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CLOSED-CYCLE HELIUM REFRIGERATION

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INTRODUCTION

THE practicability and range of applications of superconductive circuits appear likely eventually to be determined by the ease and reliability of maintaining the required low temperature. Present knowledge indicates that a temperature near the boiling point of liquid helium, or perhaps a little lower, will be required. Since there is no known means for maintaining such a temperature other than by compression and expansion of helium, the problem is reduced to the question of the feasibility of constructing a mechanical closed-cycle helium refrigerator of sufficient reliability.

In order to liquefy helium it is necessary to compress the gas to approximately 200–300 lb/in^2, cool it to approximately 15°K or colder, and expand it through a Joule–Thomson cycle to one atmosphere or less. All liquefiers or refrigerators must necessarily have this feature in common. Refrigerators differ from one another according to the way in which they accomplish the initial cooling from 300 to 15°K. The classical method, and the one first used to liquefy helium,[1] uses liquid ammonia, sulphur dioxide or Freon, liquid nitrogen and liquid hydrogen to accomplish the necessary precooling. This of course leads to a complex, bulky machine having low efficiency, poor reliability and with very poor possibility of development in the direction of simplicity and self-sufficiency. A considerably improved method of providing precooling was developed by COLLINS and has led to the standard ADL-Collins laboratory liquefier[2,3]. In this system, portions of compressed helium gas are expanded in expansion engines at intermediate temperatures to produce the refrigeration required for cooling the Joule–Thomson stream. It is possible that the COLLINS system could provide a basis for the development of a refrigerator having the required high reliability, but it appears that there may be formidable problems connected with the refinement of the expansion engines and valve mechanisms. Moreover, the COLLINS system carries the entire helium flow through countercurrent heat exchangers, which are necessarily more bulky and costly than reciprocating-flow thermal regenerators.

A new engine-expansion system has been developed[4,5] which utilizes reciprocating flow through a thermal regenerator instead of counterflow heat exchangers. This feature permits the engine valves to be operated at room temperatures, thereby eliminating many of the mechanical problems associated with them. A distinctive feature of the new expansion engine is that enthalpy is ejected from the system in the form of heat rather than as mechanical work. This eliminates the necessity of any large mechanical forces and allows for simplicity of design and high reliability.

THE SINGLE-STAGE CYCLE

A single-stage refrigerator utilizing the new cycle is shown schematically in Fig. 1. It consists of a thin-wall stainless-steel cylinder into which is

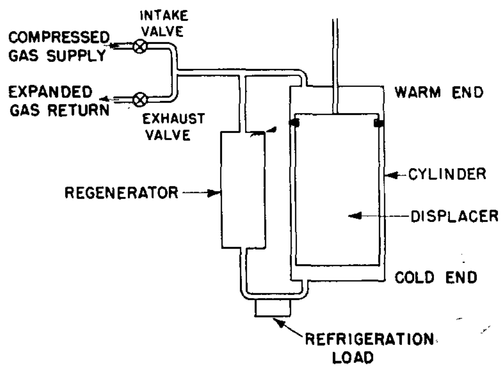


FIG. 1. Single-stage refrigerator.

fitted a displacer in the form of a fairly closely fitting plastic piston which is free to move longitudinally throughout a limited distance. A rod is attached to the room-temperature end of the displacer and is used to control its motion. The other end of the displacer–cylinder combination is at the low temperature. As the displacer is moved, the warm and cold volumes at the two ends of the cylinder are varied, one increasing at the expense of the other. However, these two volumes are connected together through a thermal regenerator having relatively low resistance to gas flow. Consequently, the pressure above and below the displacer is always approximately equal; and no appreciable force is required to move it.

The thermal regenerator consists of a thin-wall stainless-steel tube filled with circles of fine metal screen which are stacked to form a cylindrical structure just equal to the internal dimensions of the regenerator shell. The upper end of the regenerator is at approximately room temperature and the other end is at the low temperature. In spite of the large temperature gradient along the length of the regenerator, there is very little heat flow because of the poor thermal conductivity of the stacked structure.

The intake and exhaust valves, which are located at room temperature, are operated by cams attached to a crank shaft which controls the motion of the displacer. Compressed gas is supplied to the refrigerator from a more or less conventional closed-cycle compression system consisting of a reciprocating two-stage compressor and the usual ballast chambers and coolers.

The cycle may be considered to be made up of four operations: *Pressurization, Intake, Expansion* and *Exhaust*. At the beginning of the pressurization the displacer is at the extreme cold end of the cylinder so that the cold volume is a minimum and the warm volume is a maximum. The intake valve is opened, and compressed gas rushes into the warm volume (and also the free volume of the regenerator) until the pressure is equal to that of the supply. The sudden increase of pressure in the warm volume, of course, causes adiabatic heating and results in a considerable temperature rise of the gas.

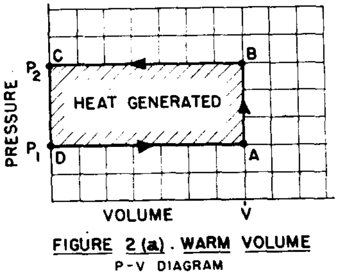
With the intake valve remaining open, the displacer is moved so as to enlarge the cold volume and diminish the warm volume. This displaces the heated gas through the regenerator to the cold end causing it to decrease in volume by an amount proportional to the ratio of the absolute temperatures at the two ends. Since the intake valve is open during the operation, the pressure is constant; and more gas flows in through the intake valve and mixes with the displaced stream entering the warm end of the regenerator. At the end of the intake stroke the cold volume has reached its maximum value and is charged with cold gas at the full intake pressure. The intake valve is closed at the end of the intake stroke.

The expansion operation is carried out by opening the exhaust valve slowly over a period of time when the displacer is essentially motionless. If the expansion occurs too abruptly, poor heat transfer will result between the exhausting gas and the metal matrix of the regenerator. As expansion occurs the gas remaining in the cold volume falls to a progressively lower temperature. The gas leaving the cold volume is partially warmed by heat removal from the working temperature (the refrigeration load in Fig. 1) and is subsequently heated all the way to room temperature by passage through the regenerator.

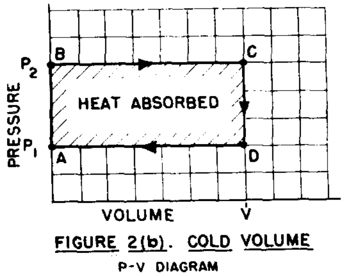
The cycle is completed by moving the displacer so as to expel the cold expanded gas remaining in the cold volume at the end of the expansion operation. The exhaust valve remains open as the displacer is lowered and is closed after the exhaust stroke is completed. The initial condition is now restored and the whole sequence of operations is repeated.

THERMODYNAMIC CONSIDERATIONS

Each cycle results in an enthalpy change of $\Delta H = +(p_1-p_2)V$ for the gas at the warm end of the refrigerator and a corresponding change of $\Delta H = -(p_1-p_2)V$ at the cold end, where p_1 and p_2 are the high and low pressures and V is the displaceable volume. This can be demonstrated by reference to Fig. 2 which shows the pressure–volume diagram for a complete cycle for (a) the warm end and (b) the cold end. The two diagrams are identical except for the fact that the p–V area is circumscribed in the counterclockwise direction at the warm end, corresponding to the generation of enthalpy, and in the clockwise direction at the cold end, corresponding to a reduction of enthalpy. Since all other processes involved are constant-pressure processes, and since there is no mechanical work delivered, it can be concluded that, ideally, an amount of heat $(p_1-p_2)V$ is pumped from the low temperature and rejected at room temperature for each cycle.



(a)



(b)

FIG. 2. p–V diagrams (A–B pressurization, B–C intake, C–D expansion, D–A exhaust). (a) Warm volume. (b) Cold volume.

There are many ways in which the machine can deviate from ideal behavior. The principal sources of inefficiency can be divided into two groups, thermal inefficiencies and pressure–volume inefficiencies. Thermal inefficiency can result from inadequate heat transfer or insufficient heat-storage capacity in the thermal regenerator or at the site of the refrigeration load. Heat leakage from the warm to the cold ends of the displacer–cylinder assembly or along the regenerator also contribute to thermal inefficiency. On the other hand, poorly arranged valve timing or restricted flow passages may seriously diminish the actual pressure–volume area, thereby reducing the enthalpy change per cycle. Finally, unnecessarily large gas volumes in the flow passages or in the regenerator, valves or connecting tubes result in gas wastage, because the static volume fills with gas and discharges each cycle without contributing to enthalpy change.

Generally speaking, minimization of thermal inefficiencies is incompatible with minimization of pressure–volume inefficiencies. For example, an ideal thermal regenerator would have extremely small flow passages, a large surface area for good heat transfer and a large volume of metal for good heat storage. This is clearly incompatible with the requirement of minimum internal gas volume and minimum restriction to gas flow. In order to achieve maximum efficiency for the refrigerator, it is necessary to optimize the design in multi-dimensional space, taking into account at least ten independent parameters. These parameters are: length and diameter of regenerator, density, specific heat and degree of subdivision of metal matrix, length and frequency of stroke, length and diameter of piston, and maximum and minimum pressure. Some of these problems have been considered in more detail in a previous publication.(4) An actual single-stage refrigerator shows an efficiency which is dependent on the temperature span, the efficiency decreasing as the span is increased. Several refrigerators have been constructed which produce 65–70 per cent of $(p_1-p_2)V$ in the form of useful refrigeration, as measured by an electrical heater, at approximately 100°K, where p_1 and p_2 are 300 and 75 lb/in^2 above atmospheric pressure.

THE MULTI-STAGE CYCLE

It has been noted that the efficiency of a single-stage refrigerator decreases as the temperature span increases. This suggests the possibility of using several stages arranged end-to-end in order to minimize the span covered by any single stage and thus permit the attainment of very low temperatures with high efficiency. Fortunately, the present cycle lends itself very well to this arrangement, as shown schematically in Fig. 3. The operating cycle in this case is exactly the same as for the single-stage machine. Refrigeration is produced at three different temperatures: 80°K, 35°K and 14°K in the example shown. The relative magnitudes of the displaced volumes determine the amount of refrigeration produced at each

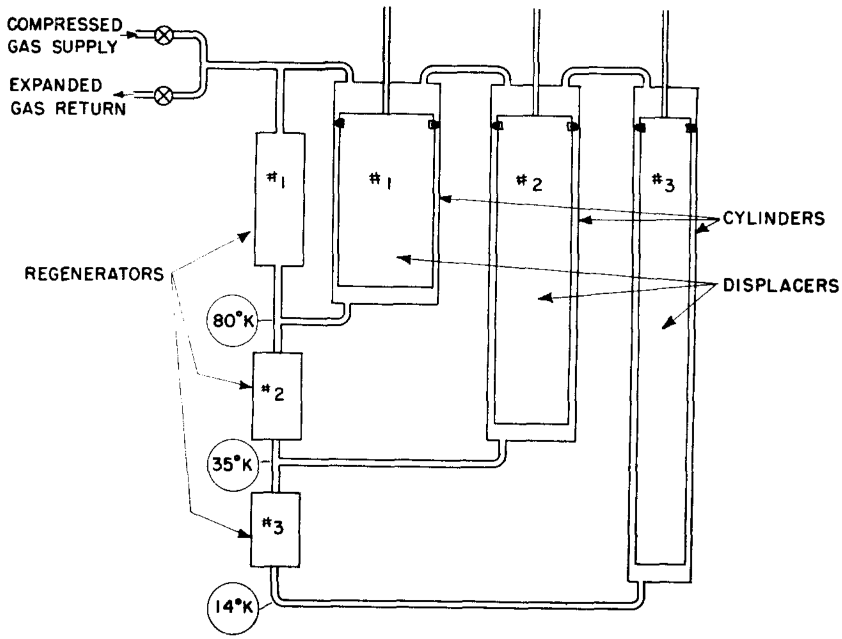


FIG. 3. Three-stage refrigerator.

temperature and will therefore vary from one application to another.

The lowest attainable temperature at the cold end of the third stage is determined by the performance of the third-stage regenerator. The heat capacity per unit volume of all metals diminishes very rapidly at temperatures below 20°K and their heat storage ability becomes poor. The difficulty is further aggravated by the fact that the heat capacity per unit volume of helium gas increases in inverse proportion to the temperature. As a practical matter it has been found that 13–14°K may be regarded as a lower limit if reasonable efficiency is to be preserved, although it is possible to achieve temperatures of 6–8°K if no heat load is applied.

In the multi-stage cycle, refrigeration produced at intermediate temperature levels is used for three general purposes. Firstly, it is used to make up for the inefficiency of the thermal regenerator which feeds gas to that particular temperature level. Secondly, it is used to intercept heat which is flowing into the colder zones, either by conduction along the displacers and cylinders or by radiation from the warm enclosure. And finally, in the event the machine is being used as a liquefier, refrigeration at the intermediate temperatures is used to cool down that portion of gas which is actually

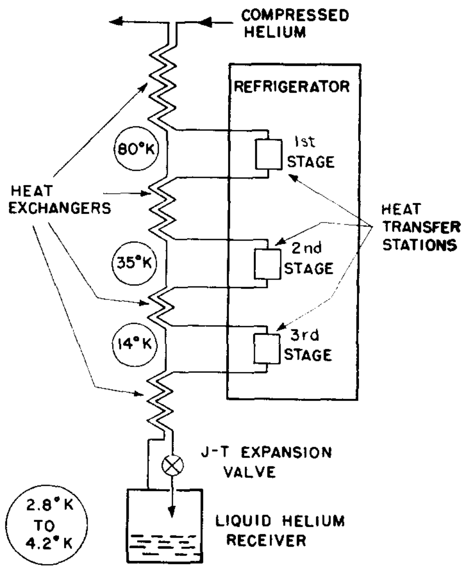


FIG. 4. Helium liquefaction cycle.

liquefied so as to minimize the refrigeration load on the final stage.

Heat transfer between the thermal load (regardless of its origin) and the refrigeration source is established by means of heat-transfer stations which form a part of the flow channels connecting the regenerators to the expansion chambers at the cold end of each cylinder. Such a heat exchanger is shown in Fig. 1, but they are omitted from Fig. 3 for simplicity.

THE HELIUM LIQUEFACTION CYCLE

A three-stage refrigerator such as is shown diagrammatically in Fig. 3 can be modified by the addition of a simple Joule–Thomson cycle to achieve helium liquefaction as shown schematically in Fig. 4. The Joule–Thomson stream, flowing continuously, is cooled by countercurrent flow with the unliquefied portion of gas and also by bringing it into contact with the heat-transfer stations of the three-stage refrigerator at 80°K, 35°K and 14°K. If helium gas at 300 lb/in^2 pressure is cooled to 14°K and expanded to 1 atm in a simple Joule–Thomson cycle, approximately 10 per cent of the gas is liquefied; the remaining 90 per cent is

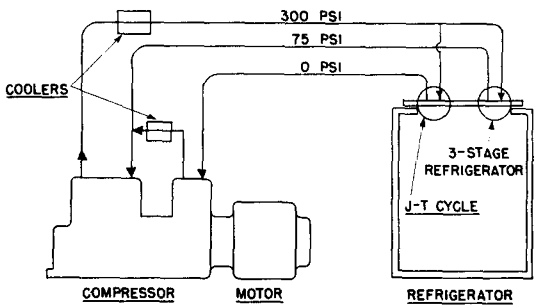


FIG. 5. Helium circulation system.

Table 1. Description of helium refrigerator

	Model no. 1	Model no. 2
*Compressor**:		
J–T cylinder intake	O p.s.i.g.	O p.s.i.g.
J–T cylinder exhaust	75 p.s.i.g.	75 p.s.i.g.
Refrigerator cylinder suction	75 p.s.i.g.	75 p.s.i.g.
Refrigerator cylinder exhaust	300 p.s.i.g.	300 p.s.i.g.
Volumetric displacement J–T cylinder	7·63 ft^3/min	3·85 ft^3/min
Volumetric displacement refrigerator cylinder	15·3 ft^3/min	3·85 ft^3/min
Power consumption	10 h.p.	3 h.p.
Weight: motor and compressor	359 lb	183 lb
Size: motor and compressor	$28 \times 17\frac{1}{8} \times 14$ in.	$20 \times 12\frac{1}{2} \times 12$ in.
Refrigerator:		
Electrical watts at 14°K	15 W	5 W
Electrical watts at 4·2°K	4·0 W	0·750 W
Electrical watts at 3·2°K	3·0 W	0·220 W
Electrical watts at 2·8°K	0·5 W	—
Size of refrigerated space	1 ft^3	0·1 ft^3
Size of refrigerator	$5 \times 2 \times 2$ ft	$27 \times 15 \times 9$ ft
Weight of refrigerator	500 lb	90 lb
Length of no. 1 cylinder	10 in.	7 in.
Length of no. 2 cylinder	20 in.	12 in.
Length of no. 3 cylinder	30 in.	17 in.
Diameter of no. 1 cylinder	$1\frac{3}{4}$ in.	$1\frac{1}{4}$ in.
Diameter of no. 2 cylinder	$1\frac{1}{8}$ in.	$\frac{1}{2}$ in.
Diameter of no. 3 cylinder	$\frac{7}{8}$ in.	$\frac{7}{16}$ in.
Stroke	1 in.	1 in.
Rev/min	75 rev/min	105 rev/min

* Model no. 1 water-cooled; no. 2 air-cooled.

recirculated and recompressed. The over-all recirculation system is shown in Fig. 5.

Several closed-cycle refrigeration systems have been built to date which have operated successfully at temperatures of 4·2°K and lower. While each of these experimental machines has differed from the others in some respects, they have all had certain characteristics in common. In each case the system was closed and indifferent to atmospheric pressure. The three operating pressures are established by pressure regulators which either draw gas from or deliver gas to a small storage tank permanently connected to the system. In all cases the compressors have been standard commercial Freon compressors somewhat modified for compression of helium. Special provisions have had to be made for adequate lubrication and oil separation from the compressed helium. Figs. 6–8 are photographs of the refrigerator, the refrigerator in its enclosure, and the compressor system, respectively, referred to as "Model no. 1" in the preceding table. Figs. 9 and 10 are photographs of "Model no. 2" refrigerator and its air-cooled compressor system.

Operating experience to date has been limited to fairly short running periods of 50 hr or less because of the developmental nature of the present program. Nevertheless, the accumulated experience leads to conviction that the reliability of these machines can be excellent. As in all mechanical equipment, the final judgment regarding reliability cannot be made until many thousands of hours of experience have been accumulated.

SUMMARY

A new closed-cycle helium refrigerator is described which essentially eliminates cold moving parts and therefore appears capable of unusual simplicity and high reliability.

It is expected that refrigerators utilizing these design principles will eventually firmly establish the feasibility of superconductive circuits and other practical applications of liquid-helium temperatures.

Present refrigerators require approximately 3 kW for each watt of energy dissipated at 4·2°K; however, it is expected that this figure can be reduced by at least 50 per cent.

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