Cryocooler Development
(1995 AIChE INSTITUTE LECTURE)

Klaus D. Timmerhaus
University of Colorado, Boulder, CO 80309

Increased demand for cryocoolers for aerospace and terrestrial applications has served as an incentive over the past decade to develop units with specific performance parameters. Since many of these units also require a high degree of reliability, considerable effort has been directed toward meeting this goal. The excellent progress that has been made is summarized, and there is every indication that the use of cryocoolers will expand greatly during the next decade, with a rapid transition from present aerospace/military applications to highly civilian uses in such areas as medicine, electronic transmission, environmental control, energy storage, and transportation.

Introduction

Cryocoolers are small refrigerators that operate at temperatures below approximately 120 K and often provide only a few watts of cooling power. Unreliability, low efficiency, size, weight, vibration, and cost have been some of the major problems associated with these cryocoolers. The seriousness of any one of these problems obviously depends on the application. For example, the use of cryocoolers for cooling infrared sensors located on satellites has become very important over the past decade. This application requires a cooler with a very high reliability (involving 5- to 10-year lifetimes with no maintenance), high efficiency, small size, low weight, and low vibration. Cost generally has not been a problem since only a few cryocoolers are currently needed for this application and the cost is small compared to the total cost of the overall satellite system. However, with the recognition of large markets for cryocooler applications in the civilian sector, cost has become an additional driving force for the development of even more efficient and reliable devices.

Some of the current applications for cryocoolers, as enumerated by Radebaugh (1995), are listed in Table 1. The largest application for cryocoolers has been in the cooling of infrared sensors in satellites by the military. Stirling cryocoolers with a refrigeration capacity of about 0.25 W at 80 K have primarily been used for this application. However, the mean-time-to-failure of about 4,000 h is totally inadequate to meet the required needs of space operations. This reliability inadequacy of the Stirling cryocooler has provided the impetus for major innovations for such units and the rapid growth of research and development of pulse-tube refrigerators over the last five years. The largest application of cryocoolers in the commercial area has been in cryopumping use for the manufacture of semiconductors. Gifford-McMahon cryocoolers providing a few watts of refrigeration at a temperature of about 15 K have been the most popular devices for this area. This choice, because of cryocooler vibration characteristics, may change as semiconductor manufacturers are forced to achieve narrower linewidths in their computer chips to provide more compact packaging of semiconductor circuits.

Even though cryocoolers can be classified by the thermodynamic cycle that is followed, there is now general agreement that there are only two types: recuperative or regenerative units (Walker, 1983). A cryocooler is a recuperative type if only recuperative heat exchangers are used in the unit, and is a regenerative type if at least one regenerative heat exchanger (defined as a regenerator) is used in the unit. Figure 1 applies this classification scheme to five common cryocooler units.

Since the recuperative heat exchangers provide two separate flow channels for the refrigerant, the flow for the latter is always continuous and in one direction, analogous to a dc electrical system. This requires either the use of valving with reciprocating compressors and expanders or the use of rotary

Table 1. Cryocooler Applications

Military
- Infrared sensors for missile guidance
- Infrared sensors for surveillance (satellite based)
- Superconducting magnets for mine sweeping

Industrial and Commercial
- Superconductors for high-speed communication and voltage standards
- Semiconductors for high-speed computers
- Cryopumps for semiconductor manufacture
- Low-level moisture sensors for ultrapure gases
- Infrared sensors for process monitoring

Medical
- Superconducting magnets for MRI systems
- SQUID magnetometers for heart and brain studies
- Blood and semen storage
- Cryosurgery

Energy
- LNG production at remote gas wells and for peak shaving
- Infrared sensors for thermal-loss measurements
- SMES for peak shaving and power conditioning

Environment
- Infrared sensors for atmospheric studies of ozone hole and greenhouse effect
- Infrared sensors for pollution monitoring
- Cryotrapping air samples at remote locations

Transportation
- Infrared sensors for aircraft night vision
- LNG for fleet vehicles

Agriculture and Biology
- Biological specimens storage

Law Enforcement
- Infrared sensors for night vision

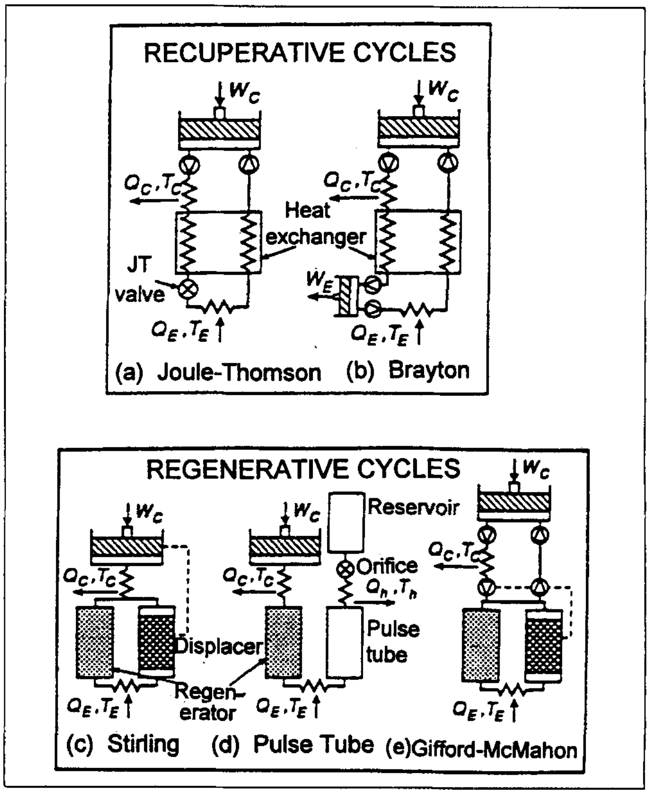

Figure 1. Most important recuperative and regenerative cycles used in cryocoolers.

or turbine compressors and expanders. In regenerative cycles the refrigerant flow is oscillatory in nature, analogous to an ac electrical system. This oscillatory effect allows the regenerator matrix to store energy for the first half of the cycle in a matrix material and release the energy during the next half of the cycle from the matrix material. To be effective, the solid matrix material placed within the regenerator must have a high heat capacity and good thermal conductivity. Some of the recent advances that have occurred in both types of cryocoolers are reviewed in the following sections.

Recuperative Systems

Joule-Thomson cryocoolers

The Joule-Thomson (J-T) effect of achieving cooling by throttling a nonideal gas is one of the oldest, but least efficient methods for attaining cryogenic temperatures. However, significant improvements in J-T cryocoolers have been achieved by applying novel methods of fabrication, incorporating more complex cycles, and utilizing special gas mixtures as refrigerants. These developments have made the J-T cryocooler competitive in many applications even when compared with cryocoolers that generally exhibit more efficient cooling cycles. Their relative simplicity combined with their small size, low mass, and absence from mechanical noise or vibration provide additional advantages for these small refrigerators.

Refrigeration in a J-T cooler, as shown in Figure 1a, is achieved by passing a high-pressure gas down a countercurrent heat exchanger, cooling it through the J-T effect by expansion to a low pressure, and then returning the cooled gas through the other side of the exchanger to precool the next incoming high-pressure gas. Repetition of this process step slowly cools the final stage until liquefaction of the gas occurs and a stable minimum temperature is reached corresponding to the existing pressure in the reservoir.

In the past, J-T cryocoolers have been fabricated by winding a finned, capillary tube on a mandril, attaching an expansion nozzle at the end of the capillary tube, and inserting the entire unit in a tightly fitting tube closed at one end with inlet and exit ports at the other end. During the past decade, Little (Little, 1990; Little and Sapozhnikov, 1994) has introduced a new method of fabricating J-T cryocoolers using a photolithographic manufacturing technique in which gas channels for the heat exchangers, expansion capillary, and liquid reservoir are etched on thin, planar, glass substrates that are fused together to form a sealed unit. These miniature cryocoolers have been fabricated in a wide range of sizes and capacities. A cryocooler operating at 80 K with a refrigeration capacity of 250 mW utilizes a heat exchanger whose channels have a width of 200 μm and a depth of 30 μm. The channels etched on the glass substrate must be controlled to a tolerance of ± 2 μm and the bond between the different substrates must withstand pressures on the order of 15 to 40 MPa. This cryocooler, used in spot cooling of electronic systems, has an overall dimension of only 7.5 cm$\times$1.4 cm$\times$0.2 cm.

Fabrication of miniature J-T cryocoolers by this approach has made it much simpler to use more complex refrigeration cycles and multistage configurations. The dual-pressure J-T cycle in which the refrigerant pressure is reduced by two isenthalpic expansions provides either a lower spot-cooling temperature or a higher coefficient of performance for the same power input. Attainment of 20 K requires two separate refrigeration systems in which a nitrogen J-T refrigeration stage provides the precooling to 77 K of a hydrogen J-T refrigeration stage. For 4 K, an additional helium stage of refrigeration cooled by the hydrogen stage is required. Use of multistage units in these miniature cryocoolers requires an order of magnitude better dimensional control of the etching process in order to match the desired flows and capacities specified for the heat exchangers, expansion capillaries, and liquid reservoirs. A thermodynamic analysis of dual-pressure and cascade J-T cycles is given by Timmerhaus and Flynn (1989).

To attain temperatures of 77 K, pure nitrogen has been used as the refrigerant in J-T cryocoolers. At 300 K, nitrogen must be compressed to a very high pressure (10–20 MPa) to achieve any significant enthalpy change. The high pressure required leads to a low compression efficiency with high stresses on compressor components, while the small enthalpy change results in a low cycle efficiency. The use of mixed refrigerants in the J-T cycle was first described by Podbielniak (1936) in a U.S. patent. More than 20 years later, Kleemenko (1959) used a gas mixture as a single-flow stream rather than the classical cascade system of three separate stages of cooling utilizing propane, ethylene, and methane, with a separate compressor for each stage. A modified version of the single-flow stream is now known as the mixed-refrigerant cascade (MRC) cycle used today for the liquefaction of natural gas. Temperatures down to 103 K were obtained by Fuderer and Andrija (1960) by using a mixed-gas refrigerant of equal amounts of nitrogen, methane, ethane, and propane operating with a 40:1 pressure ratio. Further work by Alfeev et al. (1973) with a gaseous mixture of 30 mol % nitrogen, 30 mol % methane, 20 mol % ethane, and 20 mol % propane achieved a temperature of 78 K using a 50:1 pressure ratio. The system efficiency with this gas mixture was 10 to 12 times better than when pure nitrogen was used as the refrigerant. Achieving temperatures below 70 K was attained by adding neon, hydrogen, or helium to the mixture.

The greatly enhanced cooling power of the gas mixtures when compared to that for pure nitrogen was essentially overlooked until Little (1984) verified the earlier claims and recognized the potential for the development of long-life, closed-cycle cryocoolers because of their increased reliability. In addition, he discovered (1985) that the addition of the fire-retardant Halon, CF_3Br, to the nitrogen–hydrocarbon mixture was sufficient to render the mixture nonflammable a characteristic retained in the resulting liquid solution down to 77 K or lower without precipitation because of the excellent solvent properties of the mixture. The net effect is that those needing refrigeration at nitrogen temperatures now have a series of nitrogen–hydrocarbon gas mixtures that are not only relatively safe to use but provide refrigeration efficiencies approaching 50% of Carnot (excluding compressor losses).

A simple example can highlight this advance. Consider the

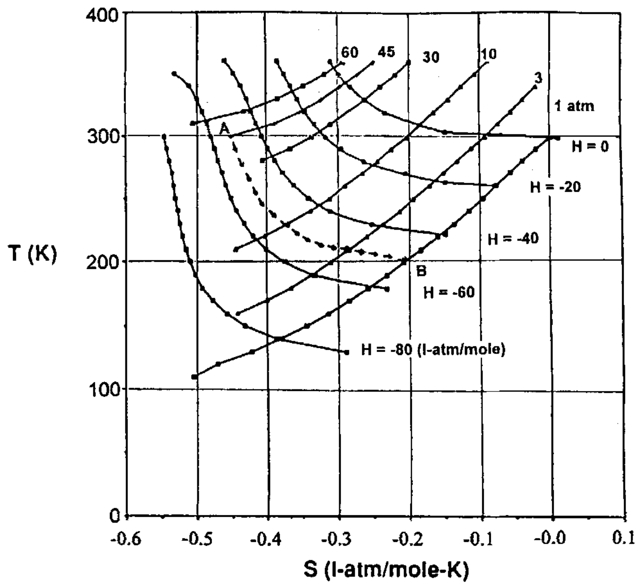

Figure 2. Isenthalpic expansion of a multicomponent gaseous mixture of 27% methane, 50% ethane, 13% propane, and 10% butane from 300 K and 4.5 MPa to 0.1 MPa (Little, 1989).

T-S diagram shown in Figure 2, as developed by Little (1989), for a hydrocarbon mixture of 27% methane, 50% ethane, 13% propane, and 10% butane on a volumetric basis. A throttling process for this gas mixture, initially at 300 K and 4.5 MPa, can ideally (constant enthalpy) achieve an exit temperature of 200 K at a final exit pressure of 0.1 MPa. Pure nitrogen gas undergoing a similar throttling process from the same inlet conditions to the same final exit pressure only achieves an exit gas temperature of 291 K. Thus, there can be as much as an elevenfold increase in the temperature drop of the refrigerant by utilizing the gas mixture instead of the pure nitrogen gas over these pressure and temperature conditions. From this one can conclude that refrigeration performance comparable to those using nitrogen gas at 12 to 15 MPa inlet pressures can be achieved with specific gas mixtures at pressures as low as 3 to 5 MPa.

The opportunity of using lower operating pressures has provided considerable incentive for the development of long-life, closed-cycle J-T cryocoolers. For example, Longsworth et al. (1995) recently described a closed-cycle system using a commercially available oil-lubricated compressor operating at about 2 MPa with a five-component nitrogen–hydrocarbon gas mixture to obtain 1 W of refrigeration at 80 K and 10 W at 93 K for an input power of 350 to 400 W. In this unit the inlet to the expansion valve is essentially all liquid, which entails extensive two-phase flow of the mixture in the recuperative heat exchanger. Considerable theoretical and experimental work is currently underway with various gas mixtures to obtain a better understanding of the optimum compositions and operating variables for specific cooling tasks and the effect of different multicomponent two-phase flow conditions that could be formed at the lower temperatures.

The effect of various gas-mixture concentrations on the efficiency of any cycle can be analyzed by evaluating the coefficient of performance (COP) of the cycle. The refrigeration effect of a J-T refrigerator is given by

$$\dot{Q}_r = \dot{n}(h_{\text{low}} - h_{\text{high}})_{\min} = \dot{n}\Delta h_{\min},$$

where $\dot{n}$ is the molar flow rate, h_{low} is the molar enthalpy of the low-pressure stream, and h_{high} is the molar enthalpy of the high-pressure stream in the recuperative heat exchanger. The $\Delta h_{\min}$ is the minimum difference in the molar enthalpies of the two streams for all locations throughout the length of the heat exchanger. The ideal work of compression is evaluated from

$$\dot{W}_{\text{ideal}} = \dot{n}[(h_{\text{high}} - h_{\text{low}}) - T_0(s_{\text{high}} - s_{\text{low}})] = \dot{n}\Delta g_0,$$

where s is molar entropy and the subscripts "low" and "high" refer to the inlet and exit locations of the compressor at the compression temperature T_0. The Δg_0 is the change in the molar Gibbs free energy also at T_0. The ideal COP of the refrigerant is then

$$\text{COP} = \dot{Q}_r/\dot{W}_{\text{ideal}} = \Delta h_{\min}/\Delta g_0.$$

This indicates that a maximum efficiency in the J-T cycle is achieved when the value of $\Delta h_{\min}/\Delta g_0$ is maximized in the temperature range of interest. A comparison of the temperature dependence of $\Delta h_{\min}/\Delta g_0$ for five refrigerant fluids operating over a pressure range between 0.1 and 5 MPa with a fixed compression temperature of 300 K is presented in Figure 3. Curve 1, representing pure nitrogen, indicates that this fluid has a small enthalpy difference at temperatures around 200 K, and this results in a COP as low as 0.03 at 300 K. At 80 K the COP increases to a value above 0.5, but the relative

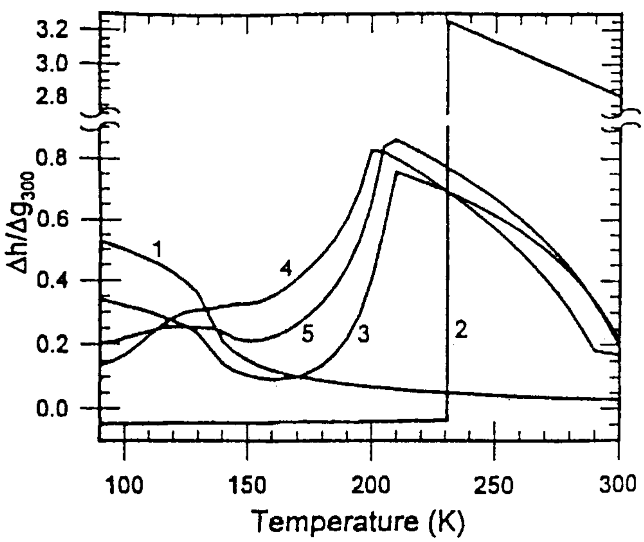

Figure 3. Temperature dependence of $\Delta h_{min}/\Delta g_0$ or COP between 0.1 and 5 MPa for different fluids.

Curve 1—pure nitrogen; curve 2—pure propane; curve 3—65 mol % nitrogen and 35 mol % propane; curve 4—30 mol % nitrogen, 30 mol % methane, 20 mol % ethane, 20 mol % propane; curve 5—40 mol % nitrogen, 17 mol % methane, 15 mol % ethane, 28 mol % propane.

Carnot efficiency is only 8.1% for the ideal cycle. Since pure propane has a large enthalpy difference at 300 K, curve 2 shows a COP of 2.8 at this temperature with a slight increase in value until the temperature approaches 230.7 K, the normal boiling point of propane. Further cooling by using propane cannot be achieved unless the pressure is reduced. Curve 3 for a binary 65 mol % nitrogen and 35 mol % propane mixture shows a minimum COP value at 160 K that is still more than three times the value obtained for pure nitrogen at 300 K. To increase the COP values at the intermediate temperatures requires the further addition of gases that exhibit intermediate boiling points as shown in curve 4 with the four-component mixture first used by Alfeev et al. (1973). With this mixture the COP has been increased to 0.103 at 80 K, resulting in a Carnot efficiency of 28.3% for the ideal cycle. Unfortunately this specific mixture still exhibits low values of $\Delta h_{\min}/\Delta g_0$ at high and low temperatures because of an excess of methane and ethane and a deficiency of nitrogen and propane. Curve 5, modified to meet these deficiencies, with 40 mol % nitrogen, 17 mol % methane, 15 mol % ethane, and 28 mol % propane provides an increase in the ideal COP to 0.19 at 80 K and a relative Carnot efficiency of 52% for the ideal cycle. In addition to the improved cooling effect of isenthalpic throttling of such mixtures, there is also an improvement in the cycle efficiency due to the enhanced heat transfer that occurs when a two-phase fluid is present in the recuperative heat exchanger. Further improvement of the J-T cycle can be anticipated during the coming decade.

Brayton cryocoolers

The other common recuperative cryocooler is the Brayton cycle refrigerator as shown in Figure 1b. The use of an expansion engine to carry out the expansion of the refrigerant leads to higher cycle efficiencies than are attainable with J-T cryocoolers. The Brayton cycle is commonly used in large liquefaction systems accompanied by a final J-T expansion. The units have a high reliability due to the use of turboexpanders operating with gas bearings. For small cryocoolers the challenge has been in fabricating the miniature turboexpanders while maintaining a high expansion efficiency and minimizing heat leak. This challenge has recently been addressed by Swift and Sixsmith (1993) with the development of a single-stage Brayton cryocooler with a small turboexpander (rotor diameter of 3.2 mm) providing 5 W of refrigeration at 65 K with neon as the working fluid. The compressor also utilizes gas bearings with an inlet pressure of 0.11 MPa and a pressure ratio of 1.6. The heat exchanger uses 300 slotted copper disks with slots 0.1 mm wide and 3 mm in length. The unit operates between 65 and 280 K with a Carnot efficiency of 7.7%. A still smaller unit is currently under development by McCormick et al. (1995) with a 2-W refrigeration capacity at 65 K. Unfortunately, the cost of these cryocoolers limits their service to space applications, which require high reliability, high thermodynamic efficiency, and low vibration.

Sorption cryocoolers

In an effort to eliminate the only moving part in the J-T cycle and thereby improve the reliability of this cycle, several research groups have recently examined the replacement of the mechanical compressor with a sorption compressor. The

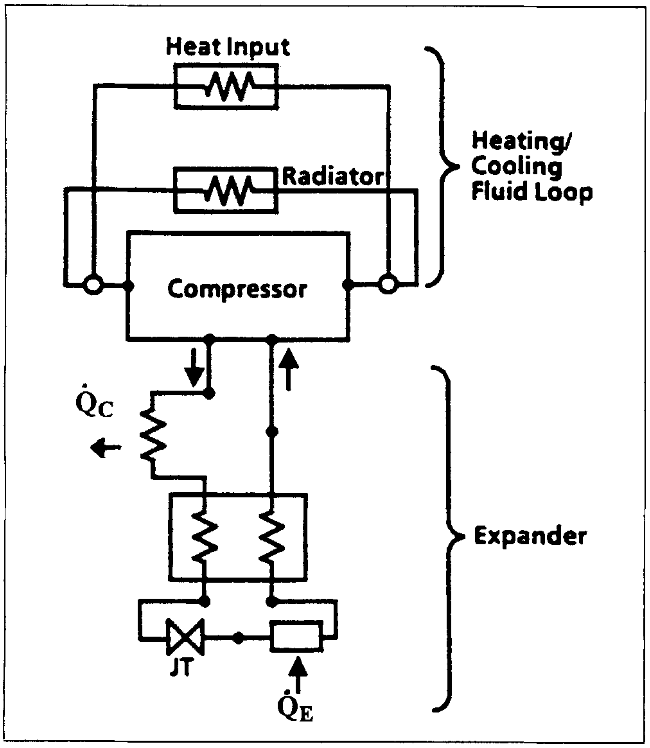

Figure 4. Heat-powered sorption cryocooler.

only moving parts in the sorption compressor are check valves that operate every few minutes to maintain steady flow in one direction.

Sorption refrigeration, as outlined in Figure 4 and summarized in detail by Wade (1992), is a method of cooling in which the gaseous refrigerant is compressed by means of a physisorption or a chemisorption process and directed through an expansion device, such as a J-T expansion valve, to obtain a net cooling. In the compressor portion of sorption refrigerators, low-pressure gas is sorbed to highly sorbent materials at prescribed temperatures. When the gas–solid system is heated with an increase in temperature by as much as 100 to 300°C, the desorbed gas is greatly pressurized and available to be used in a typical J-T refrigeration cycle to provide the required cooling.

The main disadvantage of sorption refrigerators is their high power requirement. Much of this power demand is related to the necessity of continuously cycling the temperature of the sorbent material over several hundred degrees. As the loading capacity of the sorbent decreases, the amount of sorbent required to produce a given mass of pressurized gas increases. Since all of the sorbent periodically must be heated and then cooled, increased sorbent mass results in significantly greater power requirements. The latter requirements have recently been decreased significantly by Bard and Jones (1990) with the use of regenerative heating of the sorbent bed in which heat from one sorbent unit in a cooldown state is used to heat another sorbent unit in a warm-up state. Several sorption units are needed to attain a reasonable efficiency.

In addition to the energy recovery in the compression stage, the efficiency of the compression process has been improved with the identification of better sorbent materials. Chemisorption of hydrogen with hydrides like vanadium and oxygen with praseodymium cerium oxide can be performed with the sorbents operating at 300 K and above. Most other sorbed gases require the use of physisorption on microporous carbon. Even here saran carbon (a high-density, high-surface-area carbon) has been identified as a sorbent with twice the sorption capacity of the commercially available Barneby-Cheney charcoal. With these improved sorbents and the use of the modified regenerative sorption compressor, the work input to obtain 1 W of refrigeration is approaching values of about 40 W/W at 125 K, 80 W/W at 65 K, and 195 W/W at 25 K. Additional improvements in the next few years, particularly if $LaNi_{4.8}Sn_{0.2}$ and ZrNi prove to be better hydride materials, can bring the power requirements of the sorption refrigerator at 25 K, as reported by Wade et al. (1994), closer to those exhibited by today's mechanically driven cryocoolers.

Regenerative Systems

Stirling cryocoolers

The Stirling refrigerator, which boasts the highest efficiency of all cryocoolers, is the oldest and most common of the regenerative systems. The elements of a Stirling refrigerator normally include two spaces of variable volume at different temperatures, coupled together through a regenerative heat exchanger, a heat exchanger rejecting the heat of compression at T_C, and a refrigerator absorbing the refrigeration effect of T_E. These elements can be arranged in a wide variety of configurations and operate as either single- or double-acting systems. The single-acting units are either two-piston or piston-displacer systems as shown in Figure 1c.

The ideal Stirling cycle consists of four processes. Step one involves an isothermal compression of the refrigerant in the compression stage at ambient temperature by rejecting heat, Q_C, to the surroundings. This is followed by a constant volume regenerative cooling where heat is transferred from the working fluid to the regenerator matrix. The reduction in temperature at constant volume causes a reduction in the pressure. The third step consists of an isothermal expansion in the expansion space at the refrigeration temperature T_E. In this step heat, Q_E, is absorbed from the surroundings of the expansion space. The process is completed by a constant-volume regenerative heating in which heat is transferred from the regenerator matrix to the working fluid. The increase in temperature at constant volume results in an increase in the pressure to the initial conditions. Successful operation of the cycle requires that the volume variations in the expansion space lead those in the compression space.

Thousands of small, single-stage Stirling cryocoolers have been manufactured worldwide by a dozen or more different companies. Capacities range from about 150 mW to 1 W at 80 K. Largest dimensions for the units (not including the compressor) are generally no more than 150 mm with masses less than 3 kg. Power inputs range from 40 to 50 W/W of refrigeration. This is equivalent to an efficiency of 6 to 7% of Carnot.

Many of the recent developments in Stirling cryocoolers have been directed toward improved reliability. In most applications, for example, linear motor drives have replaced the earlier rotary drives to eliminate many of the moving parts as well as to reduce the side forces existing between the piston and cylinder. Lifetimes of about 4,000 h are the norm with the linear compressors; however, Pruitt (1995) reports lifetimes greater than 15,000 h with the use of improved materials for the rubbing contact. Longer lifetimes required for space or specialized commercial applications will be achieved with the elimination of all rubbing contact. This, in part, has been achieved with piston devices by using flexure, gas, or magnetic bearings to purposefully center the piston and displacer in the cylinder housing. Diaphragm devices in which the flexing of the diaphragm results in a compression and expansion of the working gas have also provided very encouraging results.

The flexure bearing is the most common approach for centrally supporting either the piston or the displacer inside the appropriate cylinder while still permitting a clearance gap of 10 to 20 μm to serve as the necessary flow impedance for a dynamic seal. Figure 5 shows a simplified cross section of a Stirling compressor that utilizes the flexure bearing concept as originally developed by Davey (1990) at the University of Oxford. The flexure bearings provide a stiff support in the radial direction while acting as a weak spring in the axial direction. A similar arrangement is used to support the displacer.

The geometry of the spiral flexure bearing used in the Oxford Stirling cryocooler is also shown in Figure 5. The spiral flexures are fabricated from either beryllium copper or spring-grade stainless steel using a photoetching technique. A somewhat different flexure geometry has been developed by Wong et al. (1993), which uses linear arms instead of spiral arms.

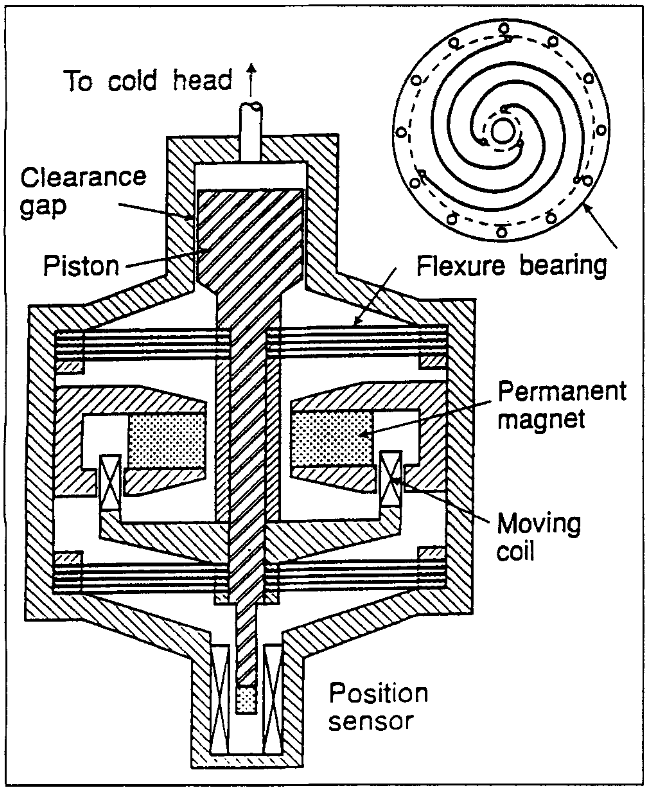

Figure 5. Cross section for an Oxford-type linear compressor using spiral flexure bearings.

Less work has been devoted to the use of gas bearings and magnetic bearings for eliminating the rubbing contact between the piston and cylinder. Duband et al. (1995) in France have reported on the use of hydrodynamic gas bearings in a two-stage Stirling cryocooler. A rotational frequency of 5 to 10 Hz was used to provide the gas-bearing effect in a 10-μm gap. The rotation to drive the gas bearing was provided by a brushless motor. Another description of hydrostatic gas bearing development is attributed to Smedley et al. (1995). In their Stirling cryocooler the gas flow for the bearing was driven by an oscillating pressure in the working space through a one-way valve and flow restriction. Magnetic bearings were first used in a NASA-supported one-stage Stirling cryocooler (Stolfi and Daniels, 1985). Even though the thermal performance and life testing were satisfactory, the cost and complexity of the magnetic bearing system discouraged further work on the project.

Reduction in the vibration associated with reciprocating motion of the linear motor drives is obtained by using dual opposed pistons, a passive balancer, or an active balancer. Typically, vibration forces are about 1 N for cryocoolers that have an input power on the order of 100 W. Axial vibration forces can be reduced to less than 0.1 N in flexure-bearing cryocoolers by using active harmonic nulling with dual opposed pistons (Mon et al., 1995). Because of the much greater radial stiffness shown by the linear flexure bearing proposed by Wong et al. (1993), there is speculation that the radial vibration modes could also be reduced down to this same level.

Pulse-tube cryocoolers

As noted earlier, space applications place additional requirements on cryocoolers with lifetimes of 10 to 15 years, low mass, and low energy consumption. These considerations have led researchers over the past decade to concentrate on the orifice pulse-tube refrigerator (OPTR), shown schematically in Figure 1d. The latter is a variation of the Stirling cryocooler in which the moving displacer is replaced by a pulse tube, orifice, and reservoir volume. A detailed review of pulse-tube refrigerators is given by Radebaugh (1990).

The OPTR operates on a cycle similar to the Stirling cycle, except that the proper phasing between mass flow and pressure is established by the passive orifice rather than by the moving displacer. In this cycle, a low-frequency compressor raises the pressure of the helium refrigerant gas to a level between 0.5 and 2.5 MPa during the first half of a sinusoidal compression cycle. (The oscillating pressure for the OPTR can be provided either by a compressor similar to that used in the Stirling refrigerator or by a Gifford-McMahon compressor that has been modified with appropriate valving to achieve the required oscillating pressure.) The high-pressure gas, after being cooled in the regenerator, adiabatically compresses the gas in the pulse tube. Approximately one-third of the compressed gas originally in the pulse tube flows through the orifice to the reservoir volume, with the heat of compression being removed in the hot exchanger. During the latter half of the sinusoidal cycle, the gas in the pulse tube is expanded adiabatically, resulting in a cooling effect. The cold expanded gas is forced past the cold heat exchanger by the gas returning from the reservoir volume through the orifice. The pulse tube serves as a buffer volume of gas that allows a temperature gradient to exist between the hot and cold ends of the pulse tube. Obviously, some mixing or turbulence is present within the buffer volume since recent experiments by Rawlins et al. (1994) have shown that the time-averaged enthalpy flow that represents the gross refrigeration capacity is only 55 to 85% of the ideal enthalpy flow. Considerable work is currently underway to determine the fluid dynamics that occur within the pulse tube. One such study (Lee et al., 1993) has used a hot-wire smoke technique to visually observe radial mixing and flow streaming over the length of the pulse tube.

By assuming simple harmonic pressure, mass flow, and temperature oscillations within the entire pulse-tube refrigerator as well as adiabatic operation within the pulse tube itself, researchers at the National Institute of Standards and Technology (NIST) (Storch and Radebaugh, 1988) have been able to develop an analytical model that provides a reasonable prediction of the refrigeration performance. More recent thermoacoustic theories (Xia, 1993, 1995) include a linear approximation with higher harmonics and realistic heat transfer and viscous effects between the gas and pulse-tube wall. The losses accounted for in these models result in a time-averaged enthalpy flow that is in reasonable agreement with the experimental values obtained earlier by Rawlins et al. (1994). Additional work is needed to clearly identify the existing loss mechanisms.

The orifice concept for the pulse-tube refrigerators was first introduced by Mikulin et al. (1984) and was further revised to the schematic shown in Figure 1e by Radebaugh et al. (1986) to achieve a refrigeration temperature of 60 K with only one stage. Further refinement in the operation of the refrigeration system permitted achieving temperatures below 40 K. However, since the Carnot efficiencies for the OPTR were still considerably less than those obtained from the improved Stirling refrigerators, the latter remained the choice for most space applications in the late 1980s in spite of the other advantages of the OPTR refrigerator.

Improved efficiencies for pulse-tube refrigerators with higher operating frequencies were achieved by Zhu et al. (1990) with the addition of a second orifice, as shown in Figure 6. The addition of this orifice, identified as the double-inlet concept, permits the gas flow needed to compress and expand the gas at the warm end of the pulse tube to bypass the regenerator and pulse tube. The reduced mass flow through the regenerator reduces the regenerator losses, particularly at high frequencies where these losses become rather large. Addition of the second orifice, described earlier, can reduce the refrigerator temperature by at least 15 to 20 K in a well-designed pulse tube operating at frequencies of 40 to 60 Hz. This has been substantiated by Ravex et al. (1992) with a temperature of 28 K, the lowest temperature achieved to date with a single-stage, double-inlet arrangement.

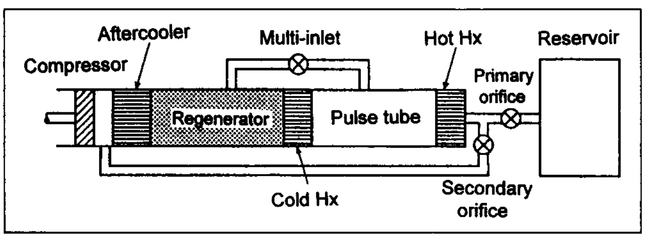

Figure 6. Double-inlet pulse-tube cryocooler utilizing a secondary orifice.

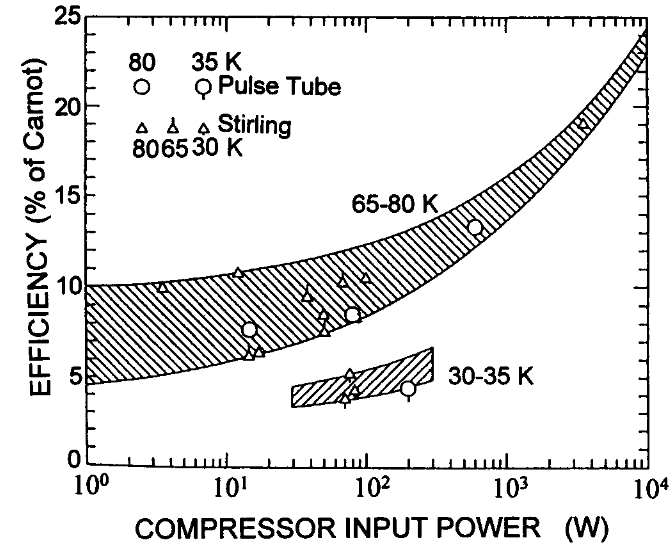

Figure 7. Comparison of the percent of Carnot efficiency for several recent pulse-tube cryocoolers with similarly powered Stirling cryocoolers (Radebaugh, 1995).

A comparison of the percent of Carnot efficiency obtained for the improved pulse-tube refrigerators with the Stirling refrigerators is shown in Figure 7. The shaded area represents the efficiency range obtained for most of the recent Stirling refrigerators, while the circles represent the individual efficiencies obtained from recent pulse-tube refrigerators. The highest power and highest efficiency unit is the NIST pulse-tube refrigerator discussed by Radebaugh (1995). This unit provided 31.1 W of refrigeration at 80 K with a rejection temperature of 316 K, equivalent to a relative Carnot efficiency of 13%. The average operating pressure was 2.5 MPa, while the operating frequency was maintained at 4.5 Hz. The other two circles with lower efficiencies in the 65–80 K range represent the efficiencies obtained by Chan et al. (1993) for the same unit but with different input powers and different cold end temperatures. The efficiency shown in the 30–35 K range is for a small unit developed by Burt et al. (1995). Even though the data for pulse-tube refrigerators are still rather limited, it is evident that the efficiencies of the most recent pulse-tube refrigerators are becoming quite competitive with those of the best Stirling refrigerators of comparable size.

Two or more pulse-tube refrigerator stages are normally used to maintain high efficiency when temperatures below about 80 K are desired. The purposes of the staging are (1) to provide net cooling at an intermediate temperature, and (2) to intercept regenerator and pulse-tube losses at a higher temperature. Three distinct methods exist for the staging arrangement. The first uses a parallel arrangement of a separate regenerator and pulse tube for each stage with the warm end of each pulse tube located at ambient temperature. In the second method, shown in Figure 8, the warm end of the lower stage pulse tube is thermally anchored to the cold end of the next higher stage in a series configuration. The third method, introduced by Zhou et al. (1993), uses a third orifice to permit a fraction of the gas removed from an optimized

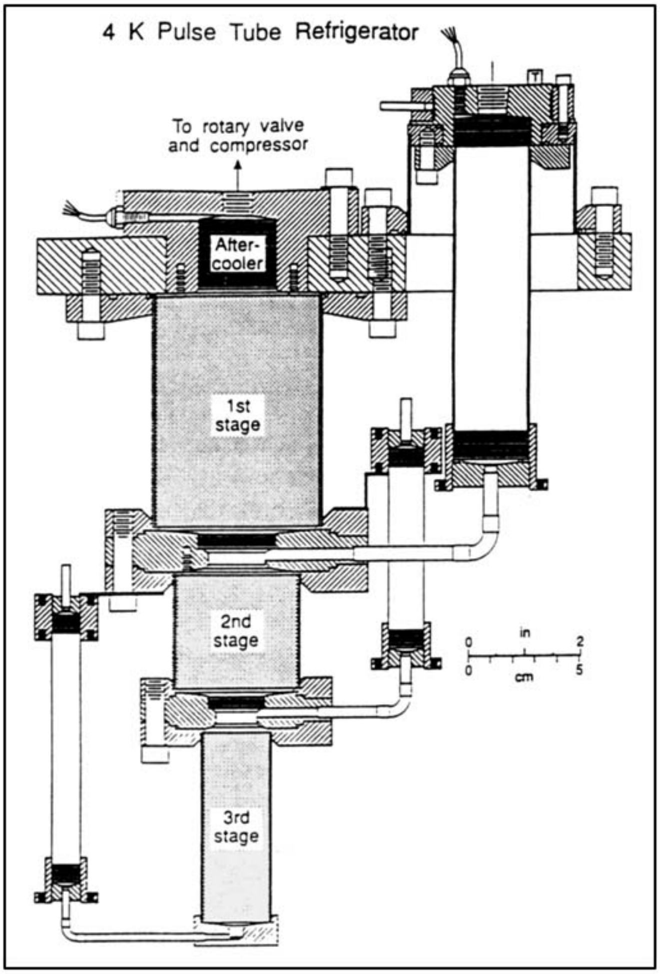

Figure 8. Three-stage orifice pulse-tube cryocooler for liquefying helium at 4.2 K.

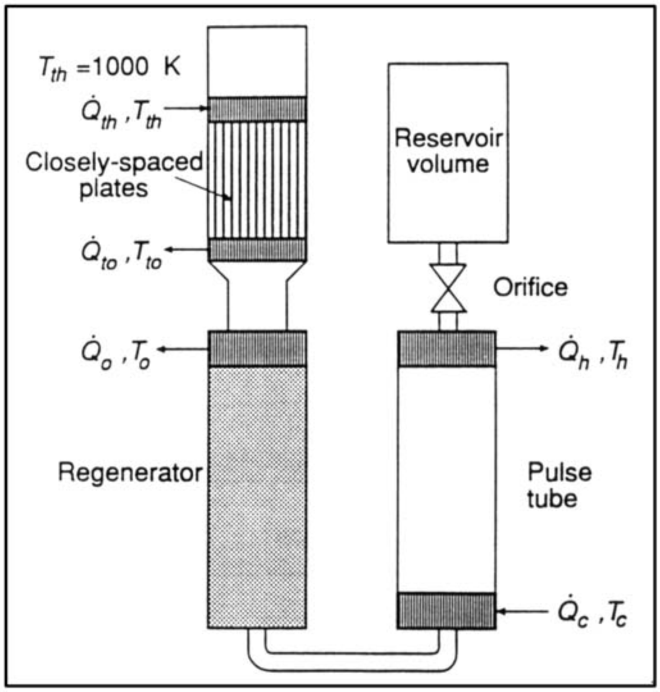

Figure 9. Thermoacoustic driven orifice pulse-tube refrigerator (TADOPTR).

location in the regenerator to enter the pulse tube at an intermediate temperature. This staging configuration, identified as the multi-inlet arrangement, maintains the simple geometrical arrangement of a single pulse tube, although it would normally require a change in diameter at the tube junction with the pulse tube to maintain a constant gas velocity in the pulse tube. The lowest temperature reported with a three-stage parallel arrangement of pulse-tube refrigerators was 3.6 K achieved by Matsubara and Gao (1994).

A recent joint program between NIST and the Los Alamos Scientific Laboratory (Radebaugh et al., 1991) used a thermoacoustic driver (TAD) instead of a mechanical compressor to drive the orifice pulse-tube refrigerator. In the arrangement shown in Figure 9, spontaneous acoustic oscillations are generated in the working fluid when the temperature gradient in the closely spaced plates of the thermoacoustic driver exceeds a critical value. Because there are no moving parts in this pulse-tube refrigerator, commonly referred to as a TADOPTR, this cryocooler has the potential for long-term reliability. The first prototype of this cryocooler with a thermoacoustic driver of about 10 m resonated at a frequency of 40 Hz with a pressure ratio of 1.10 to achieve a refrigeration temperature of 90 K. More recent thermoacoustic drivers have attained pressure ratios of about 1.20 and operate at much higher resonant frequencies to reduce the length of the driver unit. Finally, we mention, but do not discuss, another joint program currently underway between NIST, Los Alamos Scientific Laboratory, and Cryenco to develop a 1,900-L/d gas-fired TADOPTR to provide liquefied natural gas as an alternative fuel for fleet vehicles.

Gifford-McMahon cryocoolers

As noted earlier, J-T cryocoolers utilizing pure-gas refrigerants require very high operating pressures; therefore, most commercial closed-cycle cryocoolers use one or more expanders to achieve part of the cooling effect. One of the most widely used regenerative cryocoolers is the Gifford-McMahon refrigerator, shown in Figure 1e. These units can achieve temperatures of 65 to 80 K with one stage of expansion and 15–20 K with two stages of expansion. Precooling of the expansion stage is accomplished with regenerators utilizing carefully selected matrix materials. Since regenerators are essentially storage devices of energy, the matrix materials must possess a high heat capacity as well as a good thermal conductivity. Lead shot has been the regenerator material selected for regenerative cryocooler operation between 10 and 65 K. However, its heat capacity becomes ineffective below this temperature range. Without a suitable matrix material, addition of a third stage with its accompanying regenerator has made it impossible for any regenerative cryocooler to achieve a temperature below 10 to 12 K. Thus, to attain a temperature of 4.2 K to reliquefy helium boil-off has required a two-stage Gifford-McMahon refrigerator equipped with a J-T loop using a compact countercurrent heat exchanger (Longsworth, 1986).

An increase in the specific heat of a material at these low temperatures can only occur if there is some physical transi-

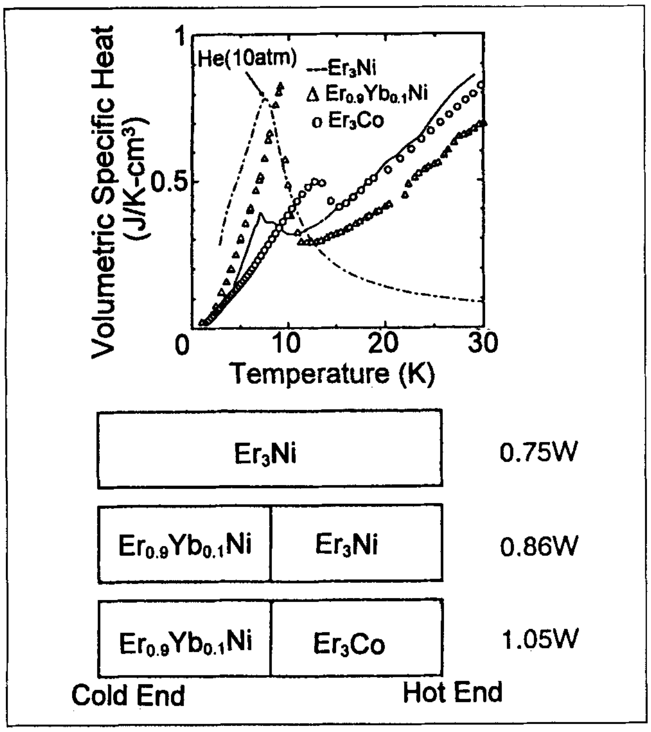

Figure 10. Specific heat for several rare-earth matrix materials between 2 and 30 K.

The use of layered rare materials provides increased cooling capacity. (Courtesy of T. Kuriyama, Toshiba.)

tion occurring in the material. Since many of the heavy rare-earth compounds exhibit a magnetic-phase transition at these low temperatures, considerable efforts have been directed to study these materials by a number of laboratories around the world. One of the most effective groups in this effort has been the one led by Hashimoto et al. (1992) at the Tokyo Institute of Technology. Some of their experimental results, as well as those of others, are shown in Figure 10. Following up on their earlier discoveries, this same research group (Hashimoto et al., 1994) has investigated the specific heats of the $ErN_{1-x}Co_x$ system and determined that the specific heats of this system are more than twice those obtained for Er_3Ni at 7 K. Use of some of these more effective regenerator matrix materials in the cold end of the second stage of a Gifford-McMahon refrigerator has increased the refrigeration power, as noted in Figure 10, over that when only Er_3Ni was used.

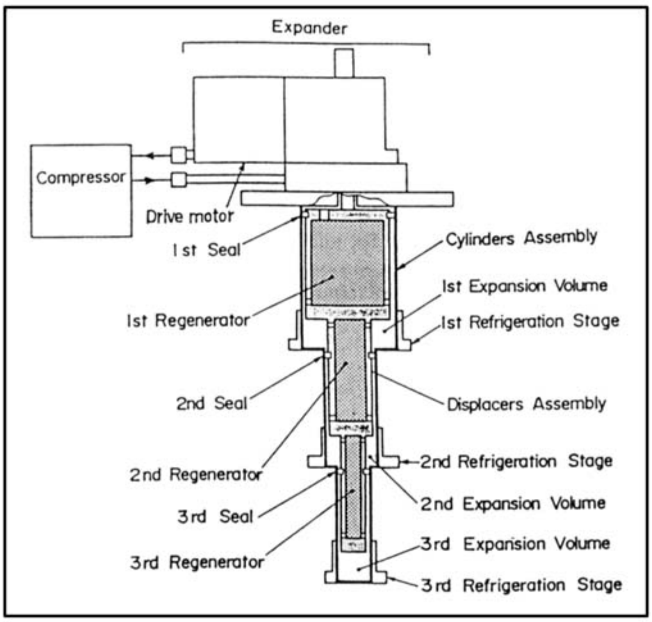

Figure 11. Cross section of a 4 K Gifford-McMahon refrigerator using rare-earth regenerator materials in the third stage.

From Hashimoto et al. (1992).

The availability of the new rare-earth compounds has made it possible to use a three-stage Gifford-McMahon refrigerator to provide sufficient cooling to reliquefy helium boil-off from the superconducting magnets serving MRI units. Nagao et al. (1994) have described such a device, shown in Figure 11, that utilizes $Er_{1.5}Ho_{1.5}Ru$ as the regenerator matrix material in the third-stage regenerator providing a refrigeration capacity of more than 150 mW at 4.2 K. Preliminary results show that this unit, which is smaller in size than the conventional 4 K Gifford-McMahon refrigerator discussed earlier, also has greater reliability as well as lower operating costs.

Summary

A number of recent advances in cryocoolers are discussed in this summary. These developments include new fabrication techniques for miniature J-T cryocoolers, the use of gas mixtures to improve the efficiency of J-T refrigerators, the incorporation of flexure and gas bearings in the compressor and expander units of Stirling cryocoolers to increase operational life, the use of sorption compressor and thermoacoustic drivers to replace the mechanical compressor and pressure oscillators in cryocoolers, and the development of new matrix materials for regenerators operating over the 4 to 10 K temperature range.

It is readily apparent that these recent improvements in cryocoolers have provided major advances in meeting the requirements set for the commercial application of cryocoolers. The increased reliability of these units provides assurance that a transition will occur from the present aerospace/military applications to civilian uses in many diverse areas, such as electronic transmission, process and environmental control, energy storage, medicine, and transportation.

Acknowledgment

Financial support of the National Science Foundation to this project is acknowledged.

Literature Cited

Alfeev, V. N., V. M. Brodyansky, V. M. Yagodin, V. A. Nikolsky, and A. V. Ivantso, U.K. Patent No. 1,336,892 (1973).

Bard, S., and J. A. Jones, "Regenerative Sorption Compressors for Cryogenic Refrigeration," *Advances in Cryogenic Engineering*, Vol. 35, Plenum Press, New York, p. 1357 (1990).

Burt, W. W., and C. K. Chan, "Demonstration of a High Performance 35 K Pulse Tube Cryocooler," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 313 (1995).

Chan, C. K., C. B. Jaco, J. Raab, E. Tward, and M. Waterman, "Miniature Pulse Tube Cooler," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 113 (1993).

Davey, G., "Review of the Oxford Cryocooler," *Advances in Cryogenic Engineering*, Vol. 35, Plenum Press, New York, p. 1423 (1990).
Duband, L., A. Ravex, and P. Rolland, "Development of a Stirling Cryocooler Using Hydrodynamic Gas Bearings," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 47 (1995).
Fuderer, M., and A. Andrija, "Verfahren zur Tiefekühling," German Patent No. 1,426,956 (1960).
Hashimoto, T., K. Gang, H. Makuucki, R. Li, A. Onishi, and Y. Kanazawa, "Improvement of Refrigeration Characteristics of the G-M Refrigerator Using New Magnetic Regenerator Material $ErNi_{1-x}Co_x$ System," *Advances in Cryogenic Engineering*, Vol. 39, Plenum Press, New York, p. 1335 (1994).
Hashimoto, T., M. Ogawa, A. Hayashi, M. Makino, R. Li, and K. Aoki, "Recent Progress on Rare Earth Magnetic Regenerator Materials," *Advances in Cryogenic Engineering*, Vol. 37, Plenum Press, New York, p. 859 (1992).
Kleemenko, A. P., "One Flow Cascade Cycle," *Proc. Int. Cong. of Refrigeration*, Paris, p. 34 (1959).
Lee, J. M., P. Kittel, K. D. Timmerhaus, and R. Radebaugh, "Flow Patterns Intrinsic to the Pulse Tube Refrigerator," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 125 (1993).
Little, W. A., and I. Sapozhnikov, "Development of Low Cost Cryogenic Refrigeration System for Cooling of Cryoelectronics," *Advances in Cryogenic Engineering*, Vol. 39, Plenum Press, New York, p. 1467 (1994).
Little, W. A., "Advances in Joule-Thomson Cooling," *Advances in Cryogenic Engineering*, Vol. 35, Plenum Press, New York, p. 1305 (1990).
Little, W. A., "Recent Developments in Joule-Thomson Cooling: Gases, Coolers and Compressors," *Proc. Int. Cryocooler Conf.*, Naval Postgraduate School, Monterey, CA, p. 3 (1988).
Little, W. A., U.S. Patent No. 919,699 (1985).
Little, W. A., "Microminiature Refrigeration," *Rev. Sci. Instrum.*, **55**, 661 (1984).
Longsworth, R. C., M. J. Boiarski, and L. A. Klusmier, "80 K Closed-Cycle Throttle Refrigerator," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 537 (1995).
Longsworth, R. C., "A 4 K Refrigerator and Interface for MRI Cryostats," *Advances in Cryogenic Engineering*, Vol. 31, Plenum Press, New York, p. 517 (1986).
Matsubara, Y., and J. L. Gao, "Novel Configuration of a Three-Stage Pulse Tube Refrigerator for Temperatures Below 4 K," *Cryogenics*, **34**, 259 (1994).
McCormick, J. A., W. L. Swift, and H. Sixsmith, "Miniaturization of Components for Low Capacity Reverse Brayton Cryocoolers," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 507 (1995).
Mikulin, E. I., A. A. Tarasov, and M. P. Shkrebyonock, "Low-Temperature Expansion Pulse Tubes," *Advances in Cryogenic Engineering*, Vol. 29, Plenum Press, New York, p. 629 (1984).
Mon, G. R., G. T. Smedley, D. L. Johnson, and R. G. Ross, Jr., "Vibration Characteristics of Stirling Cycle Cryocoolers for Space Applications," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 197 (1995).
Nagao, M., T. Inaguchi, H. Yoshimura, S. Nakamura, T. Yamada, T. Matsumoto, S. Nakagawa, K. Moutsu, and T. Watanabe, "4 K Three Stage Gifford-McMahon Cycle Refrigerator for MRI Magnet," *Advances in Cryogenic Engineering*, Vol. 39, Plenum Press, New York, p. 1327 (1994).
Podbielniak, W. J., "Art of Refrigeration," U.S. Patent No. 2,014,725 (1936).
Pruitt, G. R., "Hughes Long-life Linear Stirlings: A Status Report," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 113 (1995).
Radebaugh, R., "Recent Developments in Cryocoolers," *Proc. Int. Cong. Refrigeration*, Vol. IIIb, Paris, p. 973 (1995).
Radebaugh, R., K. M. McDermott, G. W. Swift, and R. A. Martin, "Development of a Thermoacoustically Driven Orifice Pulse Tube Refrigerator," *Proc. Int. Meeting on Cryocoolers*, Navy Rep. DTRC-91/003, p. 205 (1991).
Radebaugh, R., "A Review of Pulse Tube Refrigeration," *Advances in Cryogenic Engineering*, Vol. 35, Plenum Press, New York, p. 1191 (1990).
Radebaugh, R., J. Zimmerman, D. R. Smith, and B. Louie, "A Comparison of Three Types of Pulse Tube Refrigerators: New Methods for Reaching 60 K," *Advances in Cryogenic Engineering*, Vol. 31, Plenum Press, New York, p. 779 (1986).
Ravex, A., P. Rolland, and J. Liang, "Experimental Study and Modelisation of a Pulse Tube Refrigerator," *Proc. Int. Cryogenic Engineering Conf.*, ICEC Supplement, *Cryogenics*, **32**, 9 (1992).
Rawlins, W., R. Radebaugh, P. E. Bradley, and K. D. Timmerhaus, "Energy Flows in an Orifice Pulse Tube Refrigerator," *Advances in Cryogenic Engineering*, Vol. 39, Plenum Press, New York, p. 1449 (1994).
Smedley, G. T., R. G. Ross, Jr., and D. M. Berchowitz, "Performance Characterization of the Sunpower Cryocooler," *Cryocoolers*, Vol. 8, Plenum Press, New York, p. 149 (1995).
Stolfi, F. R., and A. Daniels, "Parametric Testing of a Linearly Driven Stirling Cryogenic Refrigerator," *Proc. Int. Cryocooler Conf.*, NBS Special Publication 698, p. 80 (1985).
Storch, P., and R. Radebaugh, "Development and Experimental Test of an Analytical Model of the Orifice Pulse Tube Refrigerator," *Advances in Cryogenic Engineering*, Vol. 33, Plenum Press, New York, p. 851 (1988).
Swift, W. L., and H. Sixsmith, "Performance of a Long Life Reverse Brayton Cryocooler," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 84 (1993).
Timmerhaus, K. D., and T. M. Flynn, *Cryogenic Process Engineering*, Plenum Press, New York (1989).
Wade, L. A., J. J. Wu, and S. Bard, "Performance, Reliability, and Life of Hydride Compressor Components for 10 to 30 K Sorption Cryocoolers," *Advances in Cryogenic Engineering*, Vol. 39, Plenum Press, New York, p. 1483 (1994).
Wade, L. A., "An Overview of the Development of Sorption Refrigerators," *Advances in Cryogenic Engineering*, Vol. 37, Plenum Press, New York, p. 1095 (1992).
Walker, G., *Cryocoolers*, Plenum Press, New York (1983).
Wong, T. E., R. B. Pan, and A. L. Johnson, "Novel Linear Flexure Bearing," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 675 (1993).
Xiao, J. H., "Thermoacoustic Theory for Regenerative Cryocoolers: A Case Study for a Pulse Tube Refrigerator," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 305 (1993); "Thermoacoustic Heat Transportation and Energy Transformation, Part 1: Formulation of the Problem," *Cryogenics*, **35**, 15 (1995); "Part 2: Isothermal Wall Thermoacoustic Effects," *Cryogenics*, **35**, 21 (1995); "Part 3: Adiabatic Wall Thermoacoustic Effects," *Cryogenics*, **35**, 27 (1995).
Zhou, Y., and Y. J. Han, "Pulse Tube Refrigerator Research," *Proc. Int. Cryocooler Conf.*, Air Force Rep. PL-CP-93-1001, p. 147 (1993).
Zhu, S., P. Wu, and Z. Chen, "Double Inlet Pulse Tube Refrigerators: An Important Improvement," *Cryogenics*, **30**, 514 (1990).

Manuscript received Jan. 11, 1996, and revision received May 3, 1996.