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MICROMINIATURE REFRIGERATION – SMALL IS BETTER

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Microminiature Joule–Thomson refrigerators have been fabricated using a novel photolithographic process. We discuss the scaling laws which allow one to design these refrigerators of arbitrarily small size and the fabrication process which makes it possible to mass produce them at low cost. Commercial systems are now available for operation with N_2 gas from ambient to 80 K with a refrigerator capacity of 250 mW. Lower temperature operation using H_2 and He is presently under development. The use of these devices for microcalorimetry, material research and other applications is reported.

In this article we discuss recent progress in the development of microminiature cryogenic gas refrigerators. What led to this development [1] was the realization that many cryogenic devices such as infra-red detectors, low-noise amplifiers, SQUID sensors and Josephson devices, while they require low temperatures, nevertheless dissipate very little heat in operation. Typically this is in the range of a few microwatts to a few milliwatts. On the other hand, commercial miniature refrigerators which might be used to cool these devices, have a refrigeration capacity of the order of 1–10 W. If one is simply interested in cooling these devices, then it is clear that a huge mismatch exists between the refrigeration capacity available and the cooling needs of the devices. In view of this we have considered the problem of how one might *micro*-miniaturize such refrigerators.

A large number of cooling cycles have been developed for operation from 4 to 100 K, such as the Gifford McMahon, Collins, and Stirling cycles. Most of these obtain their cooling by allowing the gas to do external work. This requires moving parts in the low temperature end of the refrigerator. The fabrication of these parts would present a problem if one was to scale them down in size by a factor of 10 to 100. For this reason, we chose to consider instead, the less efficient Joule–Thomson cycle which obtains its cooling by the isenthalpic expansion of the gas with no moving mechanical parts.

The operation of a refrigerator using this process may be illustrated using the temperature entropy diagram of N_2 shown in fig. 1. If the gas at 300 K and a pressure of 100 atm. at A is allowed to expand to 1 atm. through a porous

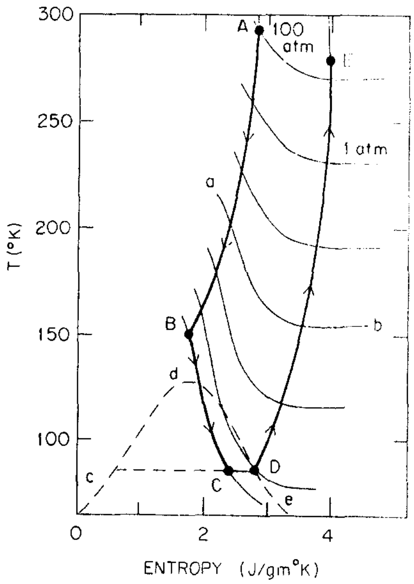

Fig. 1. Temperature–entropy diagram for N_2. Line AB is an isobar at 100 atm. and DE that of 1 atm. A line such as ab is an isenthalp. The trajectory $ABCDE$ represents the operation of a Joule–Thomson refrigerator, with the transfer of heat from the incoming gas along AB to the outgoing cold gas along DE.

plug or fine capillary, it will cool. Cooling occurs at constant enthalpy along the curved lines shown. The effect is small, being about 0.1 K/atm. for N_2 near ambient temperature. However, if this cooled gas is allowed to precool the incoming high pressure gas in a countercurrent heat exchanger the cooling effect becomes regenerative and eventually the process follows the trajector, *ABCDE* in the *T–S* diagrams resulting in liquefication of the working gas.

Two points should be noted. First, cooling will only occur if the enthalpy of the high pressure gas is lower than the enthalpy of the low pressure gas at the same temperature, i.e. the enthalpy curves must dip to the right as in curve *ab*. If this is not so, heating will occur instead. For one to obtain cooling it is necessary to operate the cooling cycle below the so called "inversion point". For nitrogen this is well above room temperature but for hydrogen and helium it is approximately 200 and 40 K, respectively. For this reason multistage Joule–Thomson refrigerators are needed for operation from ambient to 20 K or to 4 K.

Second, it should be noted that the temperature of the gas emerging from the warm end of the heat exchanger at E will be *colder* than the gas which enters at A. This temperature difference divided by the total temperature range over which the exchanger works is a measure of the inefficiency of the heat exchanger. The more efficient the exchanger, the smaller will be the temperature difference. In a practical exchanger a few degrees difference is usual. The total refrigeration capacity is then the difference between the enthalpy of point E and that of the isenthalp passing through A. It is thus of primary importance to have a heat exchanger of very high efficiency to obtain refrigeration at cryogenic temperatures. In the design and scaling of microminiature refrigerators the properties of the exchanger play a key role. We review briefly the factors which come in to the design of the heat exchanger [2].

Consider a countercurrent heat exchanger illustrated in fig. 2. The rate of transfer of heat from the hot stream to the wall, per unit length of the exchanger will be:

$$\mathrm{d}\dot{Q} = \kappa'(T - T')P\,\mathrm{d}\ell = \dot{m}C_\mathrm{p}\left(\frac{\partial T}{\partial \ell}\right)\mathrm{d}\ell, \qquad (1)$$

where κ' is a heat transfer factor describing the transfer of heat from the bulk of the warm stream through the wall and boundary layers to the bulk of the cool stream; $T - T'$ is the temperature difference of two streams; P is the perimeter of wall; $\dot{m}$ is the mass flow; C_p is the heat capacity of hot gas/mass; and ℓ is the length of exchanger.

Likewise for the cold stream we have:

$$-\mathrm{d}\dot{Q} = \dot{m}'C_\mathrm{p}'\left(\frac{\partial T'}{\partial \ell}\right)\mathrm{d}\ell. \qquad (2)$$

Using (1) and (2) and some approximation [3] one finds that

$$\int_0^\ell \frac{\mathrm{d}\dot{Q}}{T - T'} = \kappa P\ell,$$

where κ is some appropriate average of κ'; the efficiency of the exchanger η is

$$\eta = 1 - \frac{T_1 - T_1'}{T_1 - T_2'} = \frac{\kappa P\ell}{\kappa P\ell + \dot{m}C_\mathrm{p}}. \qquad (3)$$

Here T_1 is the temperature of point A, T_1' tem-

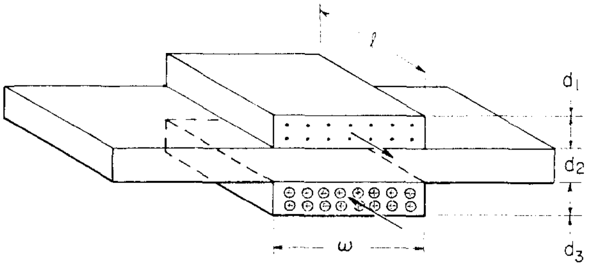

Fig. 2. Schematic view of countercurrent heat exchanger. The two streams of thickness d_1 and d_3, respectively, are separated by a wall of thickness d_2.