RECENT DEVELOPMENTS IN CRYOCOOLERS*

RAY RADEBAUGH
Thermophysics Division
National Institute of Standards and Technology
Boulder, Colorado 80303, U. S. A.

INTRODUCTION

This review of cryocoolers discusses the many advances that have occurred in the various types of cryocoolers in the last few years. Cryocoolers are small refrigerators that reach cryogenic temperatures (T < 120 K). The upper limit of size for a cryocooler has never been well defined. For this paper some discussion of small liquefiers is included, but the large natural gas and industrial gas liquefiers are not included. Most applications of cryocoolers require only a few watts of cooling power, primarily in the temperature range of 10 to 120 K. In this paper there is little discussion of cryocoolers for temperatures around 4 K, since that is the subject of another paper at this conference.

New applications of cryocoolers are appearing, and the requirements for old applications are often undergoing changes. These new and changing cryocooler requirements have been the impetus for the recent developments in cryocoolers. The lack of a suitable cryocooler to meet the requirements of a particular application has hampered the advancement of many applications. For example, superconductivity most likely would be in wide spread use now if it were not for the problems associated with the cryocoolers needed to cool the superconductors. The main problems associated with cryocoolers are: unreliability, inefficiency, size, weight, vibration, and cost. The seriousness of each of these problems depends on the application. Within the last 10 years satellite applications for cooled infrared sensors have become much more important. Obviously these applications require a cryocooler with very high reliability (5 to 10 year lifetimes and no maintenance), high efficiency, small size, low weight, and low vibration. Because only a few cryocoolers are needed for this application at this time, cost has not been a serious problem. However, cost and unreliability have been the major problems for most commercial applications.

APPLICATIONS AND REQUIREMENTS

Table 1 lists most of the current applications for cryocoolers. For many years the largest application for cryocoolers has been for use by the military in cooling infrared sensors to about 80 K for tactical applications. Refrigeration powers range from about 0.25 W to about 2 W. Stirling cryocoolers, used primarily for this application in the last ten years, have been able to meet the requirements, but their mean-time-to-failure (MTTF) of about 4000 hours is far short of the requirement for satellite and most commercial applications. Most of the new developments on Stirling cryocoolers has been in techniques to improve the reliability. The rapid growth of research and development on pulse tube refrigerators in the last 5 years has been because of its potential for improved reliability and lower cost. The largest commercial application of cryocoolers has been for cryopumps, which require a few watts of refrigeration at a temperature of about 15 K. Gifford-McMahon refrigerators have been used almost exclusively for this application. In the last year or two the vibration due to the moving displacer in the Gifford-McMahon refrigerator has become a problem to the semiconductor manufacturers as they continually strive for narrower linewidths and more compact packaging of semiconductor circuits for the computer industry.

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- **Military**
  1. Infrared sensors for missile guidance
  2. Infrared sensors for surveillance (satellite based)
  3. Gamma-ray sensors for monitoring nuclear activity
  4. Superconducting magnets for mine sweeping
- **Police**
  1. Infrared sensors for night vision
- **Environmental**
  1. Infrared sensors for atmospheric studies (satellite) of ozone hole and greenhouse effects
  2. Infrared sensors for pollution monitoring
  3. Cryotrapping air samples at remote locations
- **Commercial**
  1. Hi-T_c superconductors for cellular-phone base stations
  2. Superconductors for high-speed communication
  3. Superconductors for voltage standards
  4. Semiconductors for high speed computers
  5. Cryopumps for semiconductor manufacture
  6. Low-level moisture sensors for ultrapure gases
- **Medical**
  1. Cooling superconducting magnets for MRI systems
  2. SQUID magnetometers for heart and brain studies
  3. Liquefaction of oxygen for storage at hospitals and home use
  4. Cryogenic catheters and cryosurgery
- **Transportation**
  1. LNG for fleet vehicles
  2. Superconducting magnets in maglev trains
  3. Infrared sensors for aircraft night vision
- **Energy**
  1. LNG for peak shaving
  2. Liquefaction of natural gas at remote wells
  3. Infrared sensors for thermal loss measurements
  4. Supercond. mag. energy storage for peak shaving and power conditioning
- **Agriculture and Biology**
  1. Storage of blood and semen
  2. Storage of biological specimens
- **Industrial**
  1. Sub-zero heat treatment of steels
  2. Infrared sensors for process monitoring

Table 1: Applications of cryocoolers

TYPES OF CRYOCOOLERS

Cryocoolers are classified as recuperative types if they use only recuperative heat exchangers or as regenerative types if they use at least one regenerative heat exchanger (regenerator) /Ref.1/. Figure 1 shows the schematics of the most common recuperative and regenerative cryocoolers. The recuperative types utilize a continuous flow of the refrigerant in one direction, analogous to a DC electrical system. As a result, the compressor and any expander must have inlet and outlet valves to control the flow direction, unless rotary or

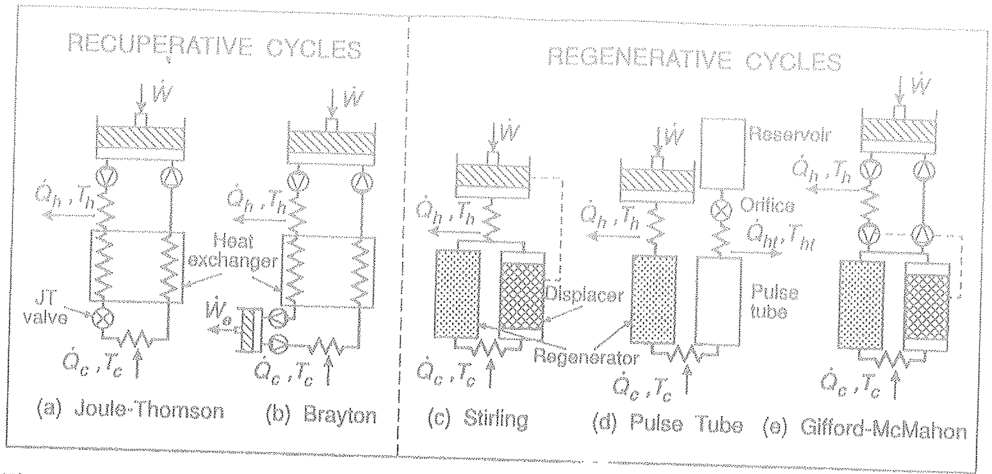

Figure 1. Schematics of five common cryocooler ty s

turbine compressors and expanders are used. The recuperative heat exchangers have two separate flow channels. In regenerative cycles the refrigerant undergoes an oscillating flow and an oscillating pressure analogous to an AC electrical system. The compressor, or pressure oscillator, for the regenerative cycles needs no inlet or outlet valve. However, an oscillating pressure can be generated from a valved compressor by using another set of valves to switch between the high and low pressure sides of the compressor. The regenerator has only one flow channel, and the heat is stored for a half-cycle in the regenerator matrix, which must have a high heat capacity. Recent advances have occurred in all of these cryocooler types, which are discussed below.

JOULE-THOMSON CRYOCOOLERS

Closed-cycle Joule-Thomson cryocoolers until recently have not been used in many applications because of low cycle efficiency and poor reliability of the compressor. Significant developments have occurred in the last few years regarding both of these problems, which have now brought this cycle from one of neglect to one of a serious contender for several applications. Plugging of the JT valve is still a persistent problem, but the use of self-regulating valves has eased the problem somewhat /Ref. 2/. The Joule-Thomson cycle has an intrinsic inefficiency associated with the irreversible expansion in the JT valve or orifice. This intrinsic inefficiency depends on the gas properties. No cooling occurs in an ideal gas when it expands at constant enthalpy through the JT valve, whereas cooling does occur in a real gas whenever the enthalpy is reduced with increasing pressure. The efficiency of the expansion process is quite high when the refrigerant is in the liquid or near-liquid state before passing through the JT valve, as in the vapor-compression cycle. (The vapor-compression cycle is the same as the Joule-Thomson cycle except there is no need for the heat exchanger shown in Figure 1a.) To achieve temperatures of 77 K nitrogen gas has been used as the refrigerant in Joule-Thomson cryocoolers. At 300 K nitrogen must be compressed to a very high pressure to bring about any significant enthalpy change. The small enthalpy change results in a low cycle efficiency and the high pressure leads to a low compression efficiency and high stresses on compressor components, hence, poor reliability. The use of gas mixtures instead of pure nitrogen or argon gives rise to much greater enthalpy changes in the gas and a resulting improvement in the cycle efficiency, even at rather low pressures. The lower pressures ease the compressor stresses and improve its efficiency. An advantage of the Joule-Thomson cryocooler is the very low level of vibration because there are no moving parts or oscillating pressures in the cold head.

Mixed Refrigerants

The use of mixed refrigerants in the Joule-Thomson cycle has a fairly long and somewhat obscure history. Missimer /Ref. 3/ reviews some of this history and discusses some recent trends regarding mixed refrigerants in terms of their ozone depleting potential (ODP) and global warming potential (GWP). In 1936 Podbielniak /Ref. 4/ received a U. S. patent on the first use of mixed refrigerants in a single flow stream using a series of heat exchangers and phase separators. In the liquefaction of natural gas, a separate refrigerant has always been used because of the impurities in the natural gas. Many stages of cooling are used to improve the efficiency. The classical cascade cycle uses propane, ethylene, and methane for the three separate stages with a separate compressor for each stage /Ref. 5/. In 1959 Kleemenko /Ref. 6/ used a single flow stream which consisted of a gas mixture for the liquefaction of natural gas. The best performance was obtained with a mixture of 65 mol% methane, 20 mol% ethane, and 15 mol% normal butane. This single flow cycle had an efficiency better than that with two flow streams. The single stream cycle is now known as the mixed refrigerant cascade (MRC) cycle for the liquefaction of natural gas. It is sometimes referred to as the auto-refrigerated cascade (ARC) or the Kleemenko cycle. It can provide cooling at any number of stages through the use of phase separators and expansion valves as shown in Figure 2. The phase separators allow only the liquid phase to be expanded at each stage, thus maintaining a high efficiency in the expansion process. The phase separators also are useful in reducing the amount of impurities, like water and oil, that are passed on to lower temperatures. In 1972 Missimer /Ref. 7/ used this same cycle with the addition of flow restrictions for added stability, but he replaced most of the hydrocarbons in the gas mixtures with CFCs and used oil lubricated air conditioning compressors for the cycle (known as the Polycold cycle) to provide about 300 W of refrigeration at 140 K. Recently the CFCs in this cycle have been replaced with non-CFCs. The Polycold cycle has been widely used for cold-trapping water vapor in vacuum systems.

In 1969 Fuderer and Andrija /Ref. 8/ were granted a German patent on a mixed refrigerant cycle in which there were no phase separators nor intermediate expansion valves. The entire mixed refrigerant was brought to the cold end for expansion through the one expansion valve, just like the conventional Joule-Thomson cycle, except the refrigerant consists of a gas mixture that is 100% liquid before expanding through the valve. This cycle is useful for applications where there is little need for refrigeration at intermediate temperatures. Temperatures down to 103 K were obtained by Fuderer and Andrija using a gas mixture of equal molar amounts of nitrogen, methane, ethane and propane and a 40:1 pressure ratio. Unfortunately, their work received little attention. They pointed out that because the mixture is in the two phase region in the heat exchanger, much greater heat transfer coefficients are possible compared with pure gas streams. In 1973 Alfeev *et al.* /Ref. 9/ from the Soviet Union used a similar mixture (30 mol% nitrogen, 30 mol% methane, 20 mol% ethane, 20 mol% propane) to achieve a temperature of 78 K using a 50:1 pressure

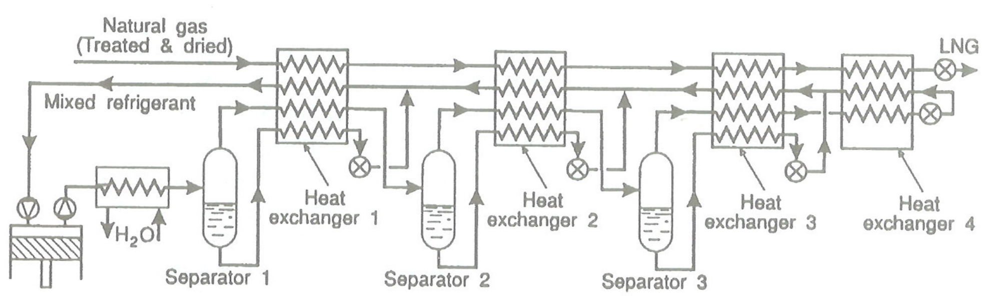

Figure 2. Schematic of a three stage mixed refrigerant cascade (MRC) cycle used for the liquefaction of natural gas.

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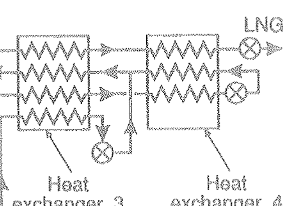

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ratio. The system efficiency was 10 to 12 times better than for pure nitrogen. Temperatures below 70 K were reached by adding neon, hydrogen or helium to the mixture. The application was for cooling infrared sensors to about 80 K, and there was no need for any significant cooling at intermediate temperatures. Even though methane, ethane, and propane have individual freezing points in the range of 85 to 90 K, some mixtures of these fluids have freezing points at least as low as 70 K /Ref. 10/.

The use of these gas mixtures was first brought to the attention of the cryocooler community outside the Soviet Union by Little /Ref. 11, 12/ who also verified the greatly enhanced cooling power of the gas mixtures compared with pure nitrogen. Temperatures of 80 K could be reached with high pressures in the range of 3 to 5 MPa. He showed that the nitrogen-hydrocarbon gas mixtures could be made non-flammable by the addition of a few percent of the fire retardant compound 13B1, CF_3Br, but at the expense of a high ozone depleting potential. In 1994 Longsworth /Ref. 13, 14/ described a closed-cycle system using a commercially available oil-lubricated compressor (rolling-piston type) with a gas mixture of 36 mol% nitrogen, 20 mol% methane, 12 mol% ethylene, 20 mol% propane, and 12 mol% isobutane to obtain 1 W of cooling at 80 K and 10 W at 93 K with a high pressure of about 2 MPa. Compressor input power was about 350 to 400 W. With these mixtures the refrigerant is 100% liquid in the high pressure line when it reaches the expansion valve. Thus, Longsworth refers to the refrigerator as a throttle refrigerator as opposed to a Joule-Thomson refrigerator, in which the high pressure gas is normally 100% in the gas phase before expansion. Currently, there is much interest in optimizing these gas mixtures in order to increase efficiency and to reduce the pressure needed to reach 80 K.

The refrigeration power of a Joule-Thomson or throttle refrigerator is given by

$$\dot{Q}_{ref} = \dot{n}[h(T, P_{low}) - h(T, P_{high})]_{\min} = \dot{n}\Delta h_{\min}, \tag{1}$$

where $\dot{n}$ is the molar flow rate and $\Delta h_{\min}$ is the minimum difference in the molar enthalpies between the low and high pressures streams for all temperatures between the two ends of the heat exchanger. The ideal work of compression is given by

$$\dot{W}_{ideal} = \dot{n}\{h(T_0, P_{high}) - h(T_0, P_{low}) - T_0[s(T_0, P_{high}) - s(T_0, P_{low})]\} = \dot{n}\Delta g_0\,, \tag{2}$$

where s is the molar entropy and Δg_0 is the change in the molar Gibbs free energy at the compression temperature T_0. The ideal coefficient of performance COP of the refrigerator is given by

$$\mathrm{COP} = \dot{Q}_{ref}/\dot{W}_{ideal} = \Delta h_{\min}/\Delta g_0\,. \tag{3}$$

The optimum gas mixture for the maximum efficiency is that which has the maximum value of $\Delta h_{\min}/\Delta g_0$ in the temperature range of interest. Figure 3a shows the enthalpy of a nitrogen-propane gas mixture for two different pressures. The increased slope in the high pressure curve at about 140 K and 280 K occurs because of the evaporation of the nitrogen and the propane, respectively. Figure 3b shows the temperature dependence of $\Delta h/\Delta g_0$ between 0.1 and 5 MPa for (1) pure nitrogen, (2) pure propane, (3) 65 mol% nitrogen-35 mol% propane, (4) 30 mol% nitrogen-30 mol% methane-20 mol% ethane-20 mol% propane, and (5) 40 mol% nitrogen-17 mol% methane-15 mol% ethane-28 mol% propane. The compression temperature was taken as 300 K in the calculation of Δg_0. The thermodynamic properties of these mixtures were calculated by using DDMIX developed at NIST /Ref. 15/. Because pure nitrogen has a small enthalpy difference at 300 K it gives a low cycle COP of only 0.0296. At 80 K the relative Carnot efficiency with pure nitrogen is only 8.1% for the ideal cycle. On the other hand, propane has a large enthalpy difference at 300 K but a small and slightly negative enthalpy difference below its boiling point (230.7 K) at 0.1 MPa. Its COP is 2.80

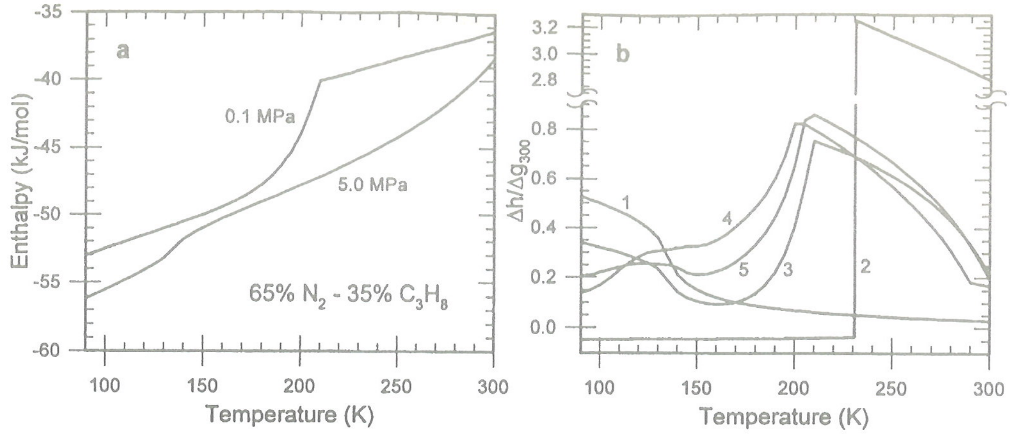
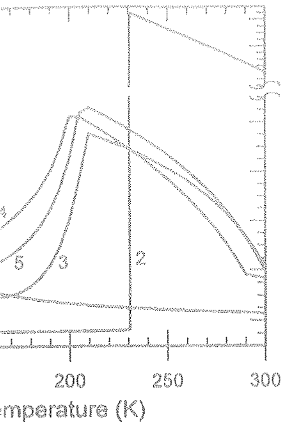


Figure 3. (a) Molar enthalpy of 65 mol% nitrogen-35 mol% propane.
(b) Normalized enthalpy difference between 0.1 MPa and 5 MPa for 5 mixtures discussed in the text.

for temperatures down to 230.7 K, but it cannot be used for cooling below that temperature with a low pressure of 0.1 MPa. For a mixture of 65 mol% nitrogen and 35 mol% propane the $\Delta h/\Delta g_0$ curve in Figure 3b shows large values at both the high and low temperatures, but lower values at intermediate temperatures. Since the minimum value of $\Delta h/\Delta g_0$ determines the cycle COP, this binary mixture has a COP of 0.0941 for temperatures below 160 K, which is over three times higher than that of pure nitrogen. In order to increase the $\Delta h/\Delta g_0$ at intermediate temperatures, gases which have intermediate boiling points must be added to the mixture. For example, see curve 4 in Figure 3b, which is the four component mixture used by Alfeev *et al.* /Ref. 9/. Here the COP has been increased to 0.103 for a temperature of 80 K, resulting in a relative Carnot efficiency of 28.3% for the ideal cycle. This ideal efficiency is only 3.5 times that of pure nitrogen, so the factor of 10 improvement seen in the experiments could be due to the enhanced heat transfer in the heat exchanger when a two-phase fluid is present. The mixture chosen by Alfeev is seriously limited in the $\Delta h/\Delta g_0$ values at high and low temperatures in Figure 3b (curve 4) due to a lack of nitrogen and propane and too much methane and ethane. The addition of more nitrogen and propane (or isobutane) as shown by curve 5 in Figure 3b increases the ideal COP to about 0.20 at 90 K and 0.19 at 80 K. At 80 K the relative Carnot efficiency is 52% for the ideal cycle. Further studies of optimum mixtures and of heat transfer in multicomponent, two-phase mixtures are needed to further improve the Joule-Thomson or throttle cycle. Further studies are also needed regarding the freezing point of gas mixtures, including gases with trace amounts of water and oil. There are very little experimental data on the freezing point of gas mixtures.

Sorption Compressors

In an effort to eliminate the only moving part in the Joule-Thomson cycle and improve the reliability of this cycle, the mechanical compressor has been replaced by sorption compressors in many recent studies. The sorption compressors also eliminate any vibration. Alternate heating and cooling of the sorption compressors cause a circulation of the refrigerant. This subject has been reviewed recently by Wade /Ref. 16/. The only moving parts in the sorption compressors are check valves which operate once every few minutes to cause steady flow in one direction. Unfortunately, the sorption compressors have only been used with pure gases, which limits their efficiency unless many stages are used. Sorption of mixed gases may be possible in principle if the selective nature of sorption can be overcome.



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Chemisorption of hydrogen with hydrides (e.g., V and $LaNi_5$) and oxygen with praseodymium cerium oxide (PCO) can be done with the compressors operating at 300 K or even higher. Most other gases require the use of physisorption on microporous carbon. Low boiling point gases are difficult to adsorb at 300 K, so physisorption compressors are generally limited to systems with cold temperatures above about 120 K. Alvarez *et al.* /Ref. 17/ discuss progress on an advanced sorption compressor system to reach 125 K using krypton with a xenon precooling stage at 170 K. They expect to achieve specific power inputs of 40 W/W at 125 K with a rejection temperature of 260 K. With improved adsorbent materials like saran carbon (a high density, high surface area carbon) the specific power input may be reduced to about 20 W/W. High efficiency in sorption coolers is only possible with the use of regenerative heating of the sorbent bed in which heat from one compressor module in a cool-down state is used to heat another compressor module in a warm-up state. Several modules are needed for high efficiency /Ref. 18/.

For temperatures down to about 65 K, Bard and Jones /Ref. 19/ proposed the use of a PCO compressor for an oxygen stage precooled with krypton and xenon stages using physisorption compressors. Specific powers of 50 W/W have been predicted for cooling to 65 K when using regenerative configurations /Ref. 16/. Johnson and Jones /Ref. 20/ proposed the use of hydride beds pumping on solid hydrogen to reach temperatures down to 10 K in a periodic mode. The concept was successfully demonstrated in 1991 /Ref. 21/ and a prototype space flight experiment (known as the Brilliant Eyes Ten-Kelvin Sorption Cryocooler Experiment or BETSCE) of this concept has been developed and ground tested /Ref. 22/. A heat load of 100 mW was maintained at 9.5 K for over 20 minutes and the system could be recycled in under 5.5 hours. In this experiment Stirling cryocoolers were used to precool the hydrogen to about 65 K.

Electrochemical Compressors

The electrochemical compressor converts neutral molecules such as hydrogen or oxygen to ionic form at one porous electrode and uses an electric field to cause these ions to migrate through a solid electrolyte membrane to a higher pressure where they are converted back to neutral molecules at the other porous electrode. The membrane material must have a large ionic conductivity and be impermeable to neutral molecules. Such a compressor operates with no moving parts. Ruggeri /Ref. 23/ has calculated an efficiency for hydrogen compression of about 60% for a pressure ratio of 135. For the same compression ratio, a mechanical compressor would require about four stages of compression and have an efficiency of about 20% or less. In order to develop high ionic conductivities, temperatures above about 600 °C would be necessary. No experimental results are presented by Ruggeri. Practical problems such as the development of high temperature seals and suitable membrane materials with high ionic conductivities and high strengths must be overcome before electrochemical compressors become a reality.

BRAYTON CRYOCOOLERS

The use of an expansion engine to carry out a reversible expansion of the gas as shown in Figure 1b leads to higher efficiencies than are possible with Joule-Thomson cryocoolers. The higher efficiency comes about because of (a) nearly isentropic expansion instead of isenthalpic expansion and (b) the use of recovered work to assist in the compression. For a temperature of 75 K the maximum recovered power is only 25% of the ideal input power, so the expansion work is usually dissipated at ambient temperature to keep the system simple. Most Brayton cryocoolers use turboexpanders with gas bearings to provide high reliability and very low vibration. The turboexpanders are very efficient in large sizes and are used in large liquefaction plants. For small cryocoolers the challenge is in fabricating the small turboexpanders and maintaining a high expansion efficiency. Swift /Ref. 24/ discusses the latest developments in a single stage turbo-Brayton cryocooler designed to provide 5 W of cooling at 65 K. Turbomachines with gas bearings are used for both the compressor and the expander. The working fluid is neon with an inlet pressure of 0.11 MPa and a pressure ratio of 1.6. A specific power of 43 W/W was obtained with the engineering model Brayton cryocooler. With a cold temperature of 65 K and a reject temperature of 280 K the Carnot efficiency is 7.7%. Though the turbomachines are very small (15 mm diameter compressor impeller and 3.2 mm diameter expander rotor) the heat exchanger is quite large (90 mm diameter by 533 mm in length). The total system mass is 11.9 kg, with the heat exchanger comprising 52% of the total mass. The heat exchanger used 300 slotted copper disks with slots 0.1 mm wide by 3 mm long. The high cost of the turbo-Brayton cryocoolers limits their use to space applications in which high reliability, high efficiency, and very low vibration are needed. An even smaller turbo-Brayton cryocooler is now being developed by McCormick *et al.* /Ref. 25/. Their goal is to provide 2 W of refrigeration at about 65 K with less than 100 W of input power.

STIRLING CRYOCOOLERS

Most of the recent developments in Stirling cryocoolers have involved methods to improve reliability. Linear motor drives have replaced rotary motor drives in most applications because they eliminate many moving parts and reduce side forces between the piston and cylinder. Lifetimes of about 4000 hours are now routine for linear compressors with dry rubbing contact. Recent work by Pruitt /Ref. 26/ has shown lifetimes in excess of 15,000 hours in linear compressors with rubbing contact. Wearout times of 3 years may be possible in some cases using improved materials for the rubbing contact. Such lifetimes may be sufficient for many commercial applications. The 5 to 10 year lifetimes needed for satellite applications as well as for some commercial applications can only be achieved when all rubbing contact is eliminated. Non-rubbing operation is achieved with piston devices by using flexure, gas, or magnetic bearings, or with diaphragm devices in which a flexing diaphragm causes compression and expansion of the working gas. Efficiencies for the conversion of electrical power to PV power have been as high as 85% in these linear compressors when operating at resonant frequencies.

Vibration caused by the reciprocating motion is reduced by using dual opposed pistons, a passive balancer, or an active balancer. Vibration forces of about 1 N are typical in such cryocoolers that have input powers of the order of 100 W. Significant advances have been made in further vibration suppression in flexure-bearing cryocoolers by using active harmonic nulling with dual opposed pistons /Ref. 27/. Axial vibration forces are often reduced to less than 0.1 N with this technique, though radial vibration forces remain unchanged at about 1 N /Ref. 28/.

Flexure Bearings

The most common technique for eliminating rubbing contact uses flexure bearings to support the piston and displacer inside their corresponding cylinders without any contact. A clearance gap of 10 to 20 μm provides the necessary flow impedance to serve as a dynamic seal. Figure 4a shows a simplified cross-section of a typical Stirling compressor with flexure bearings. A similar arrangement is used to support the displacer. In practice most compressors use two opposed pistons to eliminate most of the vibration. The flexure bearings provide a stiff support in the radial direction and act as a weak spring in the axial direction. Though flexure bearings have been used previously in many other devices, they were first used in Stirling cryocoolers in the early 1980s by Davy of the University of Oxford. Thus, the name Oxford-style Stirling cryocoolers is often used to describe these flexure supported cryocoolers. Davy /29/ reviews the development of these cryocoolers, which are now manufactured for space applications by several companies.

Figure 4b shows the geometry of the spiral flexure bearing commonly used for the Oxford-style Stirling cryocoolers. Both beryllium copper and spring-grade stainless steel

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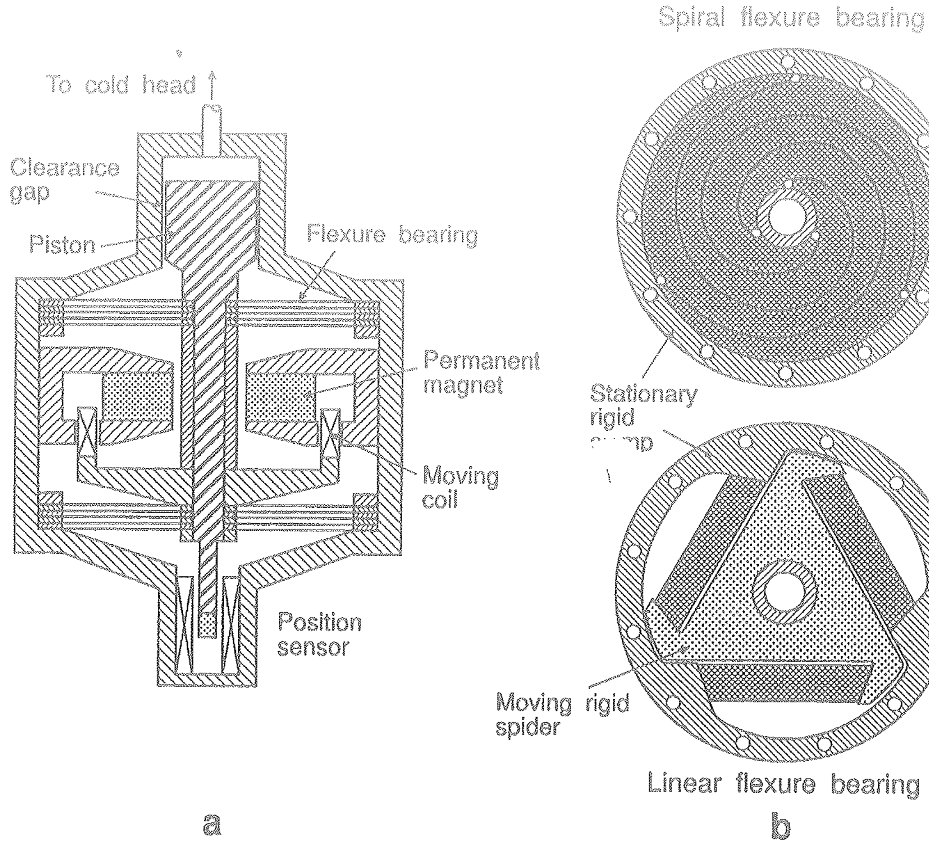


Figure 4. (a) Cross-section of the Oxford-style linear compressor showing the use of flexure bearings. (b) Two types of flexure bearings.

have been used for these flexures, which are fabricated with photoetching techniques. A new flexure geometry proposed by Wong *et al.* /Ref. 30/, also shown in Figure 4b, uses linear arms instead of spiral arms. Marquardt and Radebaugh /Ref. 31/ have shown that this new design should have much greater radial stiffness than the spiral design for the same stress level in the flexure. There is currently some speculation that the linear flexure bearings may reduce the radial vibration modes that cannot be eliminated by the active harmonic nulling.

Gas Bearings

Gas bearings have also been investigated for the elimination of rubbing contact between piston and cylinder. Work in France has focused on the use of hydrodynamic gas bearings, first in a Vuillemier cryocooler for space applications /Ref. 32/ and more recently in a two-stage Stirling cryocooler /Ref. 33/. The linear motor used moving magnets and the rotation to drive the gas bearing was provided with a brushless motor. Magnetic springs provided the axial centering and were used in the displacer drive to provide for resonance operation. A rotational frequency of 5 to 10 Hz was used to provide the gas bearing effect in a 10 μm gap. A reciprocating frequency of about 35 Hz was used.

Hydrostatic gas bearings have been used in the U. S. for a Stirling cryocooler /Ref. 34/. The gas flow for the bearing is driven by the oscillating pressure in the working space through a one-way valve (port alignment between piston and cylinder could also be used to provide one-way flow) and flow restriction. An oscillating frequency of 60 Hz was used for this cooler.

Magnetic Bearings

Magnetic bearings were first used in a one stage Stirling cryocooler developed for NASA in the early 1980s /Ref. 35/. The active magnetic bearings used electromagnetic coils to provide an attractive force on the ferromagnetic shaft in the x and y directions. Position sensors in both the compressor and displacer provided the necessary feedback to control the radial position of the piston and the displacer. Thermal performance and life testing were satisfactory, but the cost and complexity of the magnetic bearing system prevented it from being used further in the U. S. Recently a Stirling cryocooler with magnetic bearings has been under development in Japan /Ref. 36/.

Diaphragm Compressors and Expanders

A Ti-6Al-4V diaphragm with a flexible outer ring (similar to loudspeaker construction) has been used in both the compressor and expander assemblies of a single-stage 65 K Stirling cryocooler /Ref. 37/ and in a two-stage 30 K Stirling cryocooler /Ref. 38/. This construction eliminates the need for clearance gaps and precise alignment. Such diaphragm devices tend to have short strokes and large diameters. The diaphragms were made double acting, which meant that the regenerator also functioned as a recuperative heat exchanger between the gas flowing in opposite directions in the two separate systems.

PULSE TUBE CRYOCOOLERS

The three main types of pulse tube cryocoolers as discussed by Radebaugh, *et al.* /Ref. 39/ are the basic, resonant (or thermoacoustic), and orifice. The orifice pulse tube refrigerator (OPTR) is the only one which achieves cryogenic temperatures and is the only one considered in this review. Because the OPTR has no moving parts at the cold end, it has the following advantages over the Stirling cryocooler: (a) more reliable, (b) lower cost, (c) lower vibration, (d) lower EMI, (e) less sensitive to side loads, and (f) better launch survivability. However, until the last few years, its efficiency was much less than that of the Stirling refrigerator. The latest review of pulse tube refrigerators was by Radebaugh in 1990 /Ref. 40/.

The OPTR operates with an oscillating pressure, which can be provided with a compressor like that of the Stirling refrigerator. Alternatively, the oscillating pressure is sometimes provided with a Gifford-McMahon compressor and valves, with a considerable sacrifice in efficiency. A brief description of the operating principle is given here and refers to Figure 1d. The OPTR operates on a cycle similar to the Stirling cycle except the proper phasing between mass flow and pressure is established by the passive orifice instead of the moving displacer. As the gas is compressed adiabatically in the pulse tube, it is heated. About one-third of this hot, compressed gas flows through the orifice to the reservoir volume (where the pressure is always at the average pressure) and transfers heat to the hot heat exchanger. As the gas in the pulse tube is expanded adiabatically, it is cooled. The cold expanded gas is forced past the cold heat exchanger as the gas in the reservoir flows through the orifice into the pulse tube, where the pressure is low. The pulse tube provides a buffer volume of gas to maintain a temperature gradient between the hot and cold ends of the pulse tube. Any mixing or turbulence within the buffer volume introduces a heat load on the cold end. Rawlins *et al.* /Ref. 41/ have shown experimentally that the time-averaged enthalpy flow (equal to the gross refrigeration power) is only 55 to 85% of the ideal value. This pulse tube loss becomes one of the largest in the entire pulse tube refrigerator. The detailed fluid dynamics within the pulse tube are just beginning to be understood. Recently Lee *et al.* /Ref. 42/ used a visual smoke-wire technique to observe radial mixing and flow streaming over the length of the pulse tube.

The analytical model developed at NIST in the late 1980s /Ref. 40/ for the OPTR solved the conservation of mass and energy equations by assuming simple harmonic pressure, mass flow, and temperature oscillations within the entire pulse tube refrigerator. That model also assumed adiabatic processes occurred within the pulse tube. Recent thermoacoustic theories /Ref. 43-45/ also use a harmonic approximation, but they include a linear approximation with higher harmonics and realistic heat transfer and viscous effects between the gas and tube walls. The non-adiabatic effects generally account for only a few percent loss in refrigeration power compared with the adiabatic model, unless the tube size is quite small. Other losses inherent in these models appear to result in a time-averaged enthalpy flow within the pulse tube that is about 70 to 80% of the ideal value /Ref. 44,45/, in good agreement with the measurements of Rawlins *et al.*/Ref. 41/. Further work with these thermoacoustic models is needed to clearly identify the loss mechanisms. These models also include the typical acoustic equations that relate the amplitude of oscillation to the axial position. This dependence on axial position becomes important at high frequencies (greater than about 100 Hz) where the acoustic wavelength is not long compared with the length of the refrigerator.

The orifice concept for pulse tube refrigerators was first introduced by Mikulin /Ref. 46/ in 1984, who reached a low temperature of 105 K in one stage. In 1986 Radebaugh, *et al.* /Ref. 39/ changed the position of the orifice from being internal to the pulse tube to the external position shown in Figure 1d and achieved a low temperature of 60 K. After a few other improvements over the next few years, temperatures of below 40 K were achieved at NIST and several other laboratories with large single stage pulse tubes.

Improved Efficiencies

At the time of the latest review of pulse tube refrigerators in 1990 /Ref. 40/, efficiencies of the OPTR were still several times less than that of Stirling refrigerators. The low efficiency kept it from being used for most applications in spite of its many advantages compared with the Stirling refrigerator. Since 1990 several advances have been made in pulse tube refrigerators which have allowed them to achieve efficiencies nearly as high as that of Stirling refrigerators. A large OPTR constructed at NIST in 1991 produced 31.1 W of refrigeration at 80 K with a PV power input of 602 W at a rejection temperature of 316 K. The relative Carnot efficiency was 15.3% for PV work and 13% for electrical input power if the compressor were 85% efficient (typical of recent linear compressors). The average operating pressure was 2.5 MPa and the frequency was 4.5 Hz.

Improved efficiencies at higher operating frequencies were made possible by the introduction of the double inlet concept in 1990 by Zhu *et al.* /Ref. 47/. They added a second orifice (often referred to as the secondary orifice or bypass orifice), as shown in Figure 5. (The multi-inlet in this figure is discussed in the section on multiple stages.) With this orifice the gas flow needed to compress and expand the gas at the warm end of the pulse tube is taken directly from the compressor instead of passing through the regenerator and pulse tube. The reduced mass flow through the regenerator reduces the regenerator loss, particularly at high frequencies where the regenerator loss becomes quite large. This secondary orifice can reduce the cold end temperature by at least 15 to 20 K in a well designed pulse tube operating at frequencies of 40 to 60 Hz. A very efficient miniature pulse tube using the double inlet concept was reported by Chan *et al.* /Ref. 48/ in 1993. Their integral, inline pulse tube cryocooler operated at a frequency of 55 Hz and produced 0.53 W of cooling at 80 K with a compressor input power of 17.8 W and a reject temperature of 287 K (7.7% Carnot). The lowest temperature achieved with a single-stage, double inlet configuration was 28 K by Ravex *et al.* in 1992 /Ref. 49/. A comparison of efficiencies for pulse tube refrigerators with Stirling refrigerators is shown in Figure 6. The shaded band represents the efficiency range for most of the recent Stirling refrigerators. The circles in Figure 6 show the efficiency of recent pulse tube cryocoolers which have achieved high efficiency. The highest power and highest efficiency one is the NIST OPTR previously

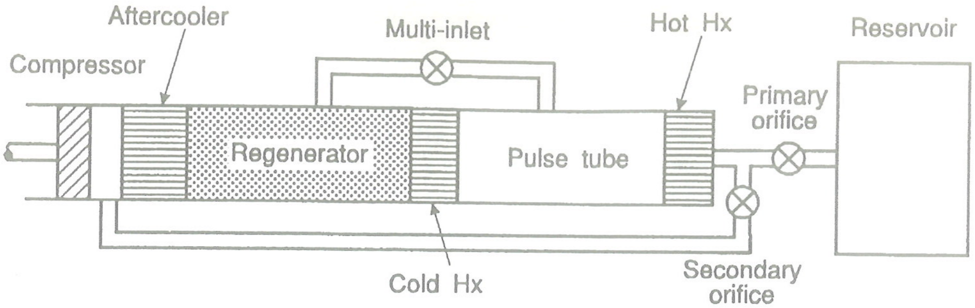


Figure 5. Schematic of the double inlet pulse tube refrigerator in which a secondary orifice is used. The mult-inlet concept for staging is also shown.

discussed. The lowest power OPTR is the miniature one by Chan *et al.* /Ref. 48/. The two mid-size OPTRs are the same device with different input powers and different cold end temperatures /Ref. 50/. Figure 6 shows that the efficiencies of the latest pulse tube cryocoolers is almost as high as the best Stirling cryocoolers of a comparable size. Because of their high efficiency as well as many advantages over Stirling refrigerators, pulse tube refrigerators have been selected to cool the Atmospheric Infrared Sounder (AIRS), an important instrument used for the Earth Observing System (EOS) satellites in the study of the ozone hole and greenhouse effects.

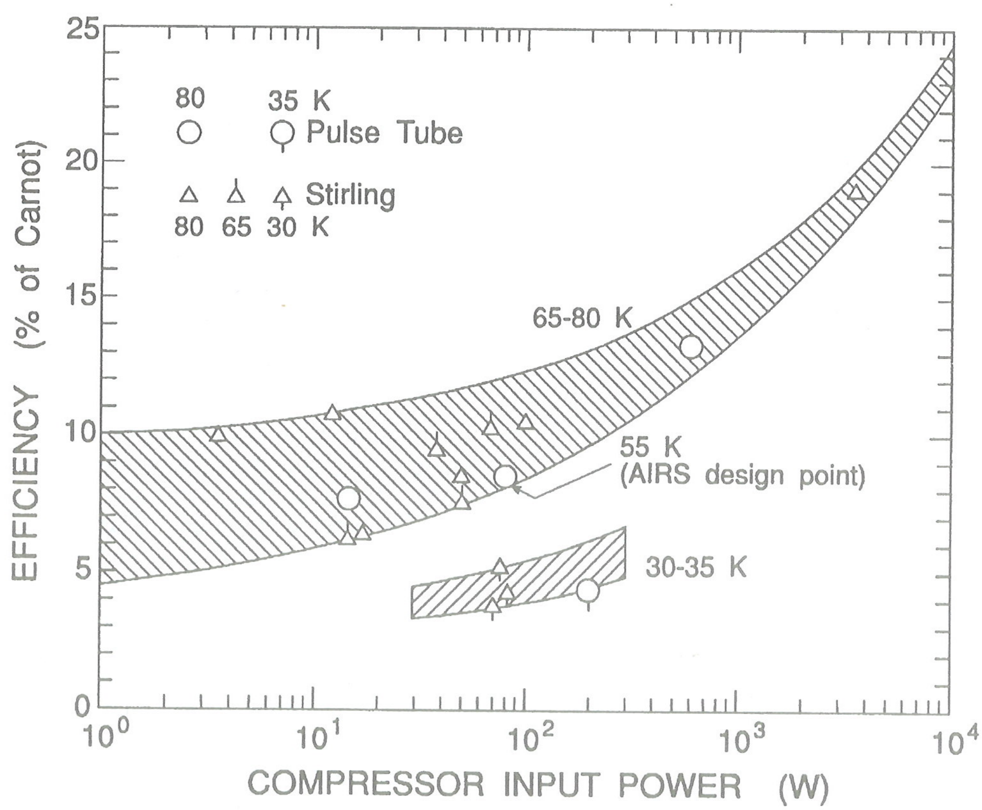


Figure 6. A comparison of the efficiency of pulse tube refrigerators with that of Stirling refrigerators.

With the orifices replaced by a warm expander, even further improvements in efficiency are possible, but at the expense of a second moving part at ambient temperature. Ishizaki and Ishizaki /Ref. 51/ used this warm expander configuration with a one-stage pulse tube and succeeded in reaching 23 K in 1992. This configuration is particularly useful for high efficiency at temperatures above about 200 K, where the orifice pulse tubes lose efficiency. Because the expansion work is dissipated in the orifice instead of being recovered as in the Stirling cryocooler, the efficiency of an ideal OPTR compared with Carnot is given by

$$\eta_{ideal} = (T_h - T_c)/T_h. \tag{4}$$

Kittel /Ref. 52/ discusses this derivation. For T_c = 75 K and T_h = 300 K, η_{ideal} = 75%, whereas for T_c = 250 K, h_{ideal} = 17%. Thus, for cryogenic temperatures practical inefficiencies dominate the intrinsic orifice inefficiency, but for near-ambient temperatures the intrinsic inefficiency severely limits the OPTR efficiency.

Multiple Stages

For temperatures below about 80 K two or more stages are normally used to maintain high efficiency. The purposes of the staging are (a) to provide for net cooling at an intermediate temperature, and/or (b) to intercept regenerator and pulse tube losses at a higher temperature. Two methods exist for the staging arrangement with pulse tube refrigerators. The first uses a separate pulse tube and regenerator for each stage with the warm end of each pulse tube at ambient temperature (parallel arrangement). If the pulse tube losses are too great for the second and lower stages, then their warm ends can be thermally anchored to the cold end of the next higher stage (series arrangement). These two arrangements, or a combination of the two, lead to the use of many pulse tubes, which may be cumbersome.

A new orifice arrangement, used by Zhou and Han in 1992 /Ref. 53/ and called a multi-inlet pulse tube, is in fact a new technique for staging, although it was not recognized as such by the authors. This arrangement, as shown in Figure 5 and known as the multi-inlet arrangement, uses a middle orifice to allow a portion of the gas to enter the pulse tube at an intermediate temperature, thereby producing refrigeration at that location. This staging arrangement maintains the simple geometrical arrangement of a single pulse tube, although it would normally have a change in diameter at the tube junction to maintain a constant gas velocity in the pulse tube. Zhou and Han showed that the use of this middle orifice lowered the cold end temperature from 59 K to 33 K. The lowest temperature achieved with a pulse tube refrigerator was 3.6 K by Matsubara and Gao in 1994 /Ref. 54/ using a three stage parallel arrangement with an unusual regenerative tube at the warm end of the third stage pulse tube. This regenerative tube allows the orifice to be placed at ambient temperature even though the junction between the warm end of the third stage pulse tube and the regenerative tube is at about 120 K.

Thermoacoustically-Driven Orifice Pulse Tube Refrigerator (TADOPTR)

In 1990 Swift, Martin and Radebaugh /Ref. 55/ proposed the use of a thermoacoustic driver (TAD) instead of a mechanical compressor to drive the OPTR, as shown in Figure 7. When the temperature gradient in the closely spaced plates of the TAD exceeds the critical value, spontaneous acoustic oscillations occur in the helium working fluid. Because there are no moving parts in the TADOPTR, it has the potential for low cost and extreme reliability. In 1990 Radebaugh *et al.* /Ref. 56/ succeeded in reaching a low temperature of 90 K with such a TADOPTR, which was the first cryogenic refrigerator with no moving parts. The thermoacoustic driver was about 10 m long and resonated at a frequency of about 40 Hz. The pressure ratio was only about 1.10, although more recent TADs have produced pressure ratios of about 1.20. Optimum pressure ratios with mechanically driven OPTRs are about 1.3 to 1.5. More recently, Godshalk *et al.* /Ref. 57/ have experimented with a TADOPTR

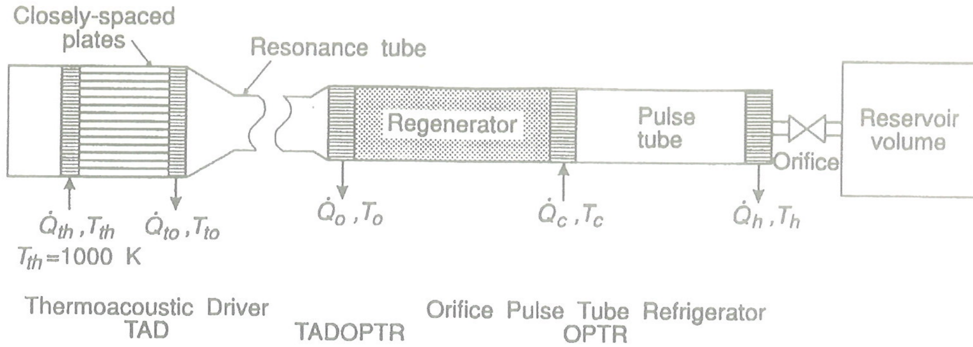


Figure 7. Schematic of the thermoacoustically driven orifice pulse tube refrigerator (TADOPTR), which has no moving parts.

operating at 400 Hz using a 1 m long TAD. By taking advantage of the large phase shift in the pressure between the two ends of the pulse tube that occurs when the acoustic wavelength is not long compared with the pulse tube length, they could reduce the regenerator loss.

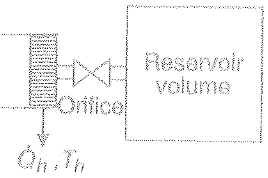
Work is currently in progress under a joint program between NIST, Los Alamos National Laboratory, and Cryenco to develop a 1900 L/day gas-fired TADOPTR to provide liquefied natural gas as an alternative fuel for fleet vehicles. It is estimated that about 30% of the gas would be burned to liquefy the remaining 70%. By comparison, very large liquefaction plants burn about 15% of the gas, but they require considerable maintenance. Such maintenance would not be economical for small systems. Overall system efficiencies for the TADOPTR are highest when primary heat sources are used rather than electric heaters.

GIFFORD-McMAHON CRYOCOOLERS

Gifford McMahon cryocoolers use oil-lubricated compressors made by the millions for the air conditioning industry. Even though they are modified for use with helium gas, their cost is quite low. The oil removal equipment is placed in the high pressure line ahead of the switching valve that generates the oscillating pressure. These cryocoolers are most commonly used in two-stage versions for cryopumps operating at temperatures of about 15 K. This is the largest commercial application of cryocoolers. The Gifford-McMahon cryocooler is also used for cooling shields to 10 to 15 K in MRI systems to reduce the boiloff rate of liquid helium or for direct cooling of Nb_3Sn superconducting magnets to 10 K. Most of the recent developments in Gifford-McMahon cryocoolers have involved the use of high heat capacity regenerator materials to reach temperatures of 4 K without the aid of a Joule-Thomson stage. Since this review focuses on temperatures above 4 K, these developments are not discussed here.

SUMMARY

Many recent advances in both recuperative and regenerative cryocoolers are discussed. Temperatures above 4 K are the primary focus of this review. Applications for such cryocoolers are presented and the requirements which these applications impose on the cryocoolers are reviewed. Many applications have been hampered by the lack of a suitable cryocooler. Recent developments in some cryocoolers have made some applications much more likely. Most applications require improved reliability. For space applications the additional requirements of high efficiency and low weight are needed, whereas for commercial applications low cost is a primary factor. Recent developments to meet some of these requirements include the use of flexure and gas bearings to eliminate rubbing contact in compressors and displacers, pulse tube refrigerators to eliminate the moving displacer in Stirling refrigerators, the use of gas mixtures to improve efficiency of Joule-Thomson



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refrigerators, and the use of sorption compressors and thermoacoustic drivers to replace mechanical compressors and pressure oscillators. The status and some future trends of these developments are presented.

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