL plant was ordered 1976 and delivered in 1977 for the large coil task (LCT) program for testing six different prototype SC magnet coils for the next generation of fusion energy Tokamaks. These single "D" shaped coils have 2.5 m x 3.5 m bore dimensions, with peak fields of 8 Tesla and weights of 47 ton. Three sources (General Dynamics/Convair, General Electric, Japan) will supply magnets using pool boiling while three (Euratom, Switzerland and Westinghouse) will require supercritical forced flow.

As initially specified the cold box would supply 866 W at 3.55 K and 4.5 g/s of liquid for power leads. The system was similar to the LBL-FNAL - 1500 W with two pressure levels, but for economics used a single turboexpander, Sulzer gas bearing TGL-32 mm. It also used Sullair compressors but because of the 3.56 K return temperature, uses a B32LB, 3354 CFM at 1 atm, (less at 0.4 atm) a machine of twice the displacement of the compressors used at LBL and FNAL. First stage motor is 400 HP and second stage 1000 HP with power input of 750 kW.

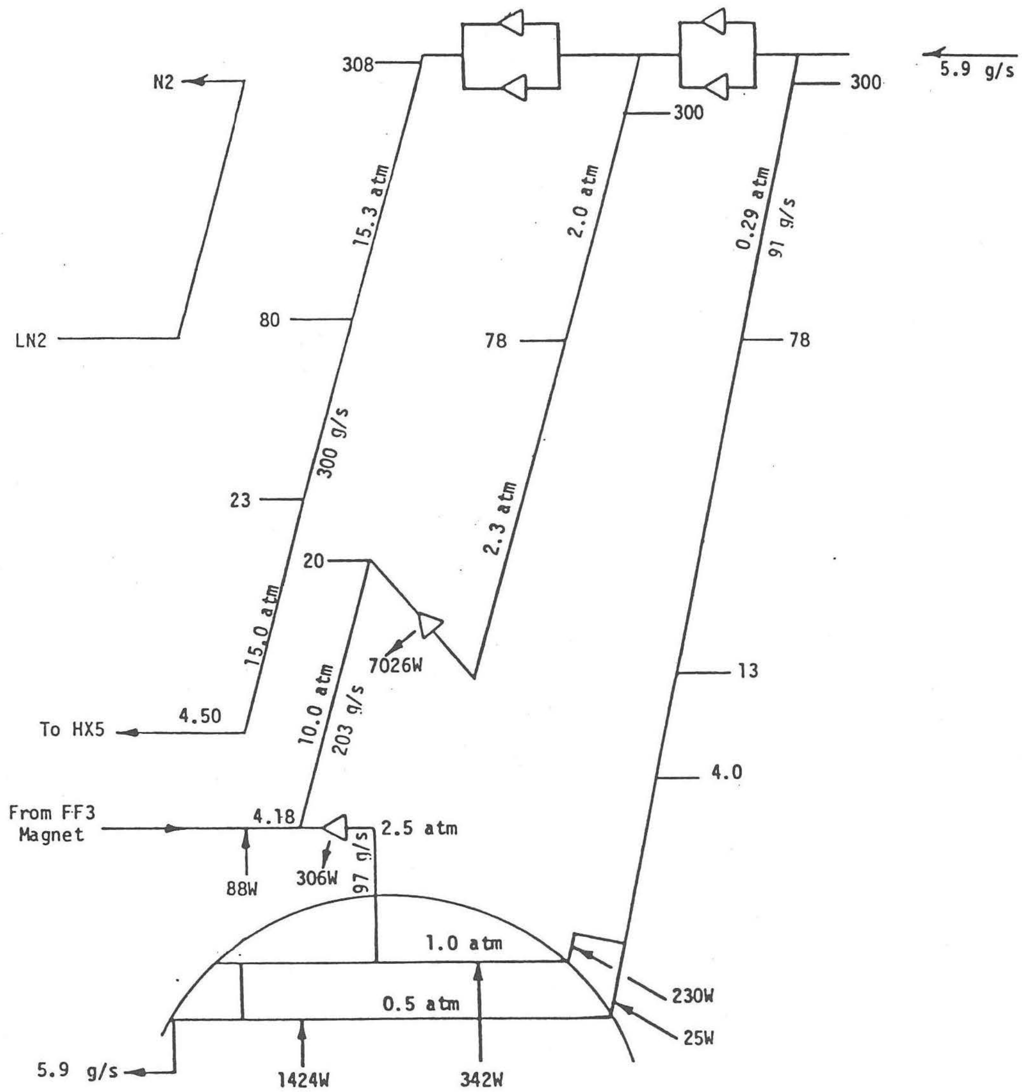
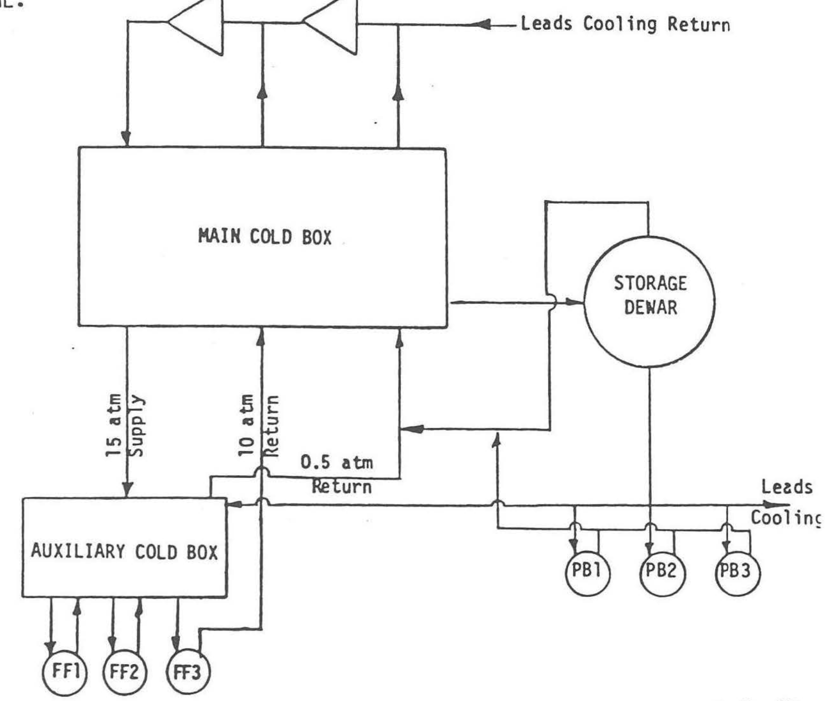


Figure 20. Modified T-S diagram for ORNL.

Ref. 33



Ref. 33

Figure 21. Simplified flow diagram modified ORNL Refrigerator.

After the six magnet designs for the LCTF were refined, it became apparent the existing plant could only supply half the requirements. There are four operating modes defined, liquefaction only (magnets warm) and three modes while holding the total system cold; liquefaction, one pool boiling test and one force cooled magnet test. (300 g/s is needed for the force cooled magnets).

A unique cost effective solution that did not require changing the cold box HX, added an external HX 5, a turbine and another set of compressors, 400 HP first stage, and 1000 HP second stage. Expected equivalent performance is about 2800 W at 4.5 K.[33]

2.0. Lawrence Livermore National Laboratory 3000 W[34]

The Mirror Fusion Test Facility (MFTF) first phase "A" was completed in early 1982 with a successful test-run of 1000 hours of operation. A large 11 m dia. x 19 m long stainless steel vacuum tank lined with 1100 m^2 LHe cryopanels encloses a 340 ton SC magnet. The "Yin-Yang" pair of magnets consists of two interlocked "C" shaped magnets such that the magnetic field increases everywhere from the center. This geometry forms a magnetic bottle to confine the hot plasma.

The cryosystem was supplied by CVI, Inc. of Columbus, Ohio under a turn-key contract. It consists of a 150,000-L, 100 kW LN open loop system, the cryopanels, the LHe distribution system to magnet and panels, and the 3000 W plus at 4.35 K, LHe refrigerator.

LHe coolant flow to the heat loads goes to a 25,000-L storage dewar and/or two 4400-L supply dewars. The large dewar provides standby coolant for refrigerator outage and storage for liquid from the magnet or panels. The smaller two dewars are ballast and phase separators. Liquid is supplied by gravity; the height of the convective loop is 20 m from the bottom of both magnet and panels to the supply dewars. Such a system can have a high degree of reliability.

Tabulated LHe heat loads are budgeted as follows:

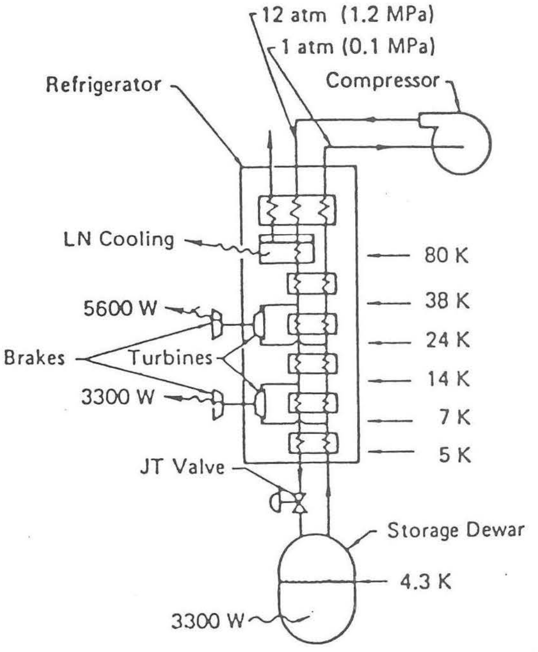
Item	Watts
Magnet	565
Cryopanels	1450
External piping	80
Dewars, supply and storage	100
External cryopumps	15
Valves and valve box	100
Reserve capacity	765
Total refrigeration load	3075 W
plus 2.5 g/s (75 L/hr)	225 W equivalent
liquefaction for power leads	3300 W total

The refrigerator, with capacity of 3300 W or 600 L/hr at 4.35 K, consists of a Claude cycle using seven counterflow Al HX and two turbines. The 3.5 m diameter by 6 m high cold box was manufactured by CVI Corp., Columbus, Ohio. The two L'Air Liquide turbines have hydrostatic (gas injected) bearings and impeller wheel diameters of about 40 mm. The warm or upper turbine expands 73 g/s, from 12 atm, 38 K inlet to 1 atm, 24 K, outlet with 0.63 efficiency and 5.6 kW out. The lower cold turbine expands 155 g/s from 12 atm, 13 K inlet to 1 atm, 7 K outlet with 0.65 efficiency and 3.3 kW out. Note this type of turbine can take a pressure ratio of 12 to 1, but efficiency is decreasing. A 12 to 1 ratio approaches maximum for efficient work extraction, 18 to 1 might require two stages.

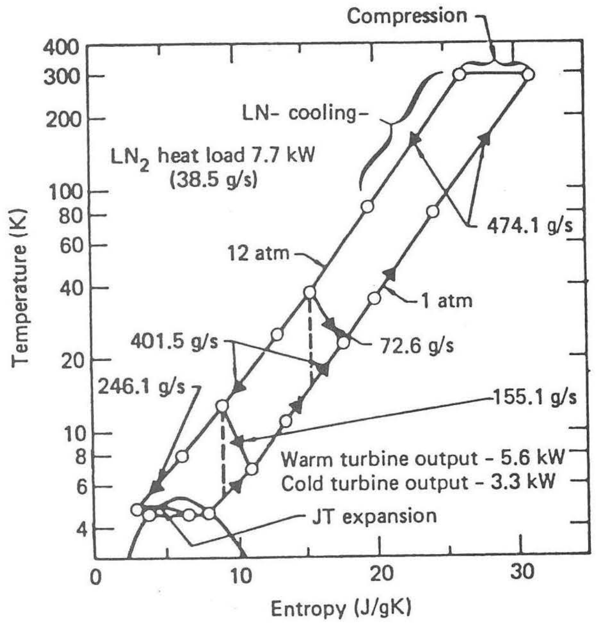
There are three oil-lubed rotary screw compressors, two first stage and one second stage. Mycom Corp., Los Angeles, a subsidiary of (Mayekawa Mfg. Co., Tokyo, Japan), packaged them in a skid 7.3 m long by 4.0 m wide by 3.4 m high with intercoolers, oil separators, control valves and a total by 2000 HP (1500 kW) electric motors. Design operating output is 475 g/s at 12.5 atm with 1 atm suction.

Because of promising fusion physics results with a smaller test model, funding approval was granted for MFTF-B. This will add another vacuum tank 19 m long containing a Yin-Yang magnet as in MFTF-A and a central cell 8 m dia. x 22 m long, for a total length of 64 m. Twenty magnet coils will be added to MFTF-A for a total of 22, with 14 central cell solenoid coils of 5 m dia. that confine the plasma to a cylindrical shape tied or plugged at each end by a Yin-Yang A and B, similar to a sausage or wurst.

The upgrade "B" will add 8000 W at 4.2 K plus 15 g/s LHe (dual pressure Claude) refrigeration, or 1700 L/hr; a 60,000-L storage and a 10,000-L supply LHe dewar. Three first stage 400 HP Howden screw compressors, and two second stage 2250 HP, all 320 mm dia by L/D 1.9 ratio will be used. Two Air Liquide and one Rotoflow expander turbines will be used. A 500 kW LN re-liquefier is also planned. The facility, when completed will be one of the world's largest, equal to 11 kW at 4.5 K and 0.18 Carnot.



Schematic of the refrigerator-compressor system



Temperature-entropy diagram for the refrigerator-compressor system

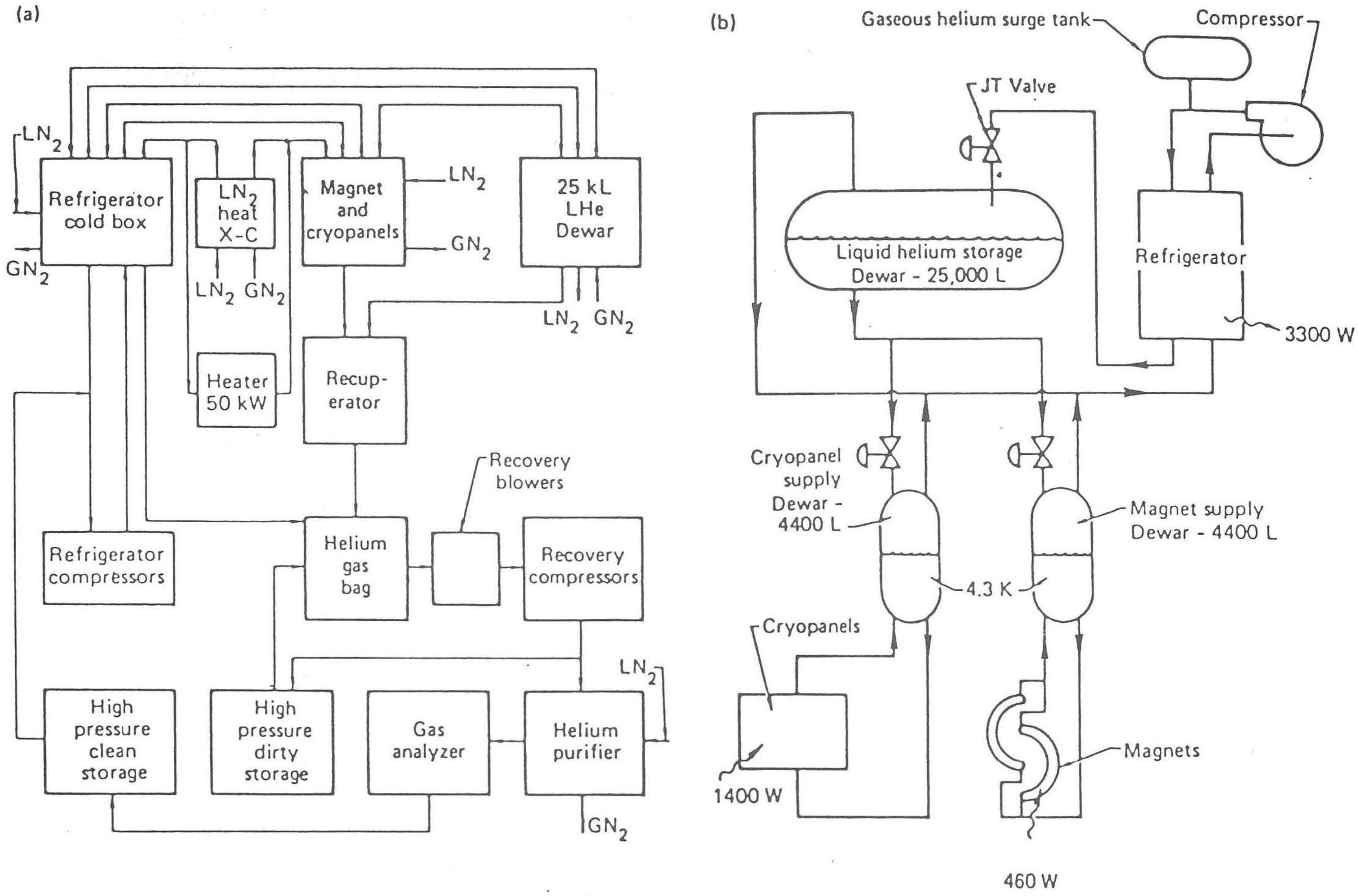


Figure 22. Livermore (LLNL) MFTF System 3000-W. Ref. 34

Photo. 17. (CBB 831-892). Coil box assembly of superconducting Yin-Yang magnet for Mirror Fusion Test Facility (MFTF) LLNL photo.

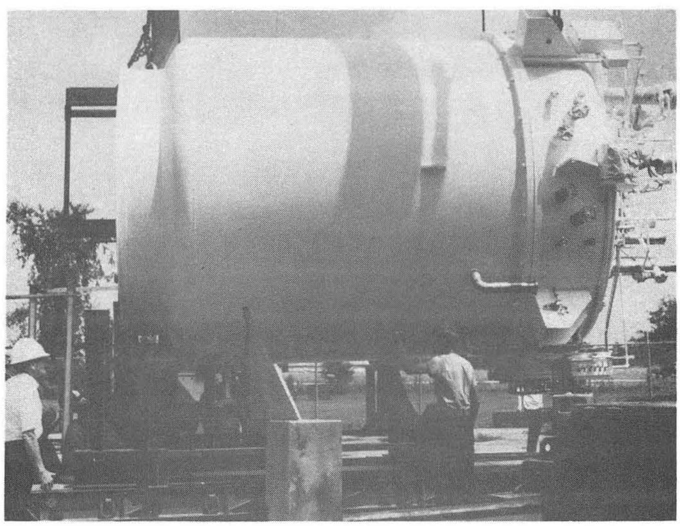


Photo. 18. (CBB 831-495). The 3000-W CVI cold box for LLNL MFTF-A on trailer. LLNL photo.

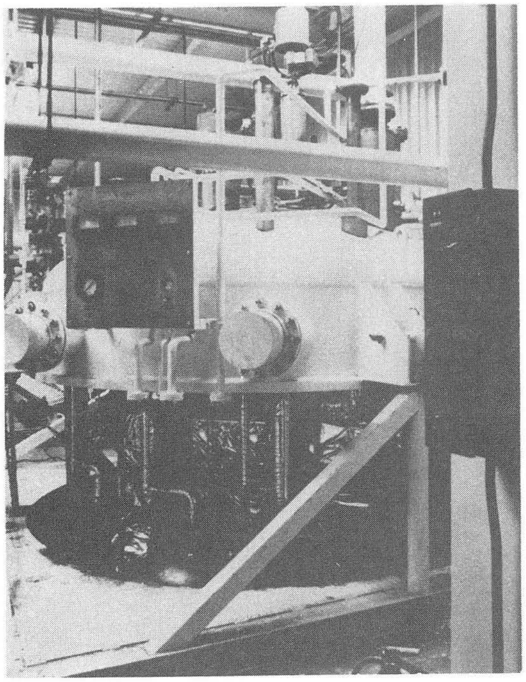


Photo. 19. (CBB 831-493). Top section of 3000-W cold box installed. LLNL photo.

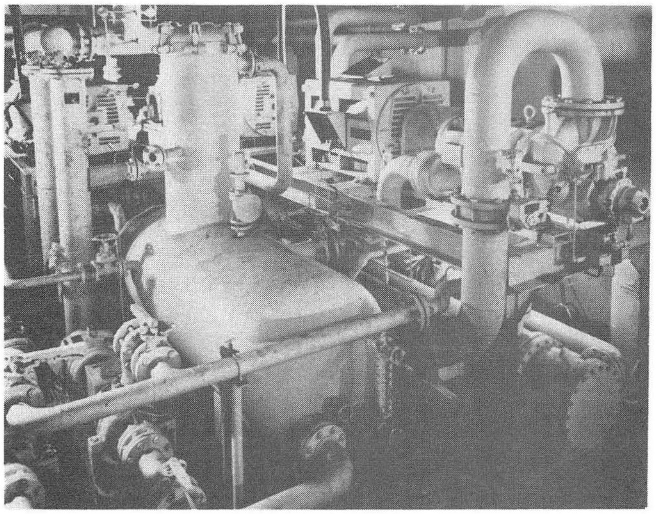


Photo. 20. (CBB 831-491). Screw compressor installation for 3000-W cold Box. LLNL photo.

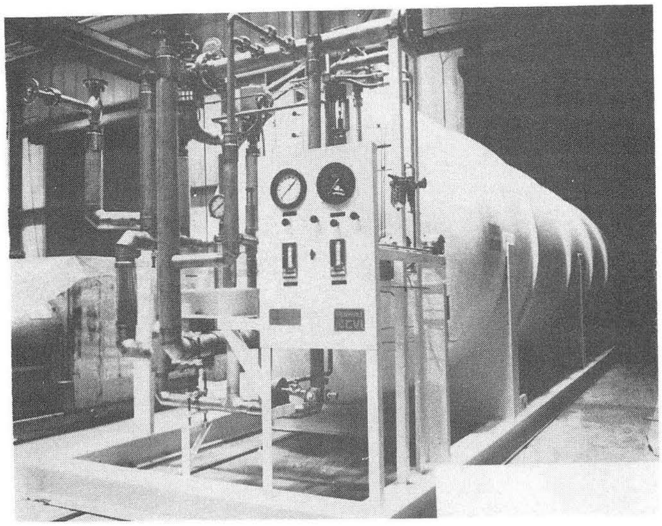


Photo. 21. (CBB 831-489). LHe supply dewar for MFTF-A system. LLNL photo.

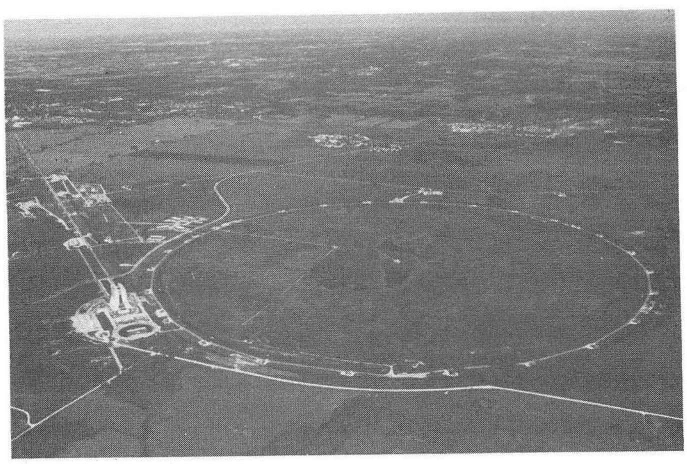


Photo. 22. (CBB 831-471). Aerial view of Fermilab with 2 km dia. ring, 24 satellite stations; at left booster ring and target areas. Photo courtesy Fermilab.

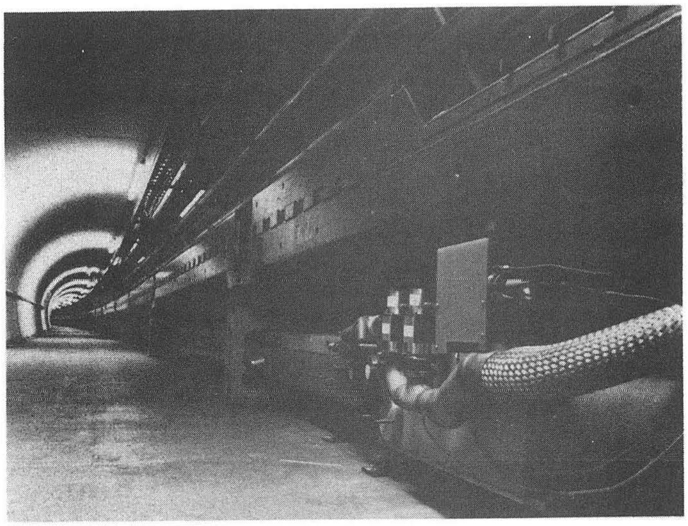


Photo. 23. (CBB 831-469). Accelerator main ring tunnel with conventional copper-iron magnets above and superconducting magnets below. Photo courtesy Fermilab.

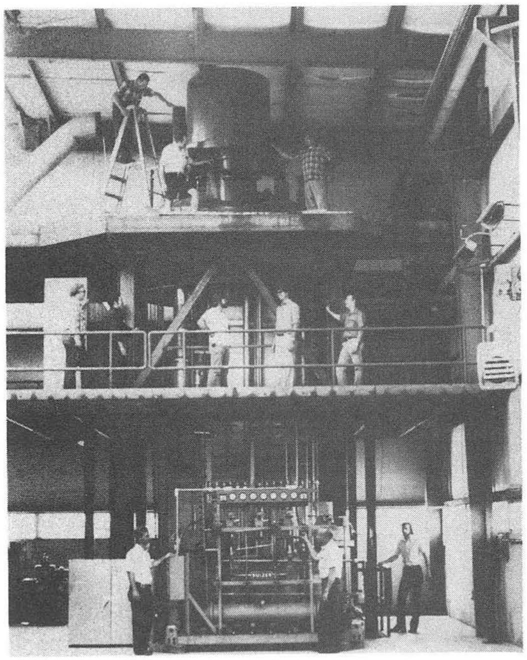


Photo. 24. (XBB 831-448). Fermilab Central Helium Liquefier (CHL) 4000 L/hr. Oil bearing turbine pods on top level, and oil sub-system on floor. Photo courtesy Fermilab.

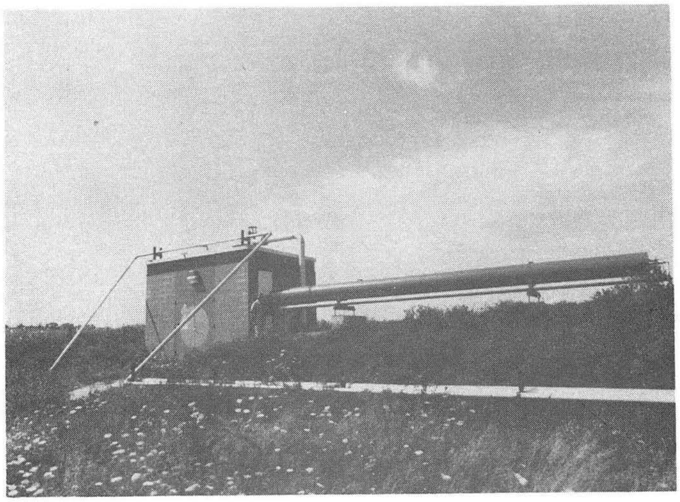


Photo. 25. (CBB 831-467). A satellite station at Fermilab, with the long (~ 10 m) cold box version extending out of the building. Photo courtesy Fermilab.

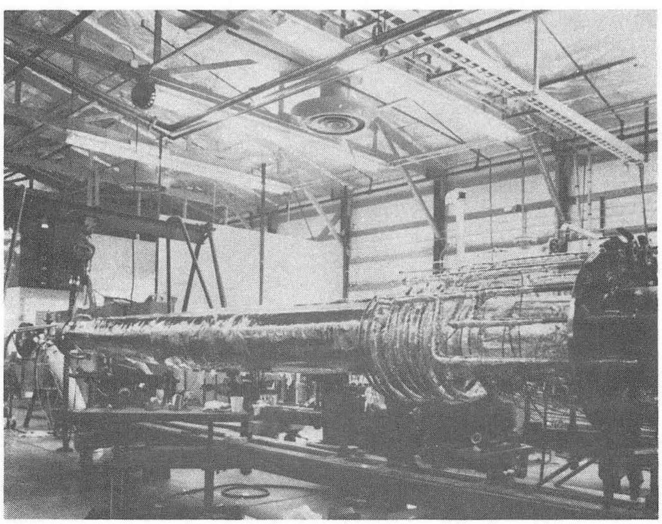


Photo. 26. (CBB 831-481). Satellite long version (~ 10 m) cold box in construction. Photo courtesy Fermilab.

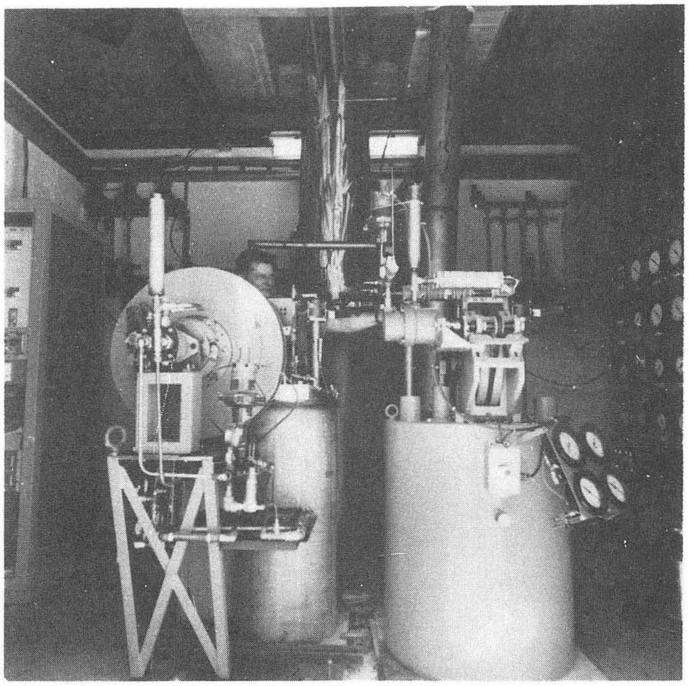


Photo. 27. (CBB 831-477). Satellite station with both dry and wet expander engines. Photo courtesy Fermilab.

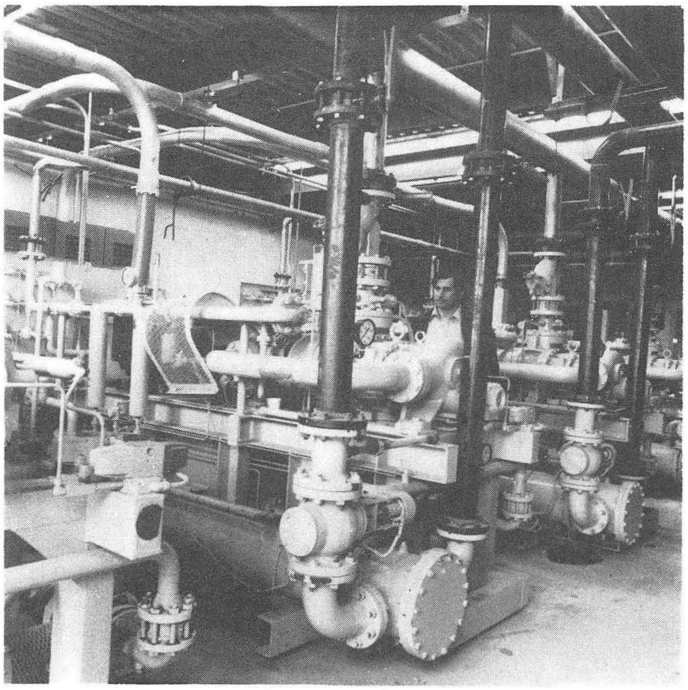


Photo. 28. (CBB 831-475). Four screw compressors installed for satellite refrigerator sector. Photo courtesy Fermilab. (FNAL)

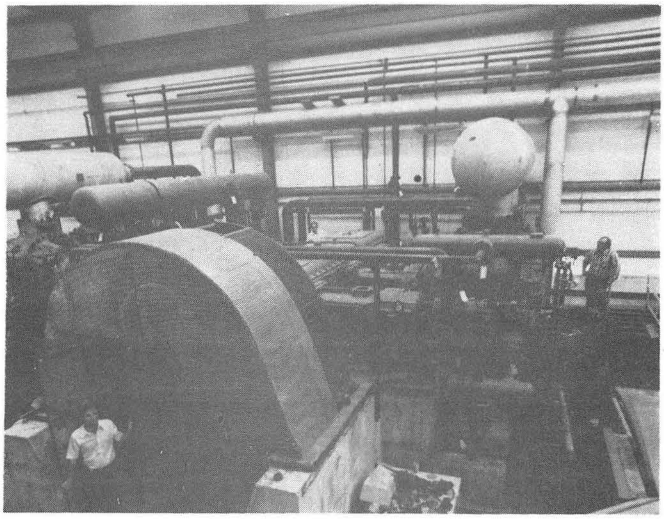


Photo. 29. (CBB 831-483). Installation of one of CHL horizontally opposed six-cylinder helium compressors. Photo courtesy FNAL.

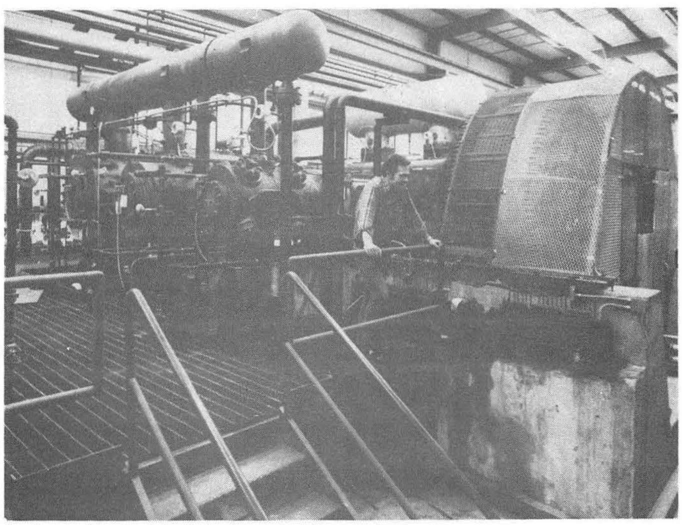


Photo. 30. (CBB 831-473). Another view of CHL helium compressor. Photo courtesy FNAL.

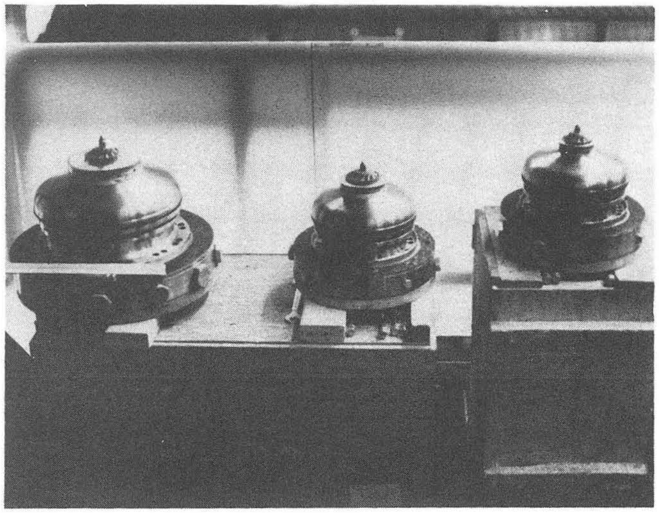


Photo. 31. (CBB 831-465). The three Sulzer oil bearing turbine cartridges used on the CHL. Photo courtesy FNAL.

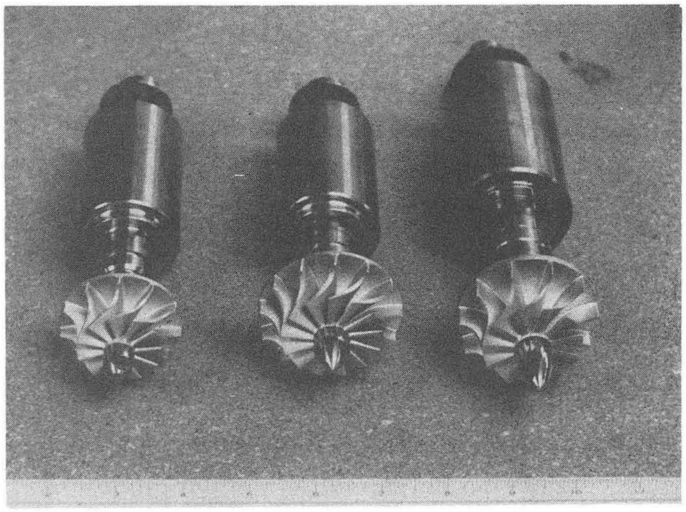


Photo. 32. (CBB 831-463). The three CHL Sulzer oil bearing turbine shafts and wheels. Work energy is removed in the oil friction and heating. Photo courtesy FNAL.

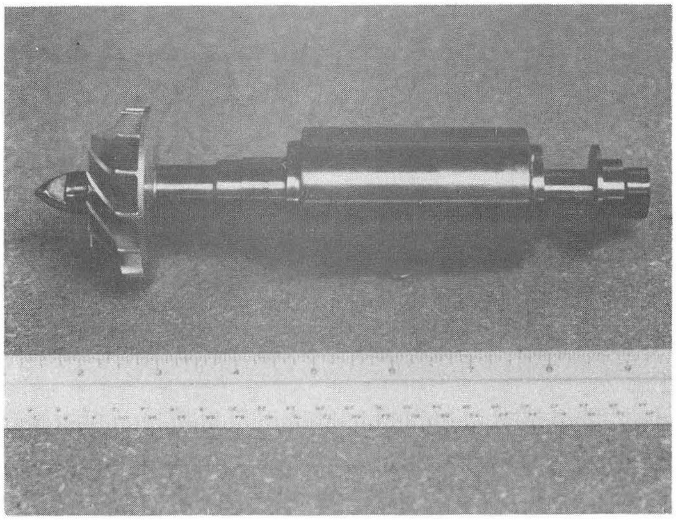


Photo. 33. (CBB 831-461). A CHL turbine wheel and shaft assembly - scale in inches. Photo courtesy FNAL.

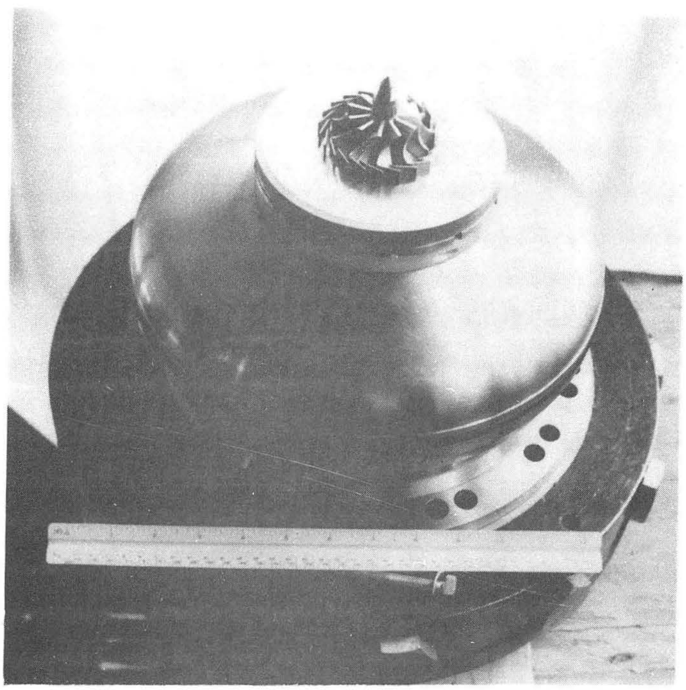


Photo. 34. (CBB 831-479). A CHL turbine cartridge assembly - scale in inches. Photo courtesy FNAL.

3.0. Fermi National Accelerator Laboratory, 23 kW

3a. Introduction

Fermilab, established in 1967 at Batavia, Illinois after energetic national competition for site location, completed the present accelerator, one of the two worlds largest, in 1972 (CERN, Geneva is the other). The machine accelerates protons from a H_2 ion source to 100 keV; to 750 keV in a Cockroft-Walton generator; thru a linear accelerator 200 ft long to 200 MeV, stacked and bunched into a booster synchrotron ring 500 ft. in diameter to 8 GeV and thence into the main ring. The synchrotron main ring is 6562 ft (2.00 km) diameter, about 4 miles circumference and final energy is 200-500 GeV (giga electron volts, 10^9). The beam exits tangentially from the ring and travels about one mile to each of three separate physics experimental areas, proton, meson and neutrino.

A synchrotron (MacMillan, LBL, 1942 and Veksler USSR 1942) accelerates particles in an orbit of fixed radius by successive voltage kicks on each pass. As the energy (velocity) increases, the AC magnetic field must increase to hold the particle in orbit. A cyclotron (Lawrence, LBL, 1932) kicks the particles across a voltage gap and they spiral outward, radius increasing; in a fixed DC magnetic field.

At Fermilab there are about 1000 conventional water-cooled AC electromagnets each 20 ft. long, that accept the protons at 8 GeV. The field rises to 1.8 Tesla (18 kilogauss) as the velocity increases to 400 GeV, from a 3 megavolt kick every trip around the fixed orbit in a 2 inch x 5 inch vacuum tube.

There was great motivation to build another accelerator with superconducting magnets. Under the vigorous leadership of R. Wilson, the first director and a former Lawrence student, a development program started in the early 70's. L. Lederman, the director since 1979, continued the leadership, thru the establishment of magnet production facilities at Fermilab. By the end of 1981, 14 units a week of accelerator quality magnets were being produced and LHe tested. This went on 7 days a week for two years. In June 1982 the machine was shutdown for installation of 774 SC dipole magnet assemblies (8400 lb), 216 SC quadrupoles and 186 SC correction elements in the ring tunnel, 17 ft. underground. Early 1983 start-up of the new machine proceded. Benefits of the superconducting magnets include: The stiffer magnetic field, 4 Tesla plus, permits particle acceleration to energy levels of 1000 GeV or 1 TeV and the machine is called Tevatron. At the same time power use is reduced by at least a factor of three, from the 40 MW electrical consumption of the original accelerator.

3b. Heat Loads and Distribution

The magnet iron for the dipoles is 15 inch wide x 10 inch high x 200 inch long, with a 7 inch dia. bore to accept the cryostat. This contains the vacuum bore tube (a "cold bore"), the Nb-Ti magnet coil immersed in single phase LHe, the "go" circuit, surrounded by the two-phase LHe return flow. LN shields complete the assembly. LHe flow is 20-30 g/s for single magnet heat load of 5 to 7 W static. The heat load is at least doubled by the electric pulse dependent on the ramp rate.

The LHe distribution system is a truly remarkable and unique design by Vander Arend and Fowler.[35] It demonstrates two important engineering principles; when confronted by an impossible design problem; break it up into small sub-sets that can be solved; and always select the most economic of any impossible solution.

Early design studies included twenty 1500 W refrigerators around the ring but this was discarded due to insufficient power available around the ring. Pumped liquid to provide forced flow two-phase coolant was also discarded. A pump test loop was built and operated to test the sub-cooled LHe heat transfer. Power input at the 4.5 K level is very expensive and less reliable, when compared with a 300 K compressor.

The present design divides the ring into six sectors, A-F, and subdivides each sector into four parts, each of which has a single refrigerator, a total of 24 "satellite" refrigerators. One unit cools, 1/24 (π 6562) = 860 ft. From one of the 24 stations, LHe flow, 40-60 g/s enters the tunnel and flows in two directions, handling about 16 dipole and 4 quadrupoles each way for a total of 40. The flow enters a sub-cooler, passes thru each 20 magnet string as a liquid heat transfer media, expands from a J-T valve at 2 atm and returns around the magnet as two-phase, thru the subcooler and back to the refrigerator.

The CHL (Central Helium Liquefier) delivers LHe into an 11,000 gal. LHe dewar, a 6000 meter transfer line and each of the 24 satellites. They deliver LHe to the main ring. Each satellite uses ~ 144 l/hr LHe, 0.5 g/s for power lead cooling and 4.5 g/s for "boosting" to 966 W.

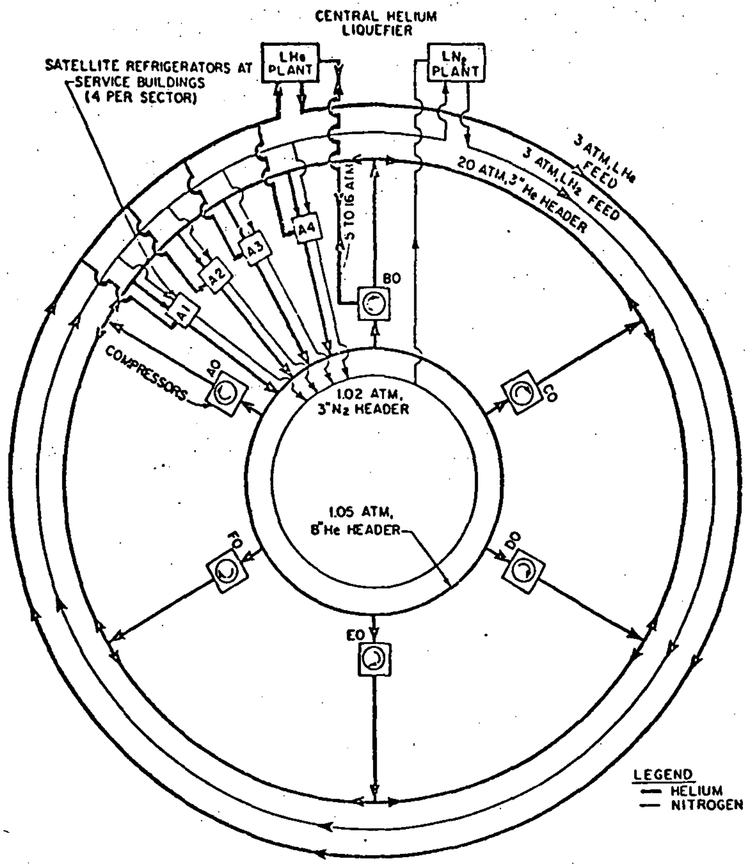


Figure 23. Layout of Fermilab Refrigeration System.

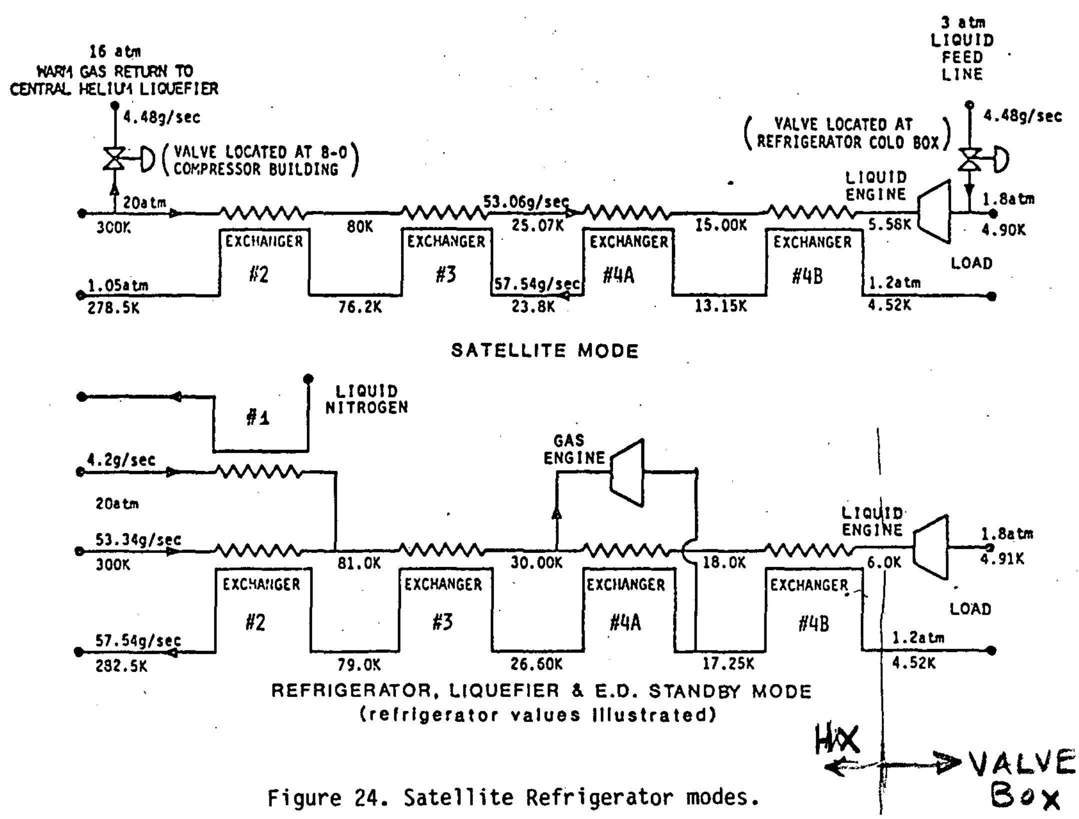


Figure 24. Satellite Refrigerator modes.

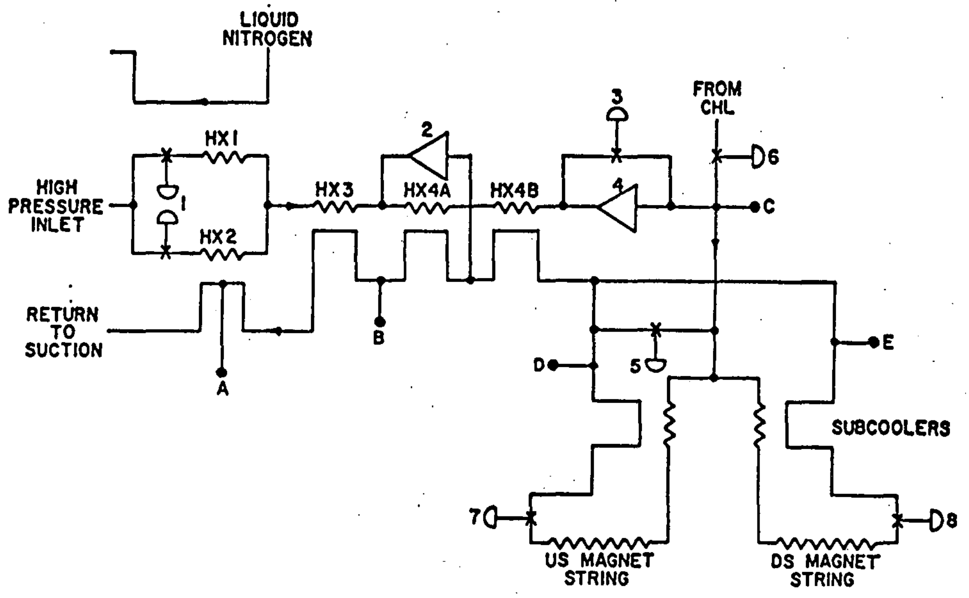
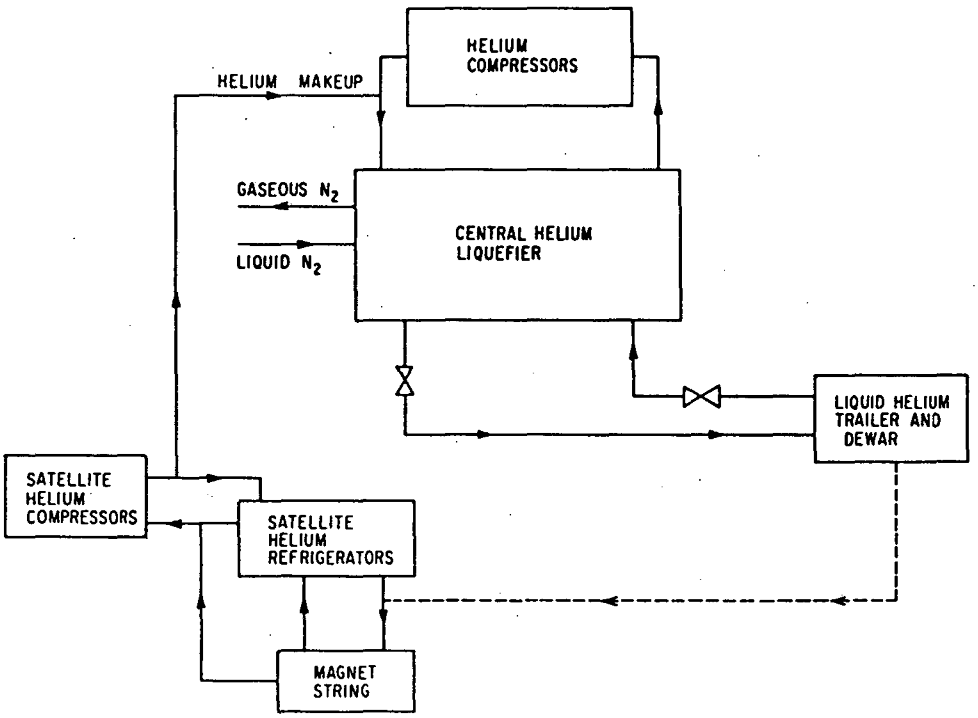


Figure 25. Satellite system schematic showing process variables (letters) and control points (numbers).



Ref. 36

Figure 26. Flow schematic of the FNAl Central Helium Liquefier.

3c. The Satellite Refer[35,37]

The refrigerator consists of a 35 ft. long HX column, a standby gas expander engine, two flow splitting sub-coolers and a wet expander engine. It can operate in four different modes. In the "satellite" mode its production is maximum for the energy doubler or Tevatron. In this mode LHe, 4.5 g/s (plus 0.5 g/s power lead flow) is supplied continuously from the central liquefier. This creates an imbalance in the HX flow, 53 g/s supply vs. 57.5 g/s return, and eliminates the need for LN pre-cooling or 30 K expander cooling. Compare this imbalance mode with a liquefier mode which has unbalanced excess flow in the forward direction. The wet engine expands from 20 atm to 1.8 atm and the cold end refrigeration comes from three sources: 0.44 from HX imbalance, 0.48 from the wet expander and 0.08 from the CHL flow. The more LHe fed to the lower end, the more cooling watts produced, the only limit being the return side HX pressure drop.

The other three modes use LN pre-cooling and the gas engine at 30 K, as well as the wet engine. They provide a stand-alone method of magnet operations without CHL support and enhance system reliability.

The refrigerator mechanical equipment was especially designed for easy construction, such that any quality workshop can produce it. The HX are 0.5 inch dia. copper tubes with 15 fins per inch, 1 inch dia., wound on 8 - 10 inch dia. mandrels, together with cotton rope to channel return side flow. The HX are built in modular units and can be arranged in a single long column or multiple parallel columns. The engines have been refined thru long development. Fermilab uses 24 units on their main ring plus another 10 or so for external beam lines and auxilliary equipment. LBL has three "satellites", Texas A and M has one and Michigan State University an upgraded unit. (800W-200L/hr)

Table 3c.1. Satellite System Parameters[37]

Mode	Consumption	Production
Satellite	129 L/hr LHe	966 W
Refrigerator	52 L/hr LN	623 W
Liquefier	84 L/hr LN	126 L/hr LHe
Energy Doubler Standby	59 L/hr LN	490 W plus (27 L/hr LHe)

Compressor: 1 to 20 atm at 57.5 g/s; 350 HP-260 kW, returns 4.5 g/s at 20 atm and 300 K to CHL in satellite mode. Mycom two-stage oil flooded rotary screw compressor.

3d. The Central Helium Liquefier (CHL)[36,38,39]

The main components are the compressors, an 11,000 gal LHe dewar, a 14,000 gal LN dewar, purifier system and cold box. The plant includes eleven 30,000 gal. tanks for medium pressure storage at 250 psi and eventually a 3000 HP 100-ton LN reliquefier. Two 2000 HP three-stage reciprocating compressors deliver ~1000 g/s at 12 atm. Return flow at 300 K (20 atm) from the 24 satellites (the 4.5 g/s LHe used at each satellite) joins the CHL process stream. The cold box has been designed for 1270 g/s, 0.6 liter LN per liter LHe, to yield 4875 liter/hr of LHe. Were the same system to operate as a refrigerator, yield would be 12,000 W. The system first operated in 1980 and by Sept. 1982 had accumulated 3200 hours.

Cold box design and fabrication,[36] by Koch Process Systems, uses three Sulzer oil-bearing turbines, Trane Al HX, and high vacuum insulation. The plant has a calculated efficiency of 0.22 Carnot; six thermal cycles were examined, as well as equipment optimized to achieve a high efficiency. To produce the required minimum of 4000 L/hr requires at least four stages of refrigeration. A cycle was selected with LN pre-cooling and three turbo-expanders at three temperature levels. A wet engine, rather than a J-T valve gave only a 0.07 performance increase and was rejected on the basis of economics and reliability.

The final cycle was optimized with the aid of a CDC 6000 Computer. HX effectiveness values were held below 0.985, and performance and constraints of oil bearing turbines were considered. Three levels of expander pressures were needed. To find the optimum state points, the turbine exit temperatures, the intermediate pressure level and the critical HX temperature differences were varied. Pressure drop and external heat leaks were held constant. By employing the First and Second Laws of thermodynamics and conservation of mass, all the state points and liquefaction rate for a given flow can be determined.

The vertical cold box is 37 ft high, 10 ft. dia. and weighs 70,000 lbs. It contains the HX, piping, valves and turbines. The vacuum shell is 31 ft long and 10 ft. dia. with 0.5 inch carbon steel wall. There are nine HX stages arranged in four brazed Al plate-fin HX sections.

Table 3d.1. Physical and Thermal Characteristics of the Heat Exchangers[36]

Heat exchanger section	Heat exchanger number	Core size	Heat transfer area, ft^2	Weight, lb
1	HX-1A HX-1B	36 in. x 25 in. x 14 ft	15,630	4,300
2	HX-2 HX-3 HX-4	25 in. x 25 in. x 22 ft	28,425	4,400
3	HX-5 HX-6	25 in. x 20 in. x 21 ft	23,135	4,000
4	HX-7 HX-8	17 in. x 12 in. x 18 ft	7,445	1,600

Table 3d.2. Thermodynamic Characteristics of Oil-Bearing Turbine[36]

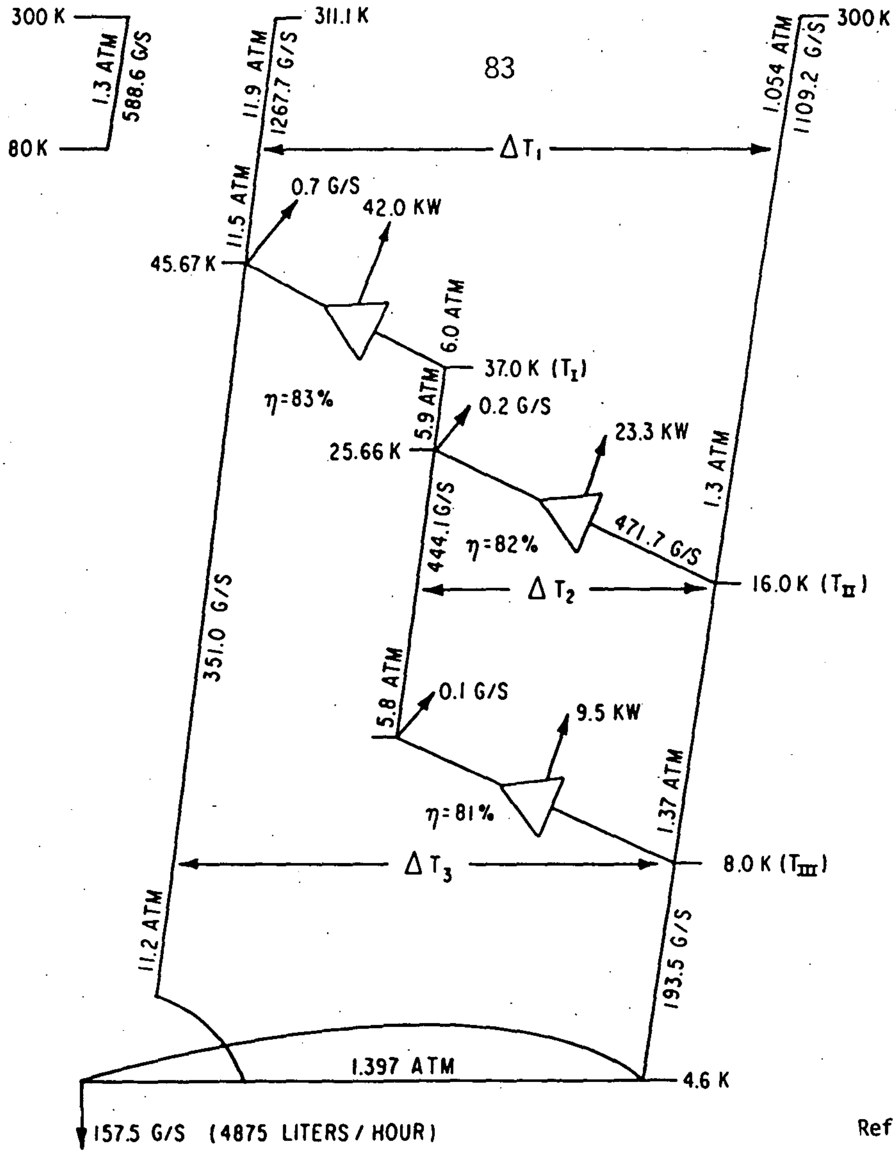
Turbine	Design speed, rpm	Inlet temperature, K	Mass flow, g/sec	Isentropic efficiency,	Power extraction, kW
Upper	73,160	45.67	916.0	82.9	42.2
Middle	75,990	25.66	471.7	82.0	23.0
Lower	59,890	12.88	444.0	81.0	9.64

Table 3d.3. Compressor Performance[39]

Item	Compression Stages		
18 Inch Stroke, 277 rpm	I	II	III
Cylinders used	1.5-34.25"	1-27"	1-15", 9", 6"
Displacement, CFM	7903	3268	1441
Inlet pressure, psia	15.4	38.6	88.0
Discharge pressure, psia	40	90	181
Discharge temp. °F	325	325	287
Inlet temp. °F	80	100	100
Actual CFM	6708	2787	1224
Mass flow, g/s	540	540	540
B.H.P./stage	611	536	502
Total	1649 HP - 1237 kW		

Overall Volumetric Efficiency - 0.84

Overall Isothermal Efficiency - 0.67



Ref. 36

Figure 27. Liquefaction cycle of the FNAL Central Helium Liquefier.

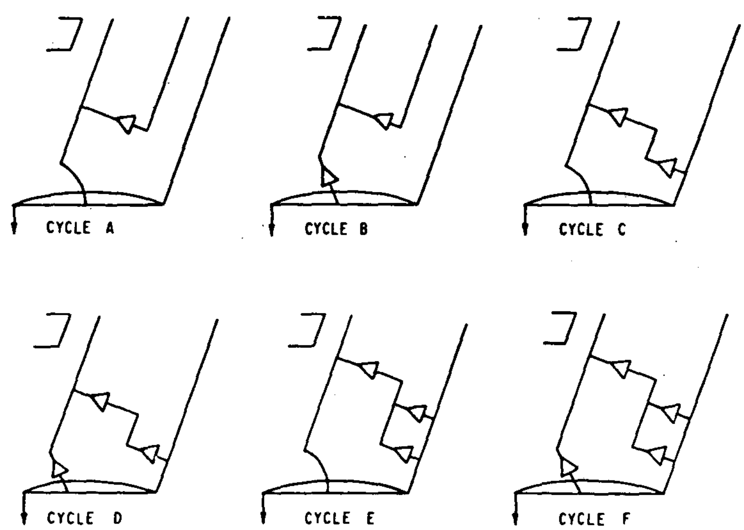


Figure 28. Cycles considered. Ref. 36

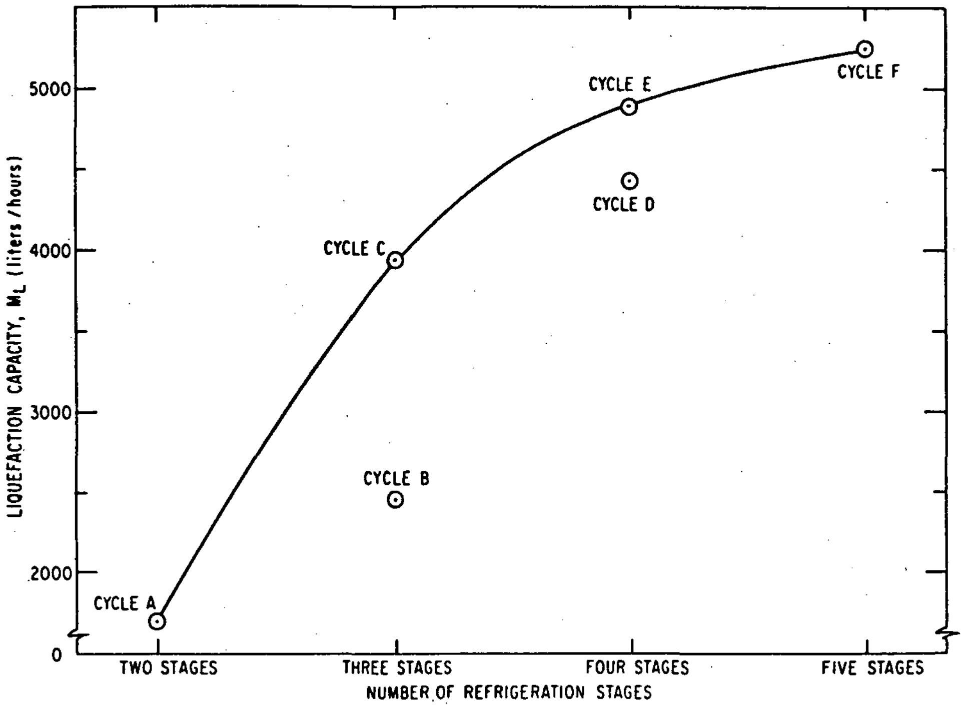
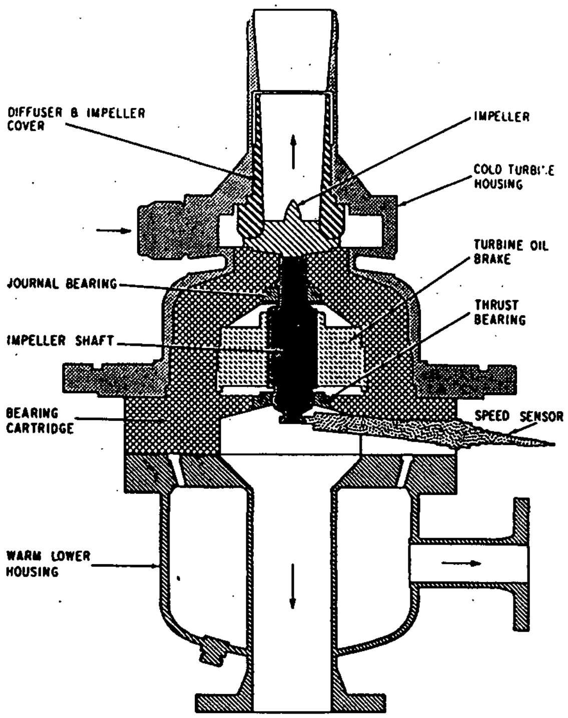


Figure 29. Thermodynamic performance of the various liquefaction cycles for the FNAL Central Helium Liquefier. Ref. 36



Ref. 36

Figure 30. Sulzer Oil Bearing Turbine.

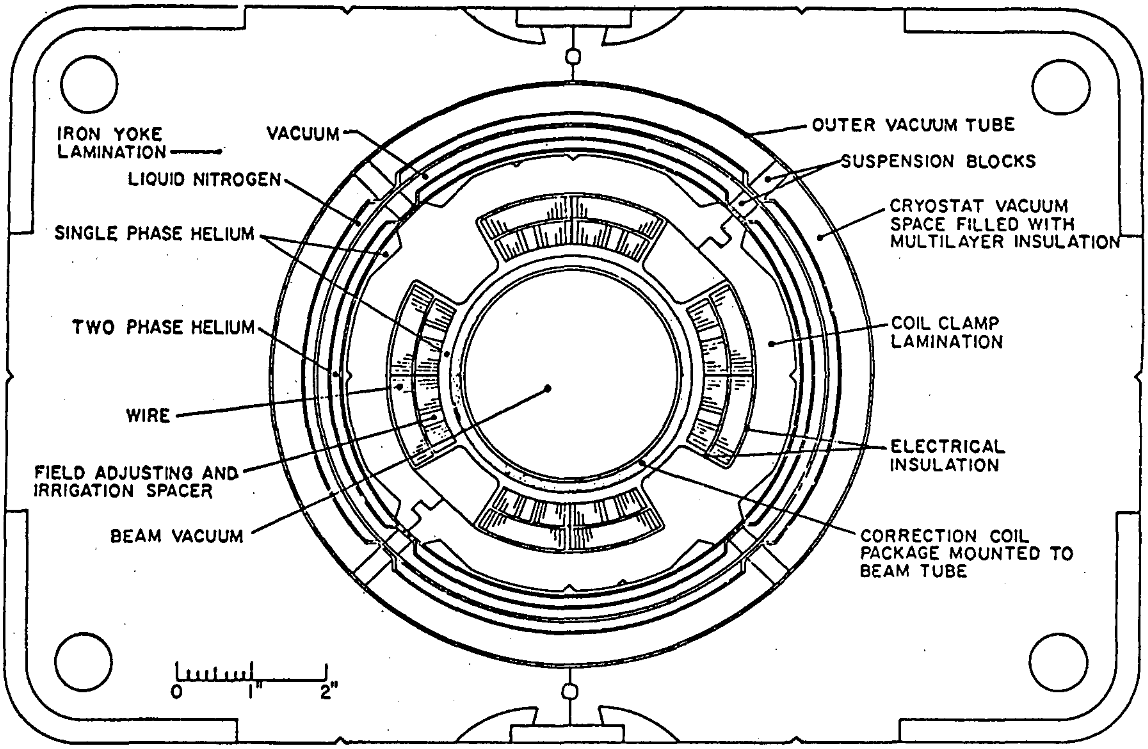


Figure 31. Cross section of quadrupole magnet.

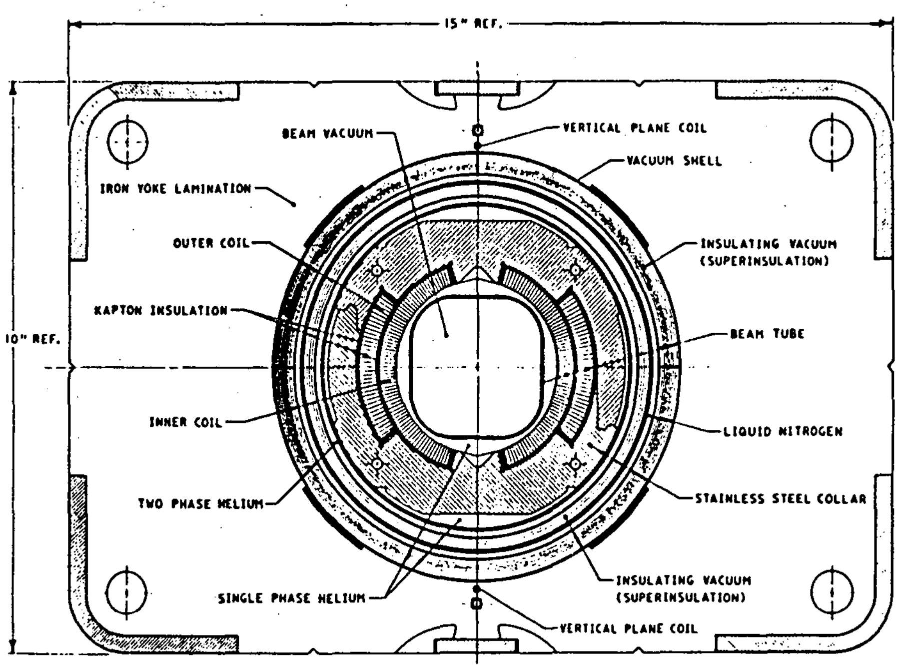


Figure 32. Energy doubler dipole magnet - Fermilab. Ref. 41

The three plant compressors[39] were obtained as surplus from an Air Force Rocketdyne plant in California. Originally they were used in a high pressure, 3000 psi air reduction plant for LOX. These Worthington six cylinder, five stage double-acting compressors, are horizontally opposed for low shaking loads and have 4000 HP synchronous motors. The machines were moved to Fermilab, rebuilt, and restaged (three) for helium at 12 atm.

Originally for air - 18 inch stroke, 277 rpm[39]

First stage - 34.25" - 2 cyl-double acting (DA)

Second stage - 27 inch	Fourth stage - 9 inch
Third stage - 15 inch	Fifth stage - 6 inch

After study of four options, the machines were restaged and fitted with graphite filled Teflon rings. Some oil lubrication is used. The synchronous motors reduce power factor and the air-cooled inter- and after-coolers also help to reduce costs. Table 3d.3 shows performance.

3e. Computer Control and Simulation[40]

For this large widely scattered system with many components and many modes the computer becomes mandatory. The computer can monitor data, trends and performance. It can control steady-state and transient modes and it can optimize power use and output. Effective alarming, interlocking and early warning are also possible. Capital costs for computer control are usually a small percentage of total large plant costs. This is not necessarily true for small systems.

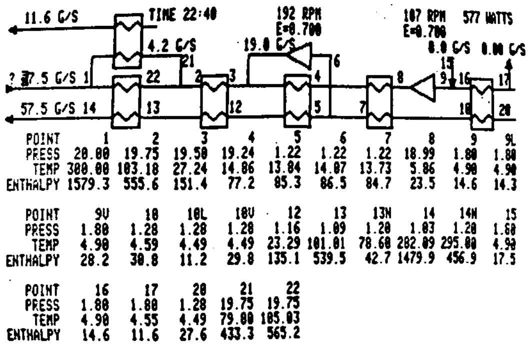
The cryogenic system requires a number of closed loop servo systems to control and monitor:

1. The CHL
2. The satellite compressor suction and discharge
3. The 24 satellite refrigerator systems together with their dual magnet strings
4. The LN system.

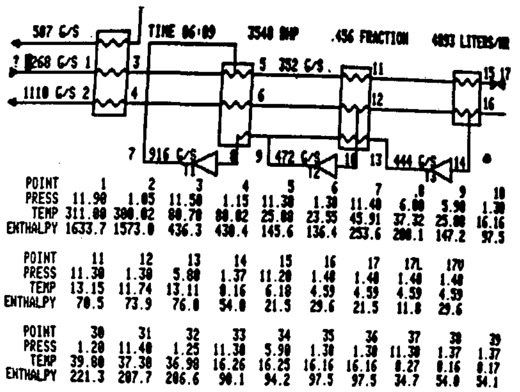
Sensing elements of vapor pressure thermometers, diodes, resistors and pressure transducers are used. As many as seven valves on each satellite system control LHe flow from the CHL, J-T flow on each magnet string, engine speed and LN pre-cool. The accelerator central computer will collect the information and provide balance for the whole system.

A comprehensive computer program was developed to examine the operation of the individual components as well as the total cryogenic system. This simulator calculates the refrigeration available over a wide range of operating parameters. It optimizes conditions, identifies equipment needing improvement and simulates pulse and fault conditions.

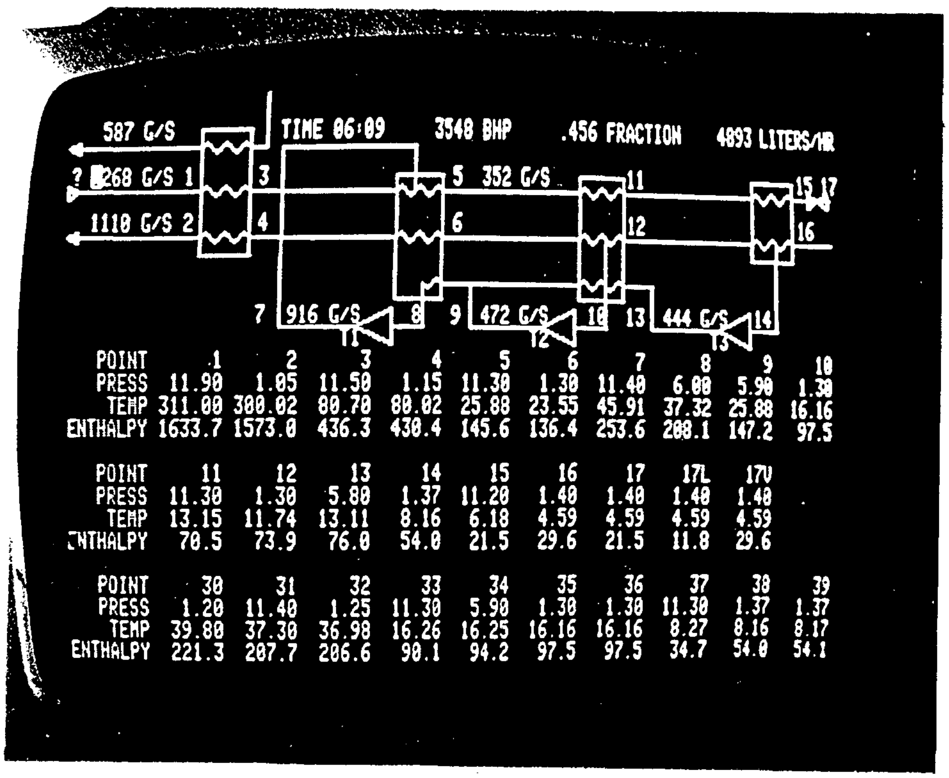
The calculation method uses enthalpy (not temperature) since the equations of heat balance at each HX are linear in enthalpy. This avoids the problems of changes of specific heat with temperature. A refrigerator with N counter-flow HX must conserve energy, so there are 2 N simultaneous equations needed to define all the necessary process points. Overall heat transfer coefficient times area, UA, must be correct and N integrations solve the heat transfer equation for each



Sample simulator display for the Satellite Refrigerator.



Sample simulator display for the Central Liquefier.



Ref. 40

Figure 33 Computer terminal display of interactive software simulator developed at Fermilab for CHL and Satellites.

HX. An iteration process solves the N energy balance equations. Both satellite and CHL results have been very successful and real operating process points confirm the algorithm. In an upgrade study for the CHL using the present HX, 0.40 flow increase (the third compressor) would require 0.40 turbine flow increase and yield 0.37 plant capacity increase.

3f. Schedule and System Merits

By late 1982 all the components had been completed. The accelerator was shut down in June 82 to install the magnets in the tunnel and that work finished in Spring 1983. Starting February 82, using the CHL and three satellites, three continuous magnet strings were cooled, filled and charged to 4000 amperes (Sector A). In December, sectors E and F were put into test operation. July 3, 1983 brought complete system operation and the birth of a new superconducting accelerator, with levels up to 700 GeV.

Some of the merits of the system are:

1. High thermal efficiency of the large CHL.
2. Flexibility and redundancy of the 24 satellites can provide a high degree of reliability.
3. Power and cryogenic transfer lines to the main ring are simple and small.
4. Low circulating mass flow due to constant temperature of the two-phase boiling LHe and the 24 feed points.
5. Relatively low LHe inventory, 22 liters/magnet.
6. The magnet warm iron implies rapid cooldown and warm-up.
7. The cold vacuum bore has system simplicity.

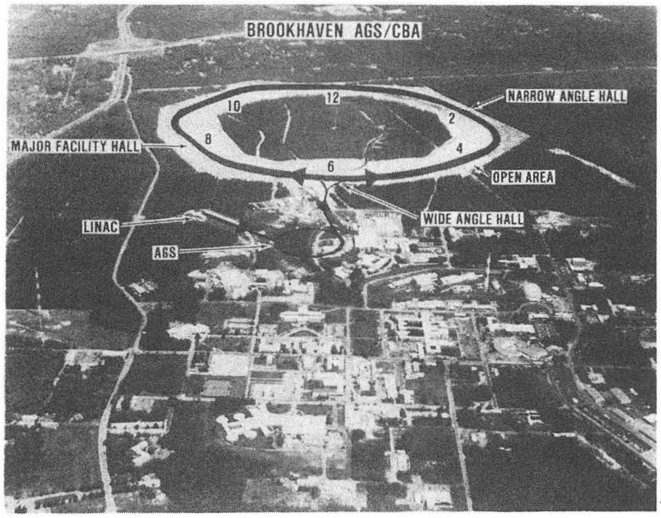


Photo. 35. (CBB 831-762). Aerial view of BNL showing the injection LINAC, the Alternating Gradient Synchrotron (AGS) and the Colliding Beam Accelerator (CBA) main ring with its six sectors. Photo courtesy Brookhaven National Laboratory.

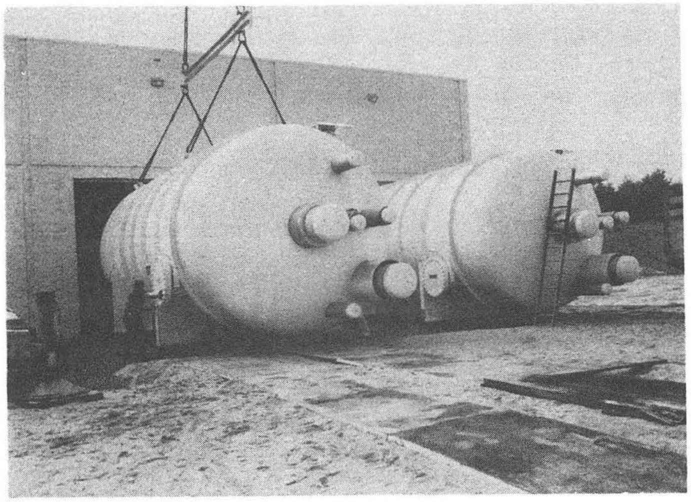


Photo. 36. (CBB 831-487). Installing first two HX tanks for the CBA helium refrigerator. These two tanks are both the same and redundant (300 K to 80 K). Photo courtesy BNL.

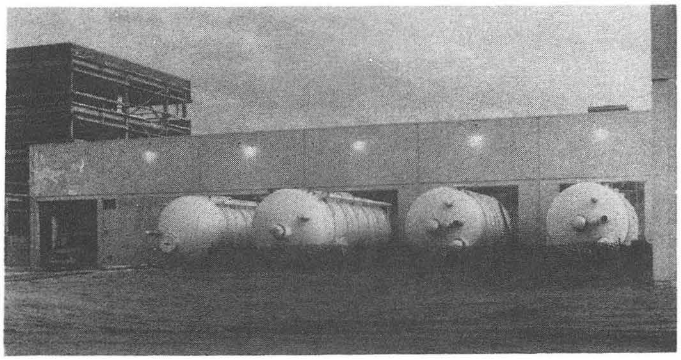


Photo. 37. (XBB 831-760). Four HX tanks installed for the CBA helium refrigerator. Control room building for CBA in background. Photo courtesy BNL.

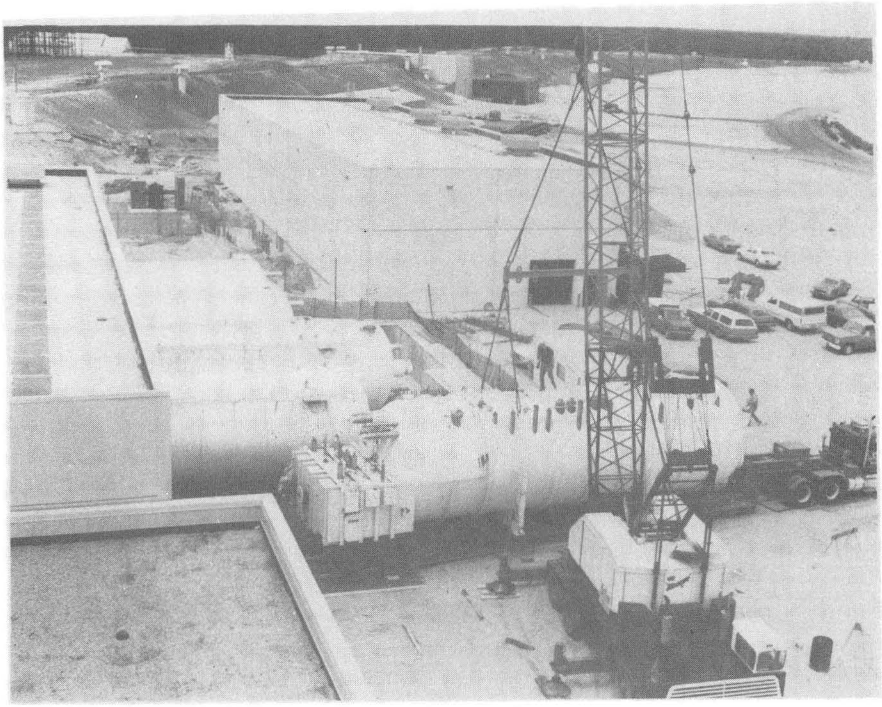


Photo. 38. (BBC 841-533). Fifth and final cold box for CBA being unloaded June 1983. Compressor building in background.

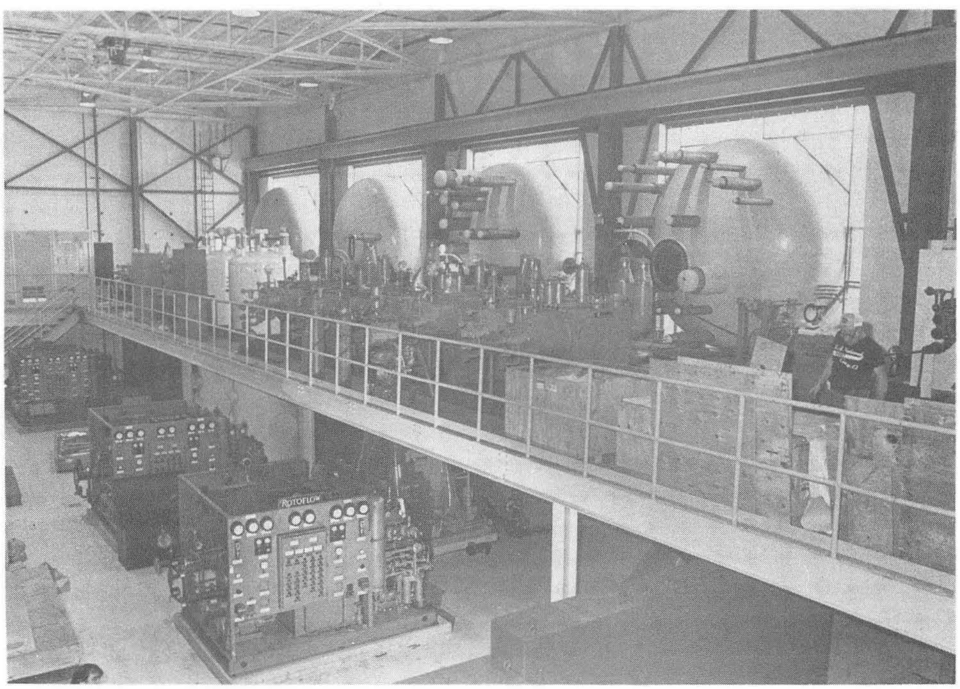


Photo. 39. (XBB 841-529). Interior view of cold box arrangement, Oct. 1983. Oil skids below for turbines, turbine pods on mezzanine, two cryo-adsorbers (white) sticking thru mezzanine. Photo courtesy BNL.

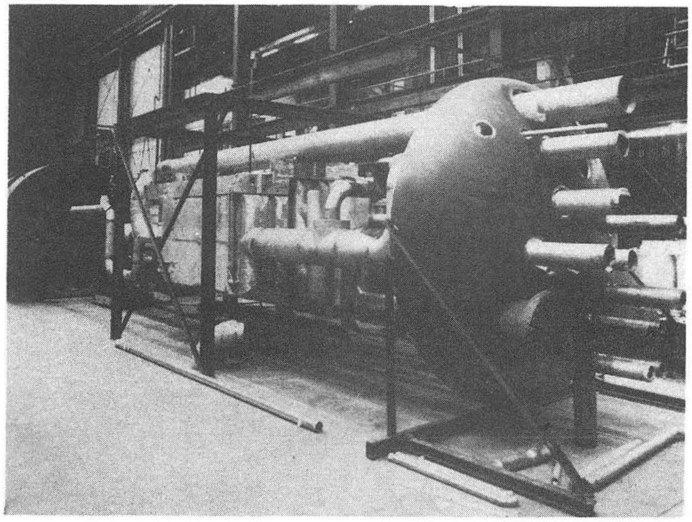


Photo. 40. (XBB 831-446). CBA cold box in construction at Koch Process Systems. Photo courtesy KPA.

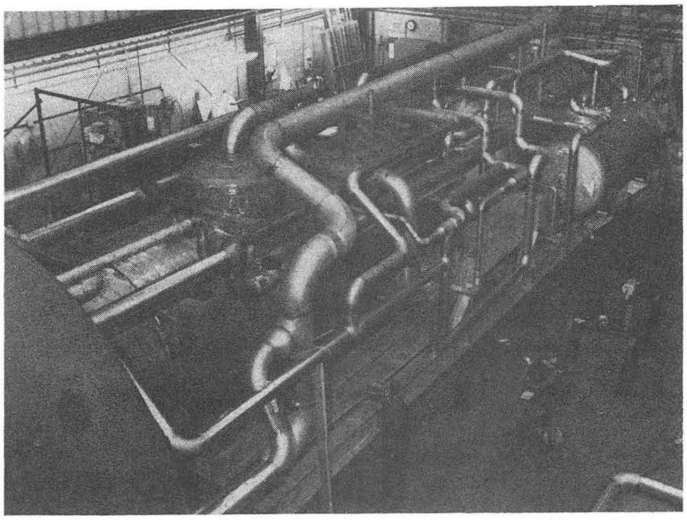


Photo. 41. (XBB 831-447). Another cold box for CBA under construction at Koch Process Systems, Westborough, Mass. Photo courtesy KPS.

4.0 Brookhaven National Laboratory, 24.8 kW[42-46]

4a. Introduction

BNL, establilshed in 1946 and located at Upton in the center of Long Island, New York, is one of the major U. S. research centers. Its first large synchrotron, the Cosmotron (1952) is retired. Its second machine, the AGS (Alternating Gradient Synchrotron, 1960) still operates. In the early 70's a new third generation proton machine was proposed. It would use the AGS for injection and consists of two accelerator/storage rings in a common tunnel, 3.8 km (2.36 miles) in circumference (1.2 km dia.). Protons are accelerated to 400 GeV and stored up to 24 hours in each ring; 1084 superconducting magnets hold the particles in orbit. Six intersection regions around the ring provide collision points for the counter-rotating beam reactions and physics experiments.

The machine name was Isabelle (Intersecting Storage Accelerator). In 1982 it was renamed the CBA (Colliding Beam Accelerator). Thru late 1981, magnet design and fabrication was in severe trouble, and consequently added long term funding was necessary. Limited high-energy physics research budgets have required a re-examination of the objectives and parameters of the new machine. By 1982, six successful magnets were completed and reached fields of 5 to 6 Tesla.

Most of the civil engineering work is complete, including ring and injection tunnels, three of the six intersecting experimental halls, and the control and refrigeration buildings. The large refrigerator is well advanced, with cold boxes and turbo-expanders completed and

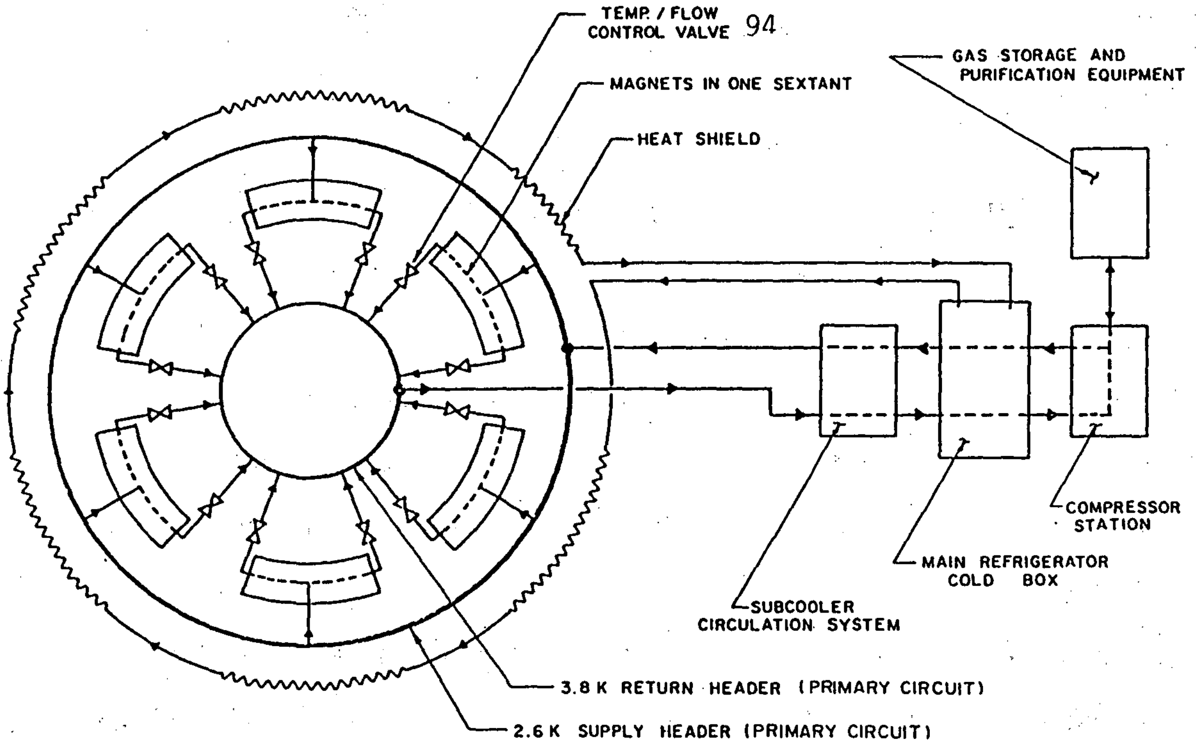


Figure 34. 400 x 400 ISABELLE helium flow schematic. Illustrates primary and shield circuits for one accelerator ring. See page 109 for later design. Ref. 45

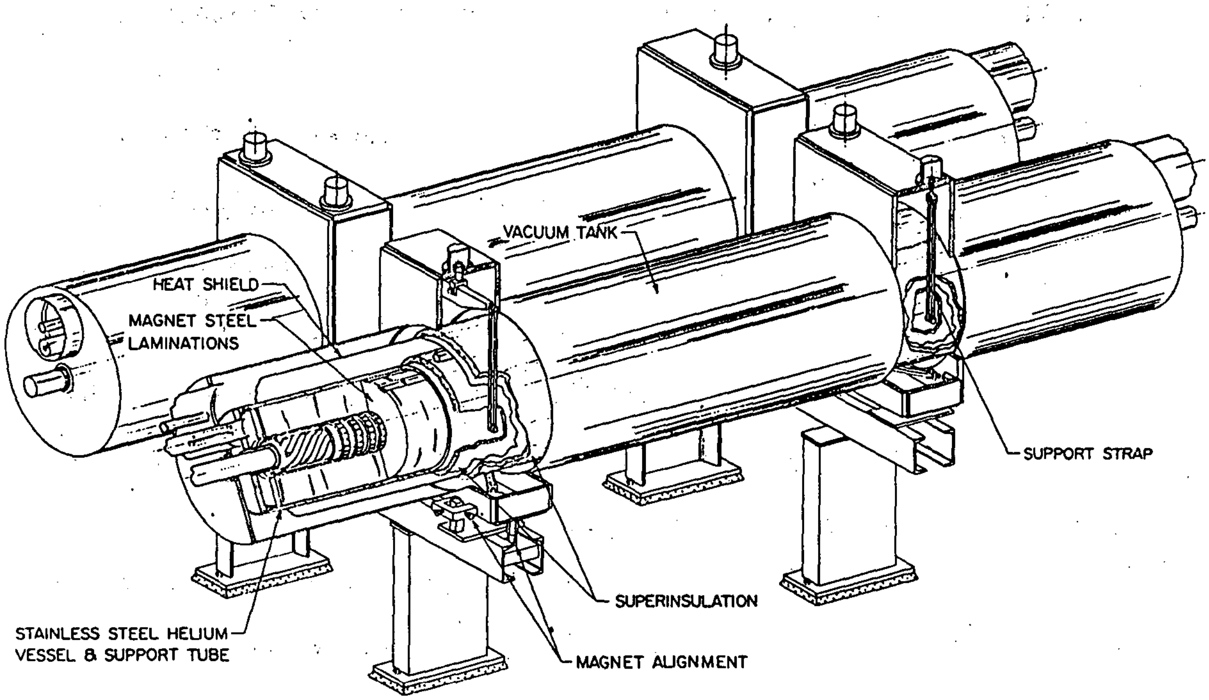


Figure 35. ISABELLE-CBA dual magnetic cold iron, warm bore.

delivered (Jan. 1983). Compressor contract signed in June, 1982 should provide complete assembly in late 1983.

4b. Magnets, Heat Load and Distribution

The dipoles consist of two layers of super-conductor, Nb-Ti alloy, surrounding a "warm bore" beam vacuum tube inside a cold laminated iron yoke. Length is 4.75 m long and weight is 6500 kg. Part of the magnet problem was the use of braid for the coils. In 1981, with the change to flat cable (23 multi-filament twisted SC wires of the Rutherford-Fermi type) the magnets were successful. Improved coil clamping also helped. All six prototype magnets reached fields of over 5 Tesla at 4.5 K before the first quench and 6 Tesla at lower temperatures.

BNL has great in-depth cryogenic experience with bubble chambers, large superconducting magnets (2.1 m H_2 B.C.), strings of accelerator beam line magnets (HEUB) and various refrigerators. BNL engineers are well versed in refrigeration techniques. Initial planning for the large accelerator was to use pool-boiling cryostats as had been used for all earlier magnet cooling. Magnet design direction motivated distribution method review and thus the force circulation cooling system (fccs) evolved. Many reasons for fccs versus pool-boiling were developed, including the reduction of control elements from 400 to 48. Supercritical flow answered the 3.8 K need.

The magnet cooling system uses super-critical forced flow in a closed loop separate from the refrigerator. This loop has a mass flow of 4054 g/s (equal to refrigerator total flow) circulated by a "cold compressor" with inlet at 3.47 K and a pressure ratio of 1.31. Wheel diameter is 4.11 inch, RPM-10-11,000. Pressure in this loop is held to a minimum of 5 atm (to provide for power lead cooling pressure drop) but can rise to design pressure of 15 atm during cooldown or magnet quench. Flow circulates thru the magnets, which range in temperature from 2.6 K to 3.8 K, to the sub-cooler on the delivery end of the refrigerator.

The sub-cooler has two stages of LHe pots, one at 2.6 K vacuum pumped with a cold two-stage pump to 0.1 atm and the second LHe pot pumped to 0.35 atm and 3.47 K. Significant amounts of power are added to the cold end by the three pumps, approximately 10 kW heat load at 4.5 K and 20 kW between 4.6 K and 9 K. Refrigeration is costly and complex when the temperature level is set below 4 K. The main refrigerator has a return pressure of 1.4 atm and therefore can only create temperature levels to 4.6 K.

The flow is distributed to the magnets arranged in sextants. The magnets (45) in a half sextant are cooled in series terminated with a temperature control valve. Thus for one ring there are 12 loops and 24* parallel paths for the two rings. Temperature control of all the magnets will be provided by a process control computer and all magnet string exit temperatures will be matched.

4c. Isabelle Main Refrigerator[42,43,44,45]

The main plant, 24.8 kW at 3.8 K has a base design value of 13.7 kW (4200 g/s from 2.6 K to 4.2 K), 100 g/s liquefaction for power lead cooling, and 55 kW at 55 K (350 g/s from 40 K to 70 K) for heat

*Later design combined the sextants to thirds and reduced the parallel paths to 12. See Fig. 34b. page 109.

shields. There is no LN pre-cooling because the plant is primarily a refrigerator, (balanced flow). Economics and operations also justify this. Five stages of turbo-expanders are used and with 0.60 isothermal compressors, system efficiency is about 0.26 Carnot. Input power is estimated at 13 MW. As a liquefier, using both sets of redundant expanders 1 and 2, 11,000 L/hr at 4.6 K can be produced.

The Claude cycle design uses five expanders. Plant efficiency increases with the number of expander steps (approach to reversibility) and plateaus at five. Cycle studies with three and five expanders were made, confirming the final selection. Expander stages 1 and 2 and 3 and 4 are series connected from 16.4 atm to 1.1 atm. Mid pressure level is 8 atm. The series connection permits high flow rate with low pressure ratios (as compared to parallel connection) and makes higher turbine efficiencies. No significant cycle improvement was found at pressures greater than the 16.4 atm selected. The 1.1 atm suction was chosen to eliminate sub-atmospheric pressures in the main compressors. This reduces air contamination possibility and improves the compressor mass flow throughput.

BNL developed a comprehensive computer program, using NBS helium properties, to examine and optimize the whole system. With heat loads, components and circuit defined, the state points can be determined from basic thermodynamics. Heat and mass balance equations are iterated until equipment efficiencies, pressure drops and heat leaks are consistent. The BNL program first calculates the required mass flow to satisfy the heat loads on the cold end. The interface needs at sub-cooler and main refrigerator are then solved from vacuum compressor estimates, and finally, turbine and HX temperatures up to 300 K are fixed.

Turbine inlet temperatures are selected to optimize the cycle. Outlet pressure for turbine 5 is set at 2.5 atm (above critical) to prevent liquid forming in the wheel, although this may not be necessary. The expanders are oil-bearing turbines made by Rotoflow Corp., Los Angeles and all have variable inlet nozzles to optimize efficiency over a wide flow range. The high flow permits large enough wheel assemblies for this construction. The five stages of turbines have shrouded wheels and No. 1 impeller wheel is 4.625 inch diameter, No. 5 is 1.75 inch and 40,000 RPM. Expanders 1,2,3,4 have redundant turbines in parallel, primarily for reliability, but the upper ones are used for cooldown from 300 K.

Heat exchanger design effectiveness does not exceed 0.98 and most are above 0.94. Total UA is about 2100 kW/K. All HX are brazed Al plate-fin in the horizontal position. Special attention to flow distribution was taken to prevent HX transfer area loss in this position. The warmest HX uses three cores 0.91m x 1.07m x 4.6m manifolded in parallel by a 0.8 m header. All HX above 152 K stop oil and water and are redundant for reliability. The HX are switchable for clean-up and frozen plug removal. Dual bed purifiers at 69K stop oxygen and nitrogen with special protection from off-cycle desorption on temperature upset. Improperly released contaminants might otherwise travel to the cold end, freeze, and then plug the system.

The modular design consists of five horizontal vacuum insulated cold boxes assembled with one end inside the building, together with interconnecting piping and turbines. The subassemblies were fabricated by Koch Process Systems in Westborough, MA. Designed for easy

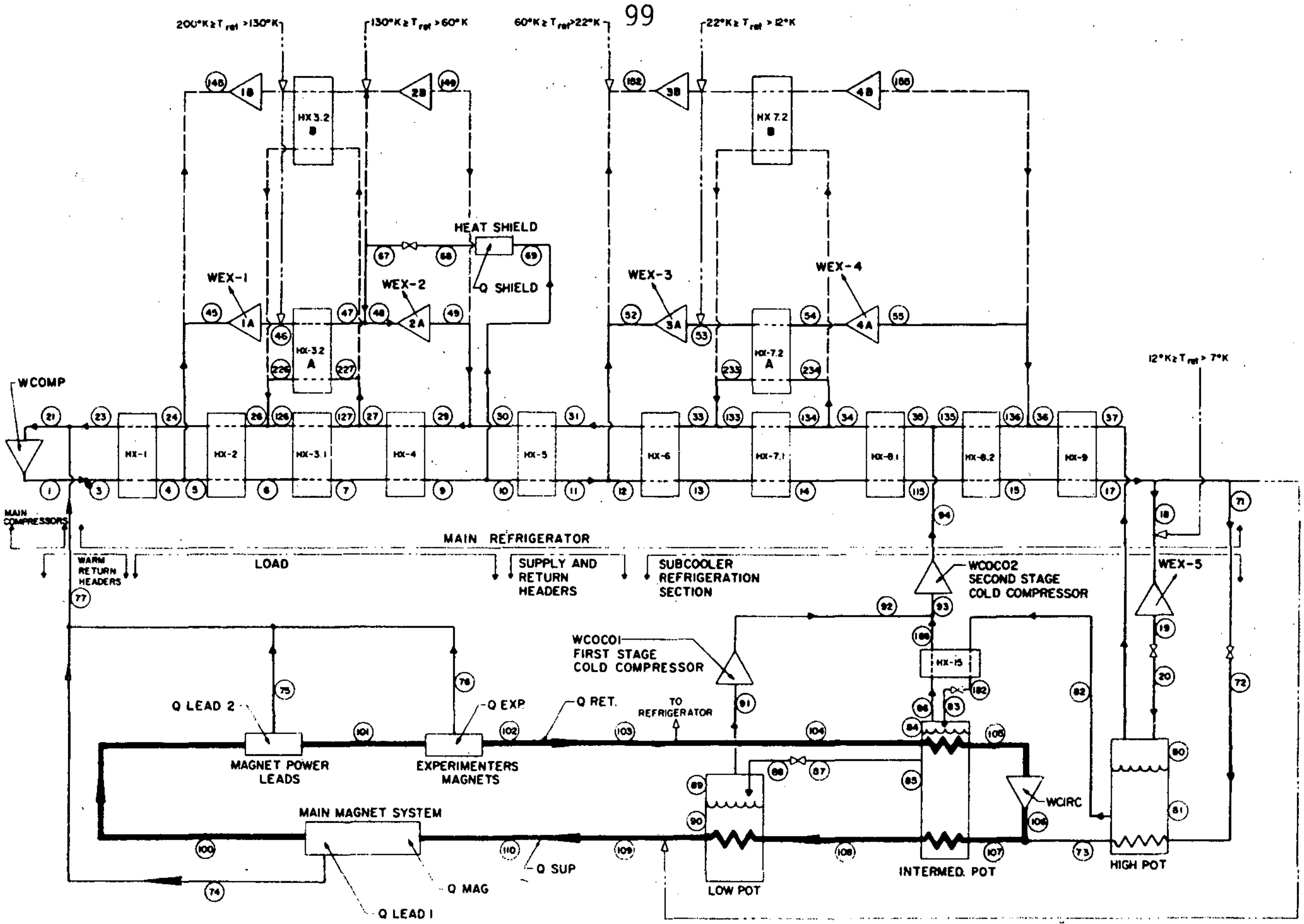
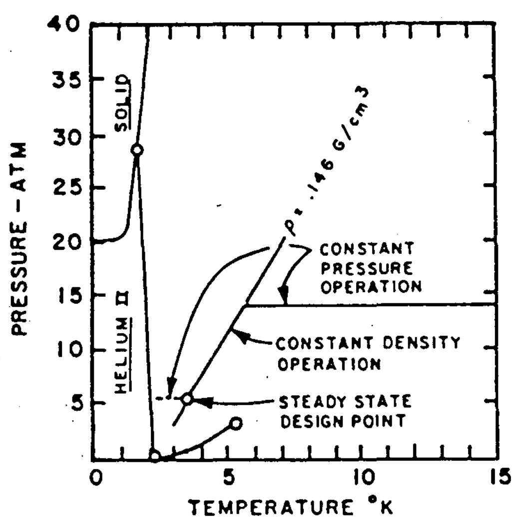


Figure 36. Flow diagram for ISABELLE Refrigerator with its redundant turbines and heat exchangers.

Ref. 43



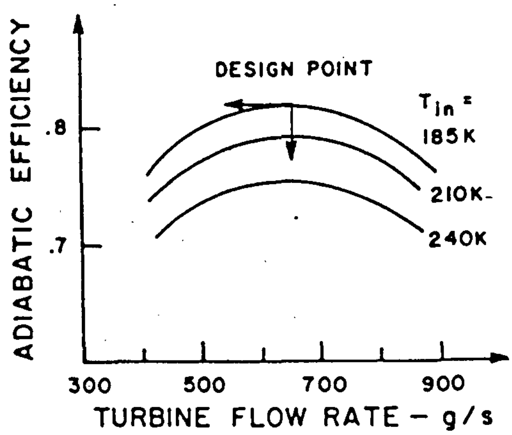


Figure 37.

Pressure vs. temperature for ISABELLE load.

Ref. 43

Figure 38. Efficiency for Turbine 1.

Ref. 43

PROGRAM ISA62

CALCULATED PERFORMANCE OF A HELIUM REFRIGERATOR WHICH UTILIZES 5 EXPANDERS AND 3 COLD COMPRESSORS. DELIVERY OF THE REFRIGERANT IS IN THE FORM OF COMPRESSED LIQUID HELIUM WHICH IS CIRCULATED BY ONE OF THE THREE COMPRESSORS IN THE CYCLE.

SUMMARY OF SYSTEM PARAMETERS

SYSTEM DESIGN STEADY-STATE LOAD REQUIREMENTS

REFRIGERATION REQUIRED-WATTS							MASS FLOW REQUIRED-GM/SEC		
QMAG	QLEAD1	QLEAD2	QSUP	QRET	QEXP	QSHLD	F74	F75	F76
7650.	200.	2300.	1500.	2000.	0.	55000.	6.	44.	50.

OTHER ESTIMATED REFRIGERATION LOADS-WATTS

QHX1	QHX2	QHX3	QHX4	QHX5	QHX6	QHX7	QHX8	QHX9	QHX10	QHX11	QHX12
947.0	3240.0	2663.0	1880.0	240.0	410.0	600.0	520.0	180.0	340.0	290.0	290.0

EXPANDER/COMPRESSOR PARAMETERS

	MAIN COMPR.	COLD COMPR. 1	COLD COMPR. 2	CIRC. COMPR.	EXPANDER 1	EXPANDER 2	EXPANDER 3	EXPANDER 4	EXPANDER 5
ADIABATIC EFFICIENCY		.680	.670	.580	.810	.820	.800	.780	.760
ISOTHERMAL EFFICIENCY	.60								
INLET PRESSURE-ATM	1.050	.095	.340	4.150	16.227	8.860	15.579	7.922	15.494
OUTLET PRESSURE-ATM	17.250	.350	1.389	5.450	9.000	1.301	8.000	1.423	2.500
INLET TEMPERATURE-K	302.00	2.46	4.61	3.47	185.00	69.37	25.00	12.38	6.29
OUTLET TEMPERATURE-K	305.00	4.83	9.64	3.79	153.64	38.93	20.19	7.10	5.14
FLOW RATE-GM/SEC	4212.2	333.2	982.4	4054.0	658.3	1014.2	1729.0	1729.0	1368.9
WORK-WATTS	12427821.	3933.	24764.	6213.	-108816.	-162034.	-42130.	-36736.	-10489.
WORK-HP	16659.	5.	33.	8.	-146.	-217.	-56.	-49.	-14.

HEAT EXCHANGER PARAMETERS

	HX1	HX2	HX3.1	HX3.2	HX4	HX5	HX6	HX7.1	HX7.2	HX8.1	HX8.2	HX9	HX15
EFFECTIVENESS	.976	.942	.978	.978	.964	.939	.973	.986	.986	.699	.954	.939	.923
NTU	28.43	8.71	29.05	28.69	12.63	10.23	6.26	17.67	17.59	1.75	5.64	3.00	5.16
AU(KW/K)	607.09	161.00	523.50	95.93	238.09	165.66	53.37	138.23	152.71	18.52	54.22	8.46	20.19

LOAD SUMMARY

	PRIMARY LOAD		SECONDARY LOAD	
	SUPPLY	RETURN	SUPPLY	RETURN
FLOW RATE-GM/SEC	4153.98	4053.98	355.91	355.91
TEMPERATURE-K	2.59	4.19	40.00	69.35
PRESSURE-ATM	5.35	4.20	15.67	9.67
DENSITY-GM/CC	.154	.139	.01842	.00668
ENTHALPY-J/GM	7.32	10.63	222.00	376.53

FLUID PROPERTIES AND FLOW RATES

PRESSURE(ATM), TEMPERATURE(K), ENTHALPY(J/GM) AND FLOW RATE(GM/SEC)

POINT	PRESS.	TEMP.	ENTHAL.	FLOW
1	17.250	305.00	1604.33	4212.15
3	16.400	305.00	1604.04	4212.15
4	16.278	185.00	980.61	4212.15
5	16.278	185.00	980.61	3553.88
6	16.181	153.64	817.52	3553.88
7	15.999	69.37	377.82	3553.88
9	15.676	40.00	222.00	3553.88
10	15.666	40.00	222.00	3197.97
11	15.636	25.00	139.41	3197.97
12	15.636	25.00	139.41	1468.94
13	15.606	20.19	111.54	1468.94
14	15.575	12.38	62.75	1468.94
115	15.558	10.45	48.84	1468.94
15	15.543	7.30	28.27	1468.94
17	15.533	6.28	23.59	1468.94
18	15.494	6.29	23.59	1368.94
19	2.500	5.14	15.93	1368.94
20	1.434	4.61	15.93	1368.94
LIQUID FRACTION AT PT. 20 IS .775				
80	1.424	4.61	29.54	386.50
81	1.424	4.61	11.99	1061.35
82	1.424	4.61	11.99	982.44
182	1.414	3.47	7.07	982.44
83	.350	3.27	7.07	982.44
LIQUID FRACTION AT PT. 83 IS .953				
84	.350	3.27	29.03	649.21
85	.350	3.27	5.99	936.41
86	.350	3.27	29.03	649.21
186	.340	4.49	36.47	649.21
87	.350	3.27	5.99	333.23
88	.100	2.49	5.99	333.23
LIQUID FRACTION AT PT. 88 IS .913				
89	.100	2.49	26.63	333.23
90	.100	2.49	4.02	304.15
91	.095	2.46	26.53	333.23
92	.350	4.83	38.33	333.23
93	.340	4.61	37.10	982.44
94	1.389	9.64	62.31	982.44

POINT	PRESS.	TEMP.	ENTHAL.	FLOW
21	1.050	302.00	1583.34	4212.15
23	1.100	302.00	1583.34	4112.15
24	1.139	178.99	944.52	4112.15
26	1.166	151.69	802.78	4112.15
126	1.166	151.69	802.78	3468.61
127	1.243	64.90	351.89	3468.61
226	1.166	151.69	802.78	643.54
227	1.243	64.90	351.89	643.54
27	1.243	64.90	351.89	4112.15
29	1.294	38.93	216.76	4112.15
30	1.306	38.93	216.76	3097.97
31	1.318	22.60	131.43	3097.97
33	1.330	20.06	118.08	3097.97
133	1.330	20.06	118.08	1468.38
134	1.348	10.87	69.07	1468.38
233	1.330	20.06	118.08	1629.58
234	1.348	10.87	69.07	1629.58
34	1.348	10.87	69.07	3097.97
35	1.389	9.64	62.31	3097.97
135	1.389	9.64	62.31	2115.53
136	1.404	7.10	47.78	2115.53
36	1.404	7.10	47.78	386.50
37	1.411	4.60	29.54	386.50
100	4.550	3.80	9.57	4147.98
101	4.550	3.99	10.13	4103.98
102	4.550	3.99	10.13	4053.98
103	4.200	4.19	10.63	4053.98
104	4.200	4.19	10.63	4053.98
105	4.150	3.47	8.47	4053.98
106	5.450	3.79	10.01	4053.98
107	5.450	3.82	10.08	4153.98
108	5.400	3.37	8.91	4153.98
109	5.350	2.59	7.32	4153.98
110	5.000	2.90	7.68	4153.98

POINT	PRESS.	TEMP.	ENTHAL.	FLOW
45	16.227	185.00	980.61	658.27
46	9.000	153.64	815.31	658.27
47	8.904	69.37	376.53	658.27
48	8.860	69.37	376.53	1014.19
49	1.301	38.93	216.76	1014.19
52	15.579	25.00	139.41	1729.03
53	8.000	20.19	115.05	1729.03
54	7.922	12.38	69.03	1729.03
55	1.423	7.10	47.78	1729.03
67	8.904	69.37	376.53	355.91
68	9.666	69.35	376.53	355.91
69	15.666	40.00	222.00	355.91
71	15.494	6.29	23.59	100.00
72	5.550	6.34	23.59	100.00
73	5.450	4.71	13.14	100.00
74	4.975	3.35	8.63	6.00
75	4.550	3.80	9.57	44.00
76	4.550	3.99	10.13	50.00
77	1.050	302.00	1583.32	100.00

Figure 39. BNL ISABELLE Refrigerator. Ref. 43

highway transport, the tanks are 4.0 m diameter with lengths of 10.8 m to 13.8 m. Four tanks contain the main refrigerator, and the fifth the sub-cooler/circulation system.

4d. Isabelle Compressors[46]

The compressor alternates were thoroughly studied by both BNL and Koch Process Systems. Centrifugals were rejected on the basis of little helium operating experience, potential surge problems, high first cost and complex maintenance and repair needs. The centrifugal compressor is not positive displacement, adds energy thru momentum increase and can be subject to operating stability problems.

Lubricated reciprocating compressors were eliminated because of their high initial costs including foundation work, complex repair and maintenance, as well as long lead time spare parts. Recips are considered to have reduced availability, α, where α = MTBF/(MTBF+ MTTR). MTBF is mean time between failure and MTTR is mean time to repair. Even with isothermal efficiency, 0.67 - 0.72, and expected power savings, maintenance and outage arguments dominate for operations. Economic mass flow capacity reductions are difficult for both recips and centrifugals.

Rotary oil-flooded screw compressors were selected based on the arguments above. MTBF are fully expected to be in the 50,000 hour range. The machines selected are all the same size; 320 mm dia., L/D 1.65 rotors. Twenty (20) first stage units with 600 HP motors will compress from 1.05 atm to 4.5 atm; four (4) second stage, 2250 HP motors, 4.5 atm to 16.5 atm. One extra machine (2250 HP) will be redundant and can be used for either stage. It can also increase flow and/or change system pressure ratios. Each first stage compressor delivers about 240 g/s for a total mass flow of 4560 g/s. The specified minimum is 4400 g/s.

The compressors are products of Howden, Ltd. Glasgow, Scotland and will be packaged onto skid assemblies by Koch in Westborough, MA. The package includes motors, oil pumps, coolers and separators and some control valves. Special care for prevention of water to oil leaks and final oil removal to less than 1 ppm is required.

4e. Schedule and Summary

By January 1983, the cold boxes and turbines had been completed and final installation started at BNL, with four cold boxes placed. Compressor fabrication and testing was well advanced with final assembly scheduled end of 1983. Final magnet and accelerator design completion is now uncertain due to economics and new physics demands. BNL has had a full cell (6 dipoles and 2 quadrupoles) prototype operating in the tunnel since March 1983.

The cold laminated iron surrounding the magnets adds some structural support and tighter field coupling. The 6500 kg magnets require 48 hours to reach 3.8 K on the test stand and similar time for warm-up. This can limit magnet production rate. The iron thermal diffusivity is low. In the accelerator, cooldown of the 6×10^6 kg magnets was allowed two weeks (estimated 10 days).

The original cooling system in 1976 proposed pool-boiling with eight refrigerators around the ring. Further study and magnet requirements of 3.8 K developed the force cooled (supercritical) circulation system, with one large central plant. Single plant and operating costs would be less; piping, vessel size and LHe inventory were said to be reduced as well as control elements. Force cooling is a desirable method for magnet cooling, but if gravity is not used then input pump power must be carefully applied. Power additions at the lower temperatures are thermodynamically expensive. The 3.8 K requirement made supercritical GHe attractive.

Supercritical gas at 5 or more atm with almost constant density, 0.139 - 0.154 g/cc., was selected for the coolant. This density is slightly greater than LHe at 1 atm (0.121 g/cc). The system is designed for up to 15 atm and when the magnets quench and dump energy, the gas undergoes no phase change, i.e. liquid to gas and hence no great pressure rise. Stability of heat transfer and magnet operation is said to be very good. However, because of no available phase change and subsequent heat of vaporization transfer, the gas temperature must rise. For small temperature limits large amounts of gas must flow. Thus for Isabelle 4000 g/s is needed between 2.6K and 3.8K, almost the same as in the upper main refrigerator.

For a superconductor the lower the temperature the higher the magnetic field and critical current. The higher magnet field means a higher energy accelerator in a smaller orbit. Design pushed Isabelle operating temperature and band width as low as possible to maximize magnet and accelerator performance. This has extended the refrigerator design into very advanced state-of-the-art rotating helium equipment. BNL engineering talent has responded well and the present designs for the cold compressors and pumps by Sixsmith at Creare, Inc., New Hampshire, and Rotoflow, Inc. are very impressive.

Such a sophisticated low temperature sub-loop on the refrigerator opens the door-way to the next level of cooling systems. Superfluid He II, LHe below the lambda point, 1.8 K, with its unique mass and heat transport properties provides an ideal (though expensive) magnet coolant. Magnet field designs up to 10 Tesla become possible at 1.8 K. Thus BNL, with the existing 3.8 km circumference tunnel could exceed 1 Tev beam energies with 10 Tesla magnets.

Table 4e.1. Fermilab-Brookhaven Comparison

Item	Fermilab	Brookhaven
Ring dia/circum.	2.0 km - 6.28 km	1.2 km - 3.8 km
Ring energy	1000 GeV	400 GeV
Magnet peak field	4.2 Tesla	5.0 Tesla
Refrig. input power	CHL: 2500 kW, 2500 kW LN plant	12 MW
	24 satellite - 6000 kW = 11 MW	
Liquefier yield, L/hr	4000 L/hr + 24(120) = 6880 L/hr	11,000 L/hr
Refrig. yield, 4.5 K	12,000 W + 24(600) = 26.4 kW	
	or 24(996 W) = 23.2 kW	
Refrig. yield, 3.8 K		24.8 kW
LHe inventory, liters	50,000 est.	200,000 est.
Magnet mass flow	25-30 g/s two-phase thru 20 magnets	170 g/s super-critical thru 45 magnets

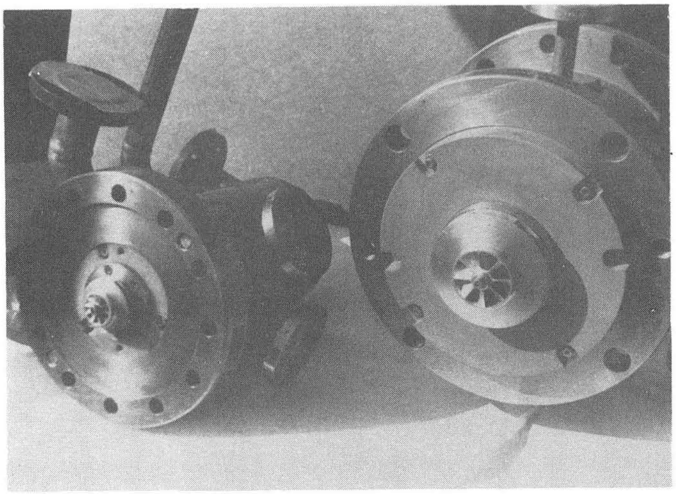


Photo. 42. (XBB 831-444). Two oil-bearing expander turbines for CBA helium refrigerator. No. 5 is on the left; 6.3 K inlet temperature, 10 kW work and 1.75 inch diameter, 40,000 RPM. No. 1 is on the right; inlet 185 K, 107 kW work, 4.5 inch diameter and 50,000 RPM. Note the shrouded impeller wheels. Variable inlet nozzles are provided. Photo courtesy Rotoflow Corp.

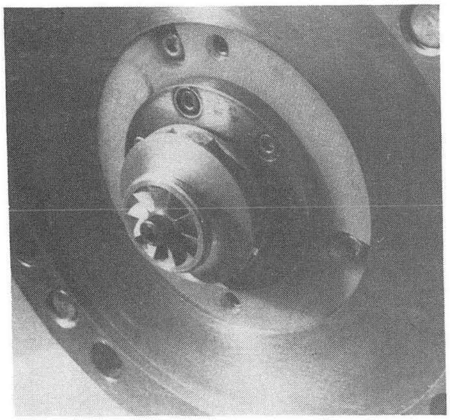


Photo. 43

(XBB 831-443). Close-up of CBA No. 5 turbine impeller wheel, 1.75 inch diam. Photo courtesy Rotoflow Corp.

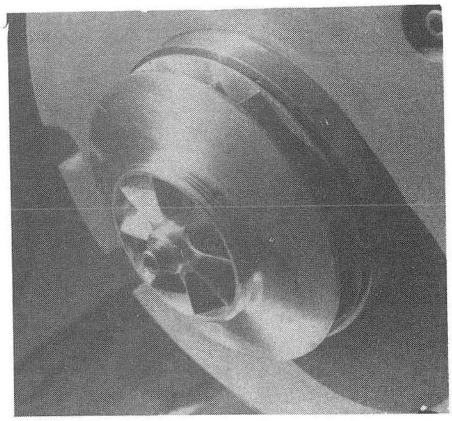


Photo. 44

(XBB 831-445). A Close-up of CBA No. 1 oil-bearing turbine impeller wheel, 4.5 inch diam. Photo courtesy Rotoflow Corp.

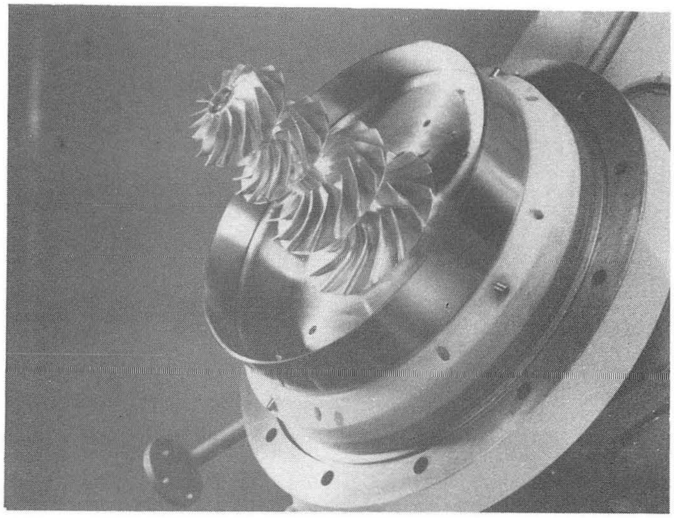


Photo. 45. (CBB 831-485). Close-up of CBA very low-temperature compressor number 1a,b and 2a,b. Wheel diameters are 4.3, 4.6; 6.00 and 6.88 inches. Photo courtesy Rotoflow Corp.

ISABELLE TURBOMACHINERY DESIGN VALUES[42]

EXPANDER	PRESS. (ATM) IN	PRESS. (ATM) OUT	TEMP. (°K) IN	TEMP. (°K) OUT	WORK kW	FLOW gm/sec	DIA. inch	SPEED 1000 rpm
1	16.2	9.0	185.0	154	107.2	648	4.5	48-50
2	8.1	1.3	69.5	30	160.5	1002	4.5	48-50
3	15.6	8.0	25.0	20	38.8	1590	2.63	48-50
4	7.0	1.4	12.4	7.1	33.8	1590	3.0	39-44
5	15.4	2.5	6.3	5.1	10.0	1304	1.75	40
COMPRESSOR								
1A 1B	.095	.350	2.46	4.7	3.6	324	4.3 4.6	18-20
2A 2B	.340	1.30	4.56	9.2	21.6	945	6.0 6.88	18-20
3	4.20	5.45	3.47	3.76	5.8	4054	4.1	10-11.5

5.0 Refrigeration Equipment Cost

A major involvement with budgets and costs is a primary engineering function. Thus a description of equipment is incomplete without the price tag. Price and cost have different meanings for seller and buyer.

In general plants are purchased by commercial and research labs as turn-key units, completely installed. Competitive bidding can result in optimum low price. Specialist fabricators are more efficient and costs to the buyer are fixed before the job start. In other cases, the research funds are limited and single units are purchased and installed by the buyer, particularly where systems are considered research and development. Usually after compressors and cold boxes are purchased under contract, installation will multiply final costs by a factor of two or three. This depends greatly on power and building facilities availability, cooling systems, and liquid and gas storage and supply. Price for plants described indicate the range. Stiff competition as well as engineering work already done has strong effect on price.

For the 1500 W refrigerator two cold boxes and one compressor system were ordered in 1975 for $1,075,000 ($395,000 for the LBL cold box) LBL purchased two screw compressors for $90,000 and did their own installation including oil-removal design. In 1976, ORNL ordered their first plant, including cold box and the two compressor skid (1400 HP) for about $1,000,000.

The 3000 W plant at LLNL is part of a large turn-key contract but estimates for the cold box and the three compressor skid (2000 HP) are $1,200,000 (1978).

The Fermilab CHL, cold box contract in 1976 was about $1,000,000. Three 4000 HP compressors were obtained surplus, rebuilt, and installed. An estimated value, 1982, would be 1.5 to 2 M$ each, and installed 10 to 15 M$ for three.

For the Fermi satellite, the HX package estimate is $100,000 and each engine $25,000 for total cold box cost of 150 k$ (1980). The two-stage Mycom compressor price (1982) is about $100,000. Oil clean-up equipment is $30,000. LBL purchased three satellite types from Cryogenic Consultants, Inc. The first included cold box, two engines and a single stage compressor with oil clean-up, $300,000 in 1977. In 1980 LBL purchased two more cold boxes for $150,000 each. Compressors were $50,000 for single stage units. Controls and oil clean-up were added by LBL.

The BNL Isabelle refrigerator included design, construction, installation and testing for 6.6 M$. The package includes five cold box tanks and about 12 turbines from Rotoflow for a 1 M$ sub-contract (1980). The compressor contract included design thru tests for 25 compressors, controls and oil handling for 11,3 M$ (1982).

From these numbers we can see that costs of compressors are less or equal to cold box costs for small units. As size increases, mass flow is dominant and compressor costs can grow to two or more times cold box costs.

TABLE
Escalation factors from a DOE high energy physics construction cost summary, Sept. 1982

FY Reference	70	71	72	73	74	75	76	77	78	79	80	81	82	83
Yearly cost Inflation, %	0.9	7.1	6.8	12.9	9.0	10.7	10.8	8.6	11.0	10.2	10.5	9.1	6.8	
Cumulative Scaling factor Converts to FY 83	3.28	2.95	2.76	2.58	2.29	2.10	1.90	1.71	1.58	1.42	1.29	1.17	1.07	1.00

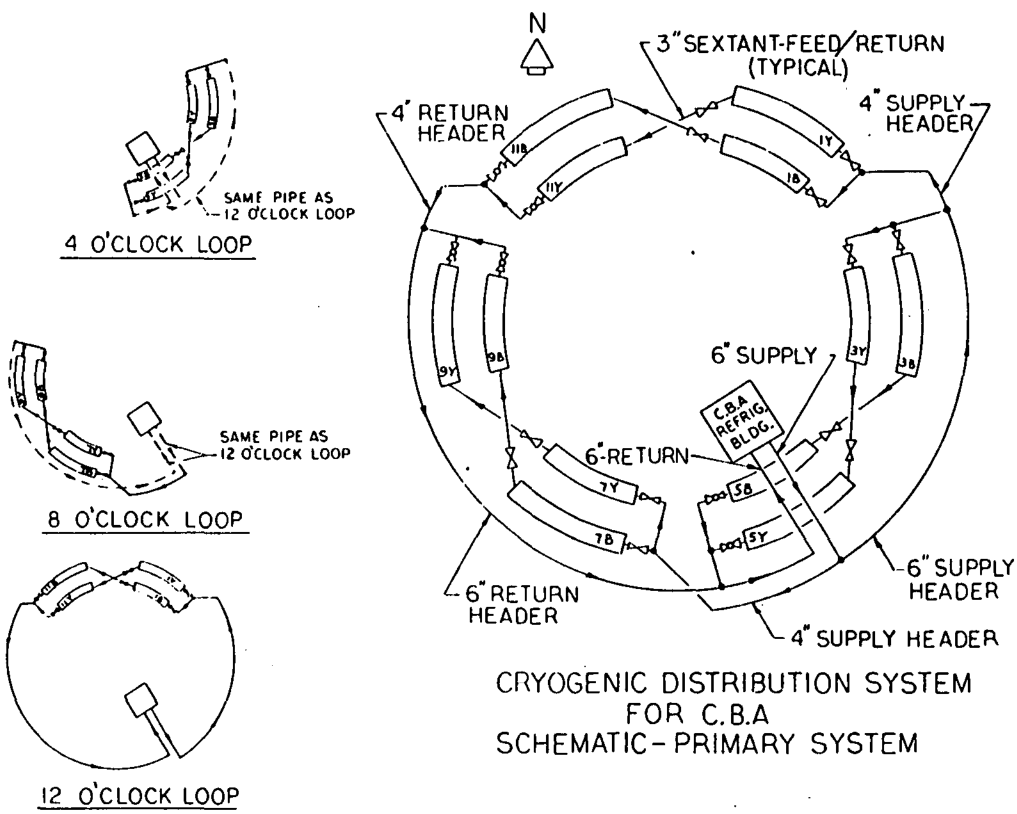


Figure 34b. Most recent schematic of CBA distribution system.

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TECHNICAL INFORMATION DEPARTMENT
LAWRENCE BERKELEY LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY, CALIFORNIA 94720