REFRIGERATION AT 4°K*[1]

T.R. Strobridge
Cryogenics Division, Institute for Basic Standards
National Bureau of Standards
Boulder, Colorado

I. INTRODUCTION

Refrigeration in the region of 4°K will be necessary for the subject superconducting devices until practical materials with higher transition temperatures become a reality. Even though helium was discovered in the sun's atmosphere one hundred years ago and was identified on earth in 1895, refrigeration technology progressed slowly until the middle of the 1940's. Today, 4°K refrigerators can be obtained in capacities ranging from 1 W to several kilowatts. The scientific literature shows that much early low temperature research was devoted to liquefying the more difficult of the normally gaseous elements. Liquefaction of the lowest temperature fluid, helium, was finally accomplished by H. Kamerlingh Onnes on July 10, 1908.[2] Onnes was awarded the Nobel prize for this contribution to the technology. Many of the succeeding experiments were concerned with the behavior of helium itself and led to the discovery of the lambda transition and the unique, interesting properties of liquid helium II. Investigations rapidly branched out to other materials and superconductivity was observed in mercury by Onnes in April of 1911.[3] Reference 1 is devoted to the refrigeration methods used today — evaporating liquid helium baths, the Simon process, the Joule-Thomson, Brayton, Claude, Stirling and Gifford-McMahon cycles.

II. THERMODYNAMIC RELATIONSHIPS

The behavior of refrigerators and their components are governed by the laws of thermodynamics. The first law of thermodynamics relates the mass and energy fluxes that occur in natural events. It is used to predict the outcome of the various processes that make up a refrigeration cycle. The first law as given in Eq. (1) and illustrated in Fig. 1 shows that modern nomenclature has clarified the concept of a control volume and a thermodynamic system:

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1. The material on refrigeration presented to the cryogenics session of the Summer Study on Superconducting Devices and Accelerators was selected from:

   T.R. Strobridge, in The Technology of Liquid Helium, published as NBS Monograph 111 (1968).

   This chapter will be available about the same time that these Proceedings will appear. Therefore, a summary is given here and the reader is referred to Ref. 1 for detail.

2. H. Kamerlingh Onnes, Communs. Phys. Lab., Univ. Leiden, No. 108 (1908).

3. H. Kamerlingh Onnes, Communs. Phys. Lab., Univ. Leiden, No. 1226 (1911).

$$Q_{c.v.} + \sum m_i \left(h_i + \frac{v_i^2}{2g_c} + z_i \frac{g}{g_c} \right) = W_{c.v.} + \sum m_e \left(h_e + \frac{v_e^2}{2g_c} + z_e \frac{g}{g_c} \right)$$

$$+ \left[m_2 \left(u_2 + \frac{v_2^2}{2g_c} + z_2 \frac{g}{g_c} \right) - m_1 \left(u_1 + \frac{v_1^2}{2g_c} + z_1 \frac{g}{g_c} \right) \right]_{c.v.} \quad . \qquad (1)$$

The terms $Q_{c.v.}$ and $W_{c.v.}$ are the heat and work crossing the control surface, m is mass, h is specific enthalpy, u is specific internal energy, V is velocity, and Z is vertical position. The subscripts i and e refer to the incoming and exiting flows, and 1 and 2 refer to states 1 and 2 in the control volume indicated by the subscript c.v. The proportionality constant that relates force, mass, time, and length in Newton's second law is g_c and the local acceleration of gravity is g. (In the SI system, $g_c = 1$ kg·m/N·sec^2.) The convention is adopted that heat transferred to the system is considered positive as is work done by the system.

The second law of thermodynamics leads to the definition of entropy and is the basis for the concepts of the thermodynamic temperature scale, the Carnot cycle, reversibility, and irreversibility. As stated, the first law predicts the outcome of an assumed process. However, the first law holds equally well for the process proceeding in the reverse direction. The permissible directions for processes are given by the second law.

The second law is useful for rating the performance of processes and cycles. Conventionally, the thermal efficiency of an actual refrigerator is compared to that of a Carnot refrigerator operating between the same temperature levels. The ratio of the amount of power required by a Carnot machine to the refrigeration produced is given by

$$\frac{W_c}{Q_{Carnot}} = \frac{T_o - T}{T} \quad , \qquad (2)$$

where W_c is the net power, Q is the refrigeration produced, T is the temperature, on the thermodynamic scale, at which the cycle is producing refrigeration, while T_o is the temperature at which heat is being rejected by the machine — usually about 300°K, the nominal temperature of the earth's atmosphere or cooling water supply. Thus, for a helium temperature refrigerator operating between 4.2 and 300°K, the best performance that could be achieved is 70.4 units of power required per unit of refrigeration. As will be seen later, it is not unusual for an actual machine to consume 100 times the Carnot power per unit of usable refrigeration. The efficiencies of various processes are also calculated by comparing them with appropriate, assumed, reversible processes.

III. REFRIGERATION CYCLES AND METHODS

1. Liquid Helium Baths

By far the most common method of providing cooling at 4.2°K is by using liquid helium, supplied from a remote liquefier, as the heat sink. Cryostat designs vary widely to accommodate the apparatus or sample that is to be cooled. However, care must be taken to make sure that the vessel can be efficiently cooled down and filled. Adequate pressure relief must be provided and the vent system must prevent atmospheric gases from entering the cold space. The cold saturated helium vapor leaving the vessel has a great deal of refrigeration potential and in some instances the vent pipe is

attached to intermediate temperature shields which reduce the amount of heat transfer from the surroundings to the low temperature space, thus decreasing the liquid evaporation rate. The heat of vaporization of liquid helium is 0.725 Wh/liter at 1 atm pressure. Thus a one watt heat load will evaporate 1.38 liters of liquid per hour and large heat loads require substantial amounts of the relatively expensive liquid. Each new cooling requirement forces a choice between refrigerating with bulk liquid helium or a closed-cycle refrigerator.

2. The Simon Process — Adiabatic Expansion

In the years before the development of the Collins liquefier, the Simon process was often used to batch-produce small amounts of liquid helium. When a heavy walled container filled with high pressure gas is allowed to vent, the fluid in the container expands isentropically and its temperature is accordingly depressed as the pressure is lowered. Although the expansion process is isentropic, as the fluid cools there will be heat transferred to it from the warmer walls of the vessel resulting in an entropy increase of the fluid. Fortunately, at liquid helium temperatures the specific heat of most container materials is very low and such entropy contributions may be negligible. In practice, the high pressure helium and the container are cooled to at least 20°K prior to the expansion process to increase the liquid yield. The Simon process has recently been used to refrigerate sodium magnet coils at approximately 7°K by the Lawrence Radiation Laboratory, Livermore, California.[4]

3. The Joule-Thomson Process

The lowest temperature stage in virtually all liquid helium temperature refrigerators consists of a counter flow heat exchanger, an expansion valve, and an evaporator. The Joule-Thomson process is defined as the adiabatic isenthalpic reduction in pressure of a fluid flowing through a restriction in a passage. This process takes place only in the expansion valve mentioned above, but its relationship with the energy transferred in the heat exchanger and the evaporator is important. Consider steady state operation of the heat exchanger, expansion valve and evaporator (where the heat load is absorbed) shown in Fig. 2. The saturated liquid in the evaporator changes phase to saturated vapor as heat enters the system from the load. The cold vapor enters the low pressure side of the heat exchanger where energy is transferred to it from the warmer high pressure stream. The high pressure stream enters the warm end of the heat exchanger at a temperature slightly higher than the exiting low pressure stream and is cooled in counterflow heat exchange with the low pressure stream. Expansion through the valve reduces the pressure and temperature of the fluid so that a part of the helium is liquefied. The fraction of the flow that is not liquefied in the expansion process joins the vapor entering the low pressure side of the heat exchanger. Thus the liquid in the evaporator is continuously being depleted by the heat load and supplied at the same rate by the expansion valve. The refrigeration or cooling effect is isothermal because it is obtained by evaporating a saturated liquid and the refrigeration temperature is controlled by the pressure in the evaporator.

Whether a fluid will heat or cool upon expansion in the Joule-Thomson process depends upon the properties of the fluid and the pressure and temperature prior to the expansion. For example, at room temperature, nitrogen will cool in the Joule-Thomson process but helium and hydrogen will heat. Therefore, the system illustrated in Fig. 2 can be made into a closed cycle nitrogen temperature Joule-Thomson refrigerator by supplying a compressor between stations 1 and 5, an insulating enclosure

4. C.E. Taylor, in Proc. 1965 Symposium on Engineering Problems of Controlled Thermonuclear Research, p. 152.

and the appropriate controls. Because of the heating effect in helium at higher temperatures, a helium refrigerator must have additional cooling means such that the temperature of the gas entering the high pressure side of the low temperature heat exchanger is below about 55°K (the maximum inversion temperature) and preferably much lower. The way in which this precooling is accomplished accounts for the differences in the thermodynamic cycles used for helium temperature refrigerators.

The amount of power required to produce a given amount of refrigeration at a given temperature is always of interest to a potential user for two reasons. First, very low temperature refrigerators require high input power relative to the cooling capacity — operating costs are obviously affected. Second, the capital cost of such equipment is proportional to the installed drive power so low efficiencies mean increases in both purchase price and operating expense. The input power to the Joule-Thomson circuit required per unit of refrigeration decreases for lower precooling temperatures (T_1 in Fig. 2) and better heat exchanger efficiencies as evidenced by smaller temperature differences between the fluid streams at the warmer end of the heat exchanger ($T_1 - T_5$). If a cycle is characterized by a certain precooling temperature and heat exchanger temperature difference, then there is only one high pressure that will give optimum performance, i.e., minimum input power (W_c) per unit of refrigeration (Q). Dean and Mann[5] have made extensive calculations of the performance of Joule-Thomson refrigerators for low temperatures. Figure 3 is a skeleton diagram illustrating their results. The plot is for a constant heat exchanger temperature difference and shows the input power per unit of refrigeration as a function of the high pressure, for various precooling temperatures. Graphs for other heat exchanger temperature differences form a family of performance surfaces. The line marked optimum is the locus of minima in the curves of constant precooling temperature, the inversion curve is superimposed and the mass flow rate per unit of refrigeration is given. Below and to the right of the line marked second law violation is a region in which the properties of the helium refrigerant will not permit a positive temperature difference between the warm and cold streams throughout the heat exchanger with the assumed precooling temperature and temperature differential ($T_1 - T_5$). The series of performance curves[5] show the effect on performance of changes in the irreversibilities of the heat exchanger, in the precooling temperature, and in the level of the high pressure of the fluid entering the heat exchanger. It must be remembered that the power requirements are for the helium Joule-Thomson circuit only and that additional power will be needed to precool the helium to the inlet temperature of the heat exchanger.

4. The Brayton and Claude Cycles

The cascaded Joule-Thomson helium refrigerator has a nitrogen circuit which precools a hydrogen circuit at about 65°K which in turn precools the helium circuit at about 15°K. The same energy removal necessary to depress the temperature of the helium at the inlet of the final heat exchanger to an acceptable level is often accomplished by using the refrigeration produced by one or more mechanical expanders of either the reciprocating or turbo-machinery varieties. A complete study of the Brayton cycle has been made[6] and the combination of the Brayton cycle with the Joule-Thomson process, known as the Claude cycle, has also been the subject of a complete analysis.[7] The schematic of the Claude cycle in Fig. 4 shows the heat absorber or evaporator thermally connected to the heat load Q. Heat exchangers III and IV and the two expansion

5. J.W. Dean and D.B. Mann, NBS Technical Note No. 227 (1965).

6. R.C. Muhlenhaupt and T.R. Strobridge, NBS Technical Note No. 366 (1968).

7. R.C. Muhlenhaupt and T.R. Strobridge, NBS Technical Note No. 354 (1967).

valves comprise the lowest temperature stages discussed in connection with the Joule-Thomson process. The second expansion valve between stations 4 and 5 is used at times to adjust the temperature profile in the heat exchanger and avoid the region of second law violation shown in Fig. 3. If the expansion valve between stations 4 and 5 is not necessary then heat exchangers III and IV are one unit and the arrangement below stations 3 and 10 is the same as in Fig. 2. In all of the heat exchangers, the low pressure gas returning to the compressor is used to cool progressively the high pressure helium moving toward the low temperature end of the refrigerators. Starting at station 1, the high pressure gas is cooled in HX-I. At station 2, a portion, $\dot{m}_2$, of the mass flow through the compressor, $\dot{m}_1$, is diverted through the expansion engine. The temperature of the helium decreases as the gas gives up energy by doing work on a piston or turbine blades in the expander (the energy is mechanically or electrically transmitted through the insulating cryostat to the surroundings). The low pressure exhaust of the expander joins the process stream returning to the room temperature compressor at station 10. The refrigeration produced by the mechanical expander is thus made available to precool the high pressure helium flowing toward the low temperature heat exchanger and expansion valve(s). The compressor effectively handles two process streams; the Brayton precooling cycle process path is 1-2-10-12 through the expander and upper heat exchangers, and the path to the low temperature stage is straight through all the heat exchangers. While the two streams can be thought of independently, they are in fact intimately mixed in the compressor, HX-I, and the low pressure side of HX-II.

For any given set of heat exchanger and expansion engine efficiencies and high pressure, there is an optimum engine inlet temperature in terms of over-all cycle performance. In addition, there is an optimum high pressure. The results of Ref. 7 provide a tool for the optimization of the Claude cycle over a wide range of parameters.

5. The Stirling Cycle

Low temperature refrigerators operating on the periodic flow Stirling cycle, incorporating one or more regenerators, lend themselves to compact construction with one piston serving for both compression and expansion in a single stage unit. For a thorough discussion of the Stirling cycle itself, see Ref. 8. Stirling cycle refrigerators are not in themselves suitable for refrigeration at liquid helium temperatures because of the rapidly decreasing specific heat of the regenerator materials at low temperatures. The lowest temperatures reported for pure Stirling cycle two-stage refrigerators are about $12^{\circ}K$. However, helium Joule-Thomson circuits are effectively precooled by this type of refrigerator. The precooling stations at two separate temperature levels are shown in Fig. 5, which is a schematic of such a refrigerator suggested by Rietdijk.[9] Energy is transferred from the helium to the Stirling cycle refrigerator through heat exchangers attached to the cold heads of the cooler. The configuration shown is analogous to a two expander Claude cycle. After leaving the lowest temperature precooling station at about $15^{\circ}K$, the helium enters the high pressure side of the Joule-Thomson heat exchanger. An expansion valve and evaporator at the low temperature end of that exchanger would complete the cycle and provide refrigeration at about $4^{\circ}K$ if the compressor suction pressure were at one atmosphere. A rather more elegant arrangement is shown in Fig. 5 which allows the compressor suction pressure to be above the vapor pressure of the helium at the refrigeration temperature. The refrigerant is held at three different pressure levels by an expansion ejector. Full compressor flow at the highest pressure enters the nozzle of the ejector where it is accelerated to

8. G. Prast, Philips Tech. Rev. 26, 1 (1965).

9. J.A. Rietdijk, in Pure and Applied Cryogenics (Pergamon Press, 1966), Vol. 6, p. 241.

high velocities with a corresponding decrease in pressure. This region of low pressure induces flow, $\dot{m}_2$, from the low temperature evaporator through the low pressure side of the final heat exchanger. The two flow streams mix and, as the velocity is decreased, experience a rise in pressure to P_2. A fraction of the fluid emerging from the ejector may be liquefied. The stream is divided with $\dot{m}_1$ returning to the compressor through the low pressure side of the heat exchangers and $\dot{m}_2$ is diverted through the high pressure side of the lowest temperature heat exchanger to an expansion valve and the evaporator. The cooling capacity of a Joule-Thomson system depends only upon the mass flow rate and enthalpy difference at the warm end of the Joule-Thomson heat exchanger and is independent of the refrigeration temperature. Therefore, the expansion ejector can offer advantages in two ways. First, lower refrigeration temperatures can be achieved without any decrease in compressor suction pressure; second, cooling may be produced at $4^\circ K$ with a compressor suction pressure in excess of two atmospheres. Savings in compressor and heat exchanger size are made in either case. Haisma[10] reports refrigeration temperatures as low as $1.75^\circ K$ with P_1 at 31 atm and P_2 at 1.15 atm. A production model of this type of refrigerator is not ready as yet, but a 10 liter/h helium liquefier featuring a noncontaminating rolling diaphragm sealed compressor, heat exchangers laminated from metal gauze, paper and resin, and an expansion ejector which permits a 2.5 atm compressor suction pressure has been marketed in Europe (Haarhuis[11]). It should be noted that several thousand Stirling cycle refrigerators are in the field in the United States today for cooling electronic equipment at temperatures higher than liquid helium.

6. The Gifford-McMahon Cycle

By itself the Gifford-McMahon refrigeration cycle has about the same low temperature limit as the Stirling cycle since there is periodic reversing flow through one or more regenerators. However, this cycle lends itself very readily to staging and refrigerators precooled at three different temperature levels have been built for liquid helium temperature service. Figure 6 schematically shows a three-stage unit. The three cylinders are made of low conductivity material and are fitted with displacers, sealed against the cylinders at the top, whose movement is controlled from outside the cryostat. The three thermal regenerators are indicated as are the three heat absorption stations at 14, 35 and $80^\circ K$. The regenerators are packed with a material such as metal screens to produce the desirable characteristics of high heat capacity per unit volume, low pressure drop, small void volume, and low axial thermal conductivity. Imagine now that all three displacers are in their lowest position and that steady-state operating conditions have been reached so that the proper thermal gradient exists in the regenerator material from room temperature to $80^\circ K$ in the first regenerator. The compressed gas supply valve is opened, and helium enters the system raising the pressure in the volume at the top of the cylinders above the displacers. When the pressure in the regenerators and the volume above the displacers reaches the maximum pressure, the displacers are all moved to the upper position, forcing the gas above them through the regenerators and into the volumes which now appear below the displacers. The gas is cooled as it gives up heat to the cold regenerators and the decrease in specific volume allows more fluid to enter the system from the compressed gas supply. The supply valve is closed and the exhaust valve is slowly opened. Refrigeration is now produced as each of the elements of fluid do work on preceding elements as they pass out of the system. The fluid is warmed as it passes through the regenerators and through the heat

10. J. Haisma and K. Roozendal, in Proc. 12th Meeting of Commission 1 of the Intern. Inst. of Refrigeration, Madrid, 1967 (to be published), Paper No. 1.37.

11. G.J. Haarhuis, in Proc. 12th Meeting of Commission 1 of the International Inst. of Refrigeration, Madrid, 1967 (to be published), Paper No. 1.36.

absorption stations. When the pressure in the system reaches its lowest level the displacers are moved to the lower position forcing the remaining cold gas out through the regenerators and to the compressor suction. At this time the temperature distribution in the regenerators has returned to its original profile. This refrigerator inherently has several mechanical advantages. Most of the low temperature apparatus is passive except the displacers which are shown as sealed at room temperature. In later models, the displacers are placed in-line and require seals at low temperatures, but the sealing problem is not serious since the pressure difference across the seals is only that which is required to induce flow through the regenerators. The displacers are operated at low speed, typically less than 100 rpm, and the light duty drive mechanism is used to operate the room temperature inlet and exhaust valves. Difficulties encountered with contaminants in the refrigerant are alleviated somewhat since the exhaust phase of the cycle tends to flush impurities back out of the system.

IV. MODERN COMPONENTS AND REFRIGERATORS

Many helium temperature refrigerators which have recently been placed in service feature complete separation of the process gases from petroleum based lubricants to eliminate fouling of the heat exchangers or regenerators and to prevent freezing of the lubricant in other critical areas. Noncontaminating reciprocating compressors can have plastic piston rings, metal diaphragms, or labyrinth seals instead of piston rings. High speed turbine compressors have been developed which have process gas lubricated bearings supporting the shaft. Gas lubricated bearings are also used for high speed turbo-expanders and a variety of reciprocating expanders are operating today. Efforts to improve heat exchanger performance have lead to a proliferation of different designs. Tubes finned both inside and out are extensively used to increase the effective heat transfer surface as are the extended surfaces employed in plate fin exchangers. Accurate prediction of the characteristics of a new heat exchanger design is very difficult and usually heat transfer coefficients and pressure drop friction factors are determined from test models.

Cascaded Joule-Thomson, single and multiple expander Claude, Stirling and Gifford-McMahon cycle 4°K refrigerators are being marketed today. Refrigeration capacities range from about 1 W up to more than 1 kW. Much larger units could be designed and fabricated if necessary. Since dynamic machinery is subject to failure, the reliability of the refrigeration system can be enhanced by providing redundancy in critical areas if continuous operation is required. Modern refrigerator efficiencies range from about 1% of Carnot for capacities of about 1 W to about 16% of Carnot for kilowatt capacity refrigerators. This means that the drive power required per unit of refrigeration ranges from about 10 000 to 500, respectively. The thermal efficiencies at all capacities could be improved, but the refrigerators would be more complex. The capital cost would probably increase to reflect development costs and complexity unless there were a corresponding increase in the size of the market which would lower manufacturing costs.

The most difficult problem in cooling a superconducting accelerator or its supporting superconducting devices will not arise because of a deficiency in refrigeration technology; rather, it is the geometry of the accelerator itself that poses the problem. The total cooling loads suggested are not unusually high, but when it is understood that refrigeration is required in small amounts at widespread locations, then the problem assumes its proper magnitude. A number of small refrigerators could be arranged around the accelerator. Here the disadvantages would be higher capital and operating costs per watt of refrigeration than for one equivalent large refrigerator, and there would be more moving members subject to mechanical failure. On the other hand, a large refrigerator would require a refrigerant distribution system to all parts of the accelerator. This low temperature piping is expensive and the heat leak would increase the refrigeration capacity required. The answer is not at all clear at this time and the problem merits thorough study.
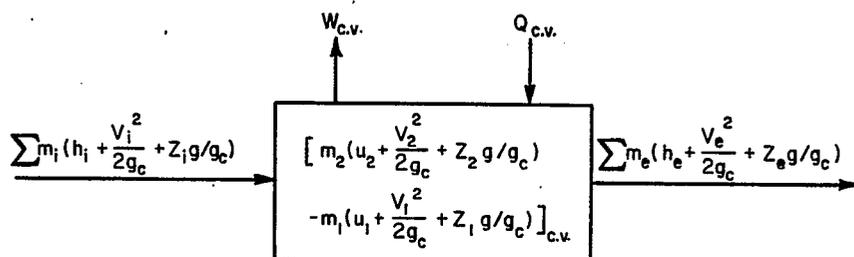
Fig. 1. Pictorial description of first law of thermodynamics for uniform
         state uniform flow process.
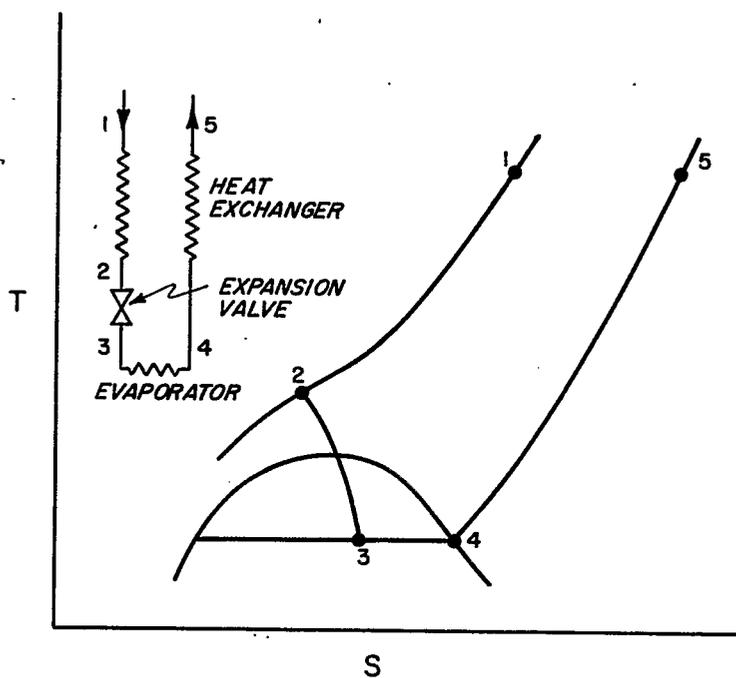
Fig. 2. Schematic of a Joule-Thomson refrigeration system.
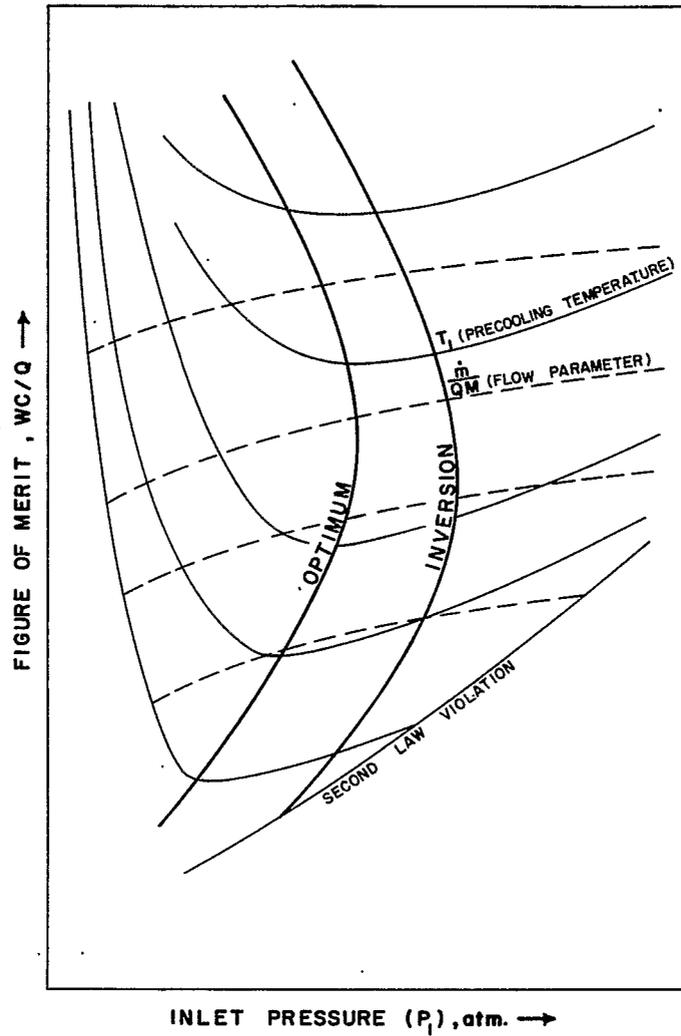
Fig. 3.   Generalized performance surface for a Joule-Thomson refrigeration
system.
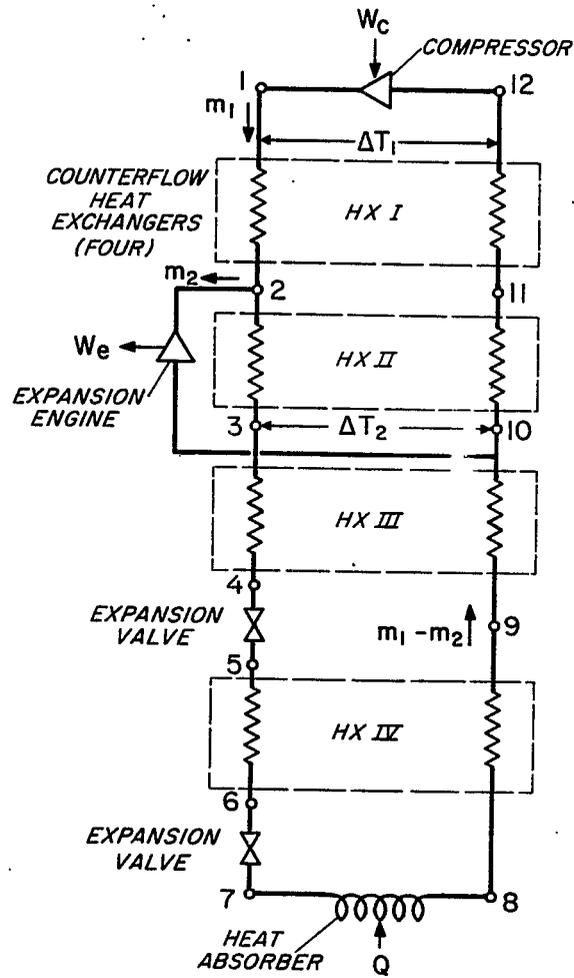
Fig. 4. Schematic of single-engine Claude refrigerator.
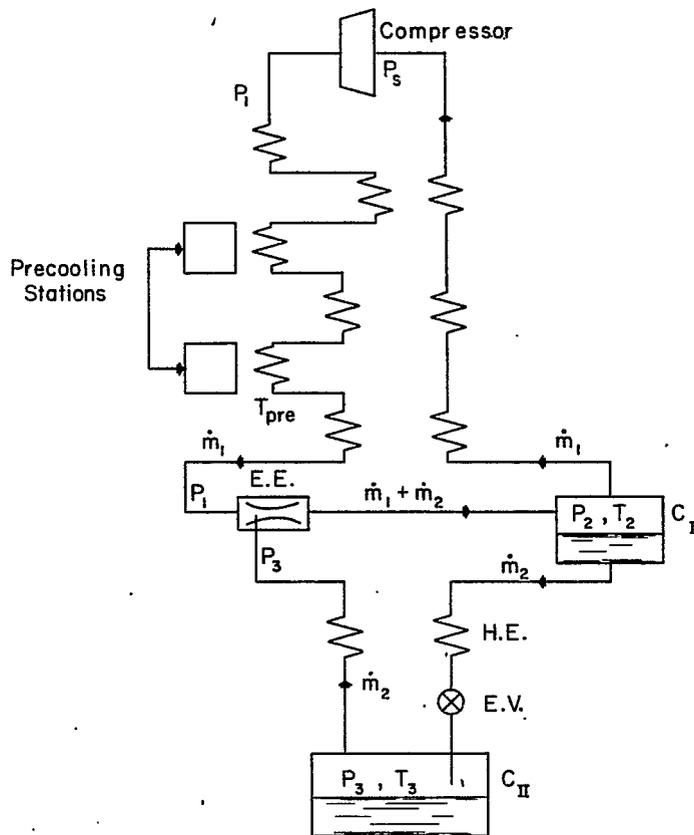
Fig. 5.   Schematic of Stirling precooled refrigerator with an expansion
          ejector.

COMPRESSED GAS SUPPLY

EXPANDED GAS RETURN

CYLINDER

DISPLACER

80°K

35°K

REGENERATOR

14°K

HEAT ABSORPTION STATION
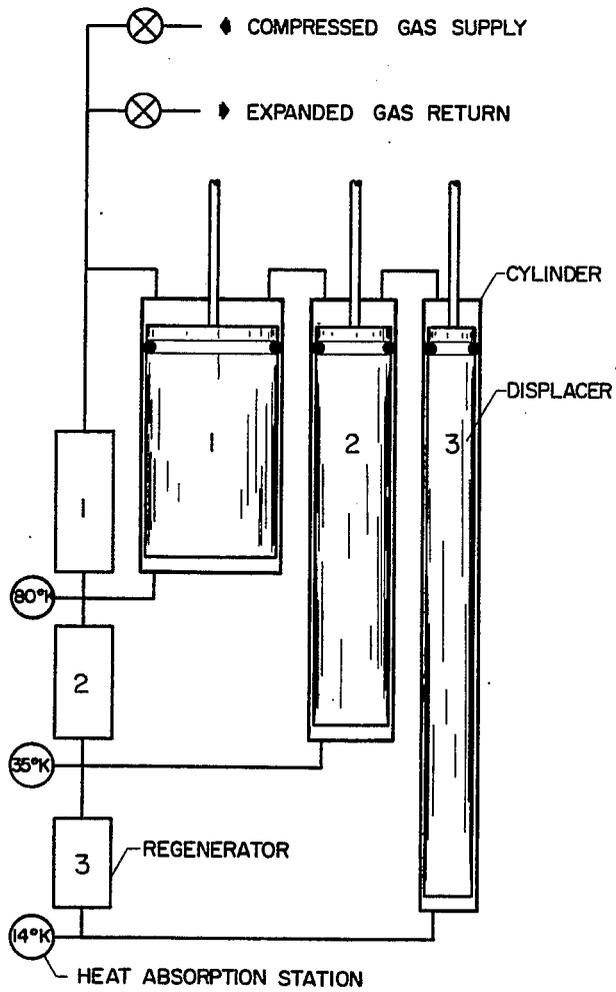
Fig. 6.   Schematic of Gifford-McMahon cycle.