Sorption Compressor/Mechanical Expander Hybrid Refrigeration

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Experience with DSN ground-based cryogenic refrigerators has proved the reliability of the basic two-stage Gifford-McMahon helium refrigerator. A very long life cryogenic refrigeration system appears possible by combining this expansion system or a turbo expansion system with a hydride sorption compressor, in place of the usual motor-driven piston compressor. To test the feasibility of this system, a commercial Gifford-McMahon refrigerator was tested using hydrogen gas as the working fluid. Although no attempt was made to optimize the system for hydrogen operation, the refrigerator developed 1.3 W at 30 K and 6.6 W at 60 K. The results of this test and of theoretical performances of the hydrid compressor are presented, coupled to these expansion systems.

I. Introduction

Sorption refrigeration is a method of cooling wherein gas is compressed by means of physical surface adsorption or chemical internal absorption, and then passed through an expansion device such as a Joule-Thomson (J-T) expansion valve, thus creating net cooling. For example, hydrogen is chemically absorbed into certain rare Earth metallic compounds such as $LaNi_5$. At room temperature, the partial pressure of hydrogen on $LaNi_5$ is about 2 atm, while at 373 K, the $LaNi_5$ hydride decomposes and the hydrogen is released at 40 atm. By heating and cooling a series of such canisters [1], a continuous flow of high-pressure hydrogen gas can be generated, and this gas can be expanded to provide cooling to liquid hydrogen temperatures (14 K to 33 K). A somewhat similar process occurs between charcoal and nitrogen, except that the nitrogen is physically adsorbed onto the solid charcoal surface rather than chemically adsorbed. Cooling to about 80 K or above can be obtained with this method. Charcoal adsorption tends to occur best at temperatures below 250 K and desorption occurs best above 400 K.

Both physical and chemical sorption cooling have recently been demonstrated [2], [3], and [4], and the results are most encouraging in terms of extending the life of both ground-based and flight-borne refrigeration systems to 10 years or more. Since sorption systems have virtually no wear-related moving parts except for very long life, room temperature, low frequency valves, they have a potential life expectancy of

many decades. Recent tests have demonstrated hydride sorption compressor operation for 5800 h [2], hydride cryogenic systems operation between 14 K and 29 K for over 1000 h [2] and feasibility testing of charcoal/nitrogen cryogenic systems between 100 K and 120 K [4].

Since sorption systems are powered by low-grade heat (above approximately 400 K), space-borne systems are particularly attractive applications. Direct solar heat can be used or even waste heat from a radioactive thermoelectric generator. Unfortunately, sorption refrigeration also has its disadvantages. Although hydrogen is absorbed at temperatures above 300 K, nitrogen requires adsorption temperatures generally below 250 K. In a low Earth orbit, this temperature requires fairly large and heavy radiators, unless a fluid loop is used to reject the heat at varying levels, e.g., 400 K, 300 K, 250 K, etc.

An alternative to nitrogen physical adsorption for cooling in the 65 K to 100 K region is an oxygen chemisorption J-T refrigeration system. This system, which is still in an experimental phase at JPL (NASA patent pending), may eventually prove to be very efficient both in terms of power and weight.

For the present, however, alternatives to J-T expansion of sorbed gas are Gifford-McMahon (G-M) expansion or turbo-expansion. These alternative expansion systems would allow hydride compressors to function as the gas supplier for a first-stage refrigeration system (65 K to 100 K). Due to the high inversion temperature (about 200 K) of hydrogen gas, a first-stage refrigeration system is necessary to precool the gas before it is expanded in a J-T system. The G-M or turbo-expanded hydride system could then serve as a first stage for a hydride J-T system.

As mentioned, the hydride compressor itself has virtually no wear-related moving parts and is thus expected to last at least 10 years [2]. Present G-M expanders typically have a 20,000 h to 30,000 h mean time between maintenance (MTBM) [5], and they are lightweight as well as highly reliable. A 9 kg expander can provide about 15 W of cooling at 70 K. For the past two decades, G-M expanders have been used as the primary means of cooling for almost all ground-based cryogenic radio antenna systems. Furthermore, their failure mode in the presence of trace contamination is in the form of gradual deterioration of performance, as opposed to catastrophic failure, as is the case with most other mechanical expanders. Lightweight redundancy or multiple units with lower duty cycles are clearly possible. Furthermore, systems research can likely increase the present single-unit G-M maintenance-free life to 40,000 to 50,000 h (P. Kerney, private communication, CTI Cryogenic, Waltham, MA, 1985, 1986 and R. Longsworth, private communication, Air Products Corp., Allentown, PA,

1986). Gas-bearing turbo-expanders, by contrast, have no wear-related moving parts and can thus be expected to last at least 10 years. They also are very lightweight (less than 1 kg for 15 W of cooling at 70 K).

The following section describes data obtained by running a G-M expander with hydrogen and helium gas. Based on these measurements, as well as predictions of turbo-expander hydrogen performance, various system performance estimates are made for hydride-driven G-M and turbo-expansion refrigerators.

II. Gifford–McMahon Expansion

A. Test Results

A series of tests was performed with an unmodified stock CTI Model 21 two-stage G-M refrigerator, using both helium and hydrogen as the working fluid. The data for both first- and second-stage performance for helium are shown in Fig. 1. With no power on the second stage, the refrigerator first-stage performance varied from 2 W at 40 K to 6 W at 70 K. Alternatively, with no power on the first stage, the second-stage performance varied from 1 W at 15 K to 2.75 W at 30 K. The helium flow rate was measured as 7.5 SCFM (0.587 gm/sec) when the pressure ratio was set at 18.2×10^5 Pa/6.89×10^5 Pa (18.0 atm/6.8 atm). (Note that 1 atm = 1.0133×10^5 Pa.) In general, these values are all quite close to those predicted by CTI for the Model 21.

The performance of an open-cycle hydrogen G-M system operating off of bottled hydrogen gas was generally somewhat higher for the first stage, but lower for the second stage. A variety of pressure ratios were tested in order to determine how overall system performance varied (Fig. 2). With a pressure ratio of 18.69 atm/6.78 atm and a flow rate of 9.7 SCFM hydrogen (0.381 gm/sec), approximately 6.6 W of power was obtained at 60 K with no second-stage load. When the pressure ratio was increased to about 18.7 atm/2.4 atm, the cooling power increased to about 9.3 W at 60 K with 11.4 SCFM (0.446 gm/sec) hydrogen. This compares with the helium measured cooling rate of about 4.7 W at 60 K with 7.5 SCFM (0.587 gm/sec) helium.

With no load on the first stage, the second-stage hydrogen performance was severely degraded compared to the helium performance. Even with a very high pressure ratio (18.7 atm/ 2.7 atm), the refrigerator produced only about 1.3 W at 30 K, compared to about 2.75 W at 30 K for helium at 18.0 atm/ 6.8 atm. The lower second-stage hydrogen performance is likely due to partial liquefaction of the hydrogen gas, which has a saturation temperature of about 24 K at 2.7 atm. Partial liquefaction in the regenerator could cause thermal "short-

circuits" and greatly reduce overall second-stage regenerator performance. It should be mentioned that the G–M test unit was not optimized for specific use with hydrogen, and therefore this data should not be used to make a final comparison between helium and hydrogen operation. The actual unit tested was designed for use with helium, as are all present G–M units, and thus some improvement may be expected if the regenerator, rpm rate, etc., are specifically designed for hydrogen gas.

B. Analysis

Based on the test data, a plot of usable cooling power at 65 K vs hydrogen pressure ratio is shown in Fig. 3. The first-stage regenerator effectiveness, ϵ, for a G–M refrigerator may be considered to vary from about 99.57 percent [6] to 98.5 percent [7]. For the analytical calculations, a regenerator effectiveness of 99.0 percent is assumed. For ground operations, a 300 K heat sink is reasonable, while for Earth orbital low power operations, 200 K is attainable by means of direct passive radiators for high Earth orbits, or by thermoelectric coolers for low Earth orbits.

A computer program has been developed at JPL for predicting hydride refrigerator performance [8]. The program agrees well with hydride J–T test data, and has shown that high-pressure ratio hydride compressor systems take only slightly more power to operate than do low-pressure ratio hydride compressor systems, because most of the energy input to a hydride compressor is used to liberate the hydrogen gas, while a much smaller part is used to compress the gas, or increase the overall thermal capacity of the system.

Based on the values shown in Fig. 3, approximately 8.2 ±0.5 kW of power is necessary to generate 11.4 SCFM of hydrogen gas in the pressure range of $0.1 < P_L/P_H < 0.6$. Thus, the required specific power of a hydride-powered G–M system can be considered to vary from about 1000 W/W at $P_L/P_H = 0.36$ and 300 K heat sink to as little as 645 W/W at $P_L/P_H = 0.1$ and 200-K heat sink (ϵ = 99.0 percent). A summary of the required hydride compressor specific powers is given in Table 1.

III. Turbo Expansion

Based on calculations made by Walter Swift of Creare, Inc., Handover, NH (private communication, 1986) a turbo expander for hydrogen gas can be made as small as about 1/8-in. diameter for a 5-W load at 65 K. With a pressure ratio of 2 atm/0.2 atm, a turbo expander can be expected to have up to about 40 percent of ideal expansion (isentropic) efficiency. Significantly larger turbo-expanders, e.g., 100 W at 65 K, can be expected to have efficiencies as high as 80 per-

cent of isentropic expansion. Unfortunately, virtually no test data presently exist for miniature hydrogen turbo expanders, although miniature helium turbo expanders have generally ranged in the 30 to 60 percent isentropic efficiency range (W. Swift, private communication, Creare, Inc., Hanover, NH, 1986).

A sorption computer program was used to compute the necessary power to generate turbo-expanded cooling, at 65 K, assuming a counterflow heat exchanger effectiveness of 98 percent, a pressure ratio of 2 atm/0.2 atm, and an isentropic expansion efficiency of 40 percent. For a 300 K initial hydrogen temperature, the required specific power was 326 W/W, and for a 200 K heat sink, the required specific power was 259 W/W. When the isentropic expansion efficiency was increased to 80 percent, as would be the case for a larger turbo expansion system, the required specific powers were then cut by more than half. A summary of the results of the turbo expansion analyses is shown in Table 2.

IV. Summary and Conclusions

The hydrogen/G–M test results and the analyses of the hydride/G–M and hydride/turbo expansion refrigerators indicate that both systems use significantly more power than other developmental long-life refrigeration systems. In general, small developmental mechanical systems require between 40 W/W to 100 W/W for 65 K cooling [9] while the small hydride/G–M system can be expected to require about 1000 W/W for ground applications (300 K heat sink), or 645 W/W for space applications (200 K heat sink). The hydride/turbo expander is also predicted to require a high amount of power (although somewhat lower than the hydride/G–M system). For ground applications, calculations have predicted that small coolers require about 326 W/W, and space coolers require about 260 W/W. Large hydrogen turbo or G–M expansion systems, e.g., 100 W at 65 K, are likely to require less than half as much specific power.

Although these first-stage sorption refrigeration systems require considerably more power than their mechanical refrigeration counterparts, they do have some specific advantages. First, they can be operated directly from low-grade waste heat (e.g., solar or radioactive waste heat) as opposed to electricity. Hydride compressors operate quite satisfactorily with heat source temperatures between 75° C to 100° C. Second, and probably most importantly, they have a very long predicted lifetime (at least 50,000 h MTBM). Due to their simplicity and a minimum of wearing parts, their reliability should be extremely high. Although both first-stage systems are above the air freezing temperature of about 64 K, the G–M system is likely to be somewhat more "forgiving" in regard to gas contamination than the turbo-expanded system. Third,

for the case of the hydride/turbo expansion combination, this refrigerator offers nearly vibrationless operation. The very tiny turbo expander is supported on a thin film of gas and rotates typically at several hundred thousand rpm. The resulting vibration is virtually below the measurement threshold, as is the vibration due to the gas flow in the reversible chemical-hydride compressor. This lack of vibration can be of crucial importance for highly sensitive infrared detector applications. It should be mentioned, however, that the vibration level of the G–M system can be minimized by counterbalancing with a second G–M unit.

Finally, although these individual first-stage sorption systems require a relatively high amount of power, when used in series with lower-staged sorption refrigeration systems for temperatures below 15 K, the overall power and weight of the multi-staged sorption systems can be significantly less than multi-stage mechanical systems [9].

References

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**Table 1. Hydride-powered G-M performance estimates
for 10 W at 65 K**

P_L/P_H	Spec Power @ 300 K Heatsink	Spec Power* @ 200 K Heatsink
0.10	1000	645
0.36	1330	720

*These values assume the test regenerator effectiveness was 99.0%.

Table 2. Hydride-powered turbo performance estimates

Cooling Power @ 65 kW	Turbine Efficiency, %	Spec Power @ 300 K Heatsink	Spec Power @ 200 K Heatsink
5	40	326	259
100	80	125	114

Assumptions: $\epsilon_{HX} = 98\%$
$P_L/P_H = 0.10$
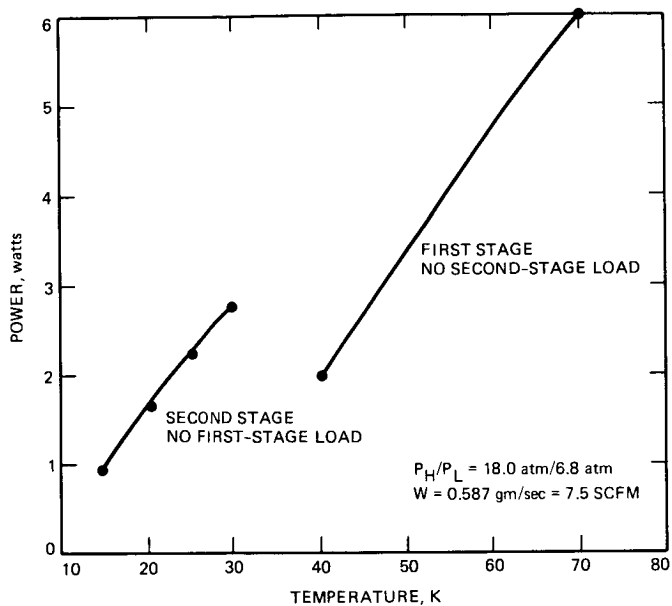
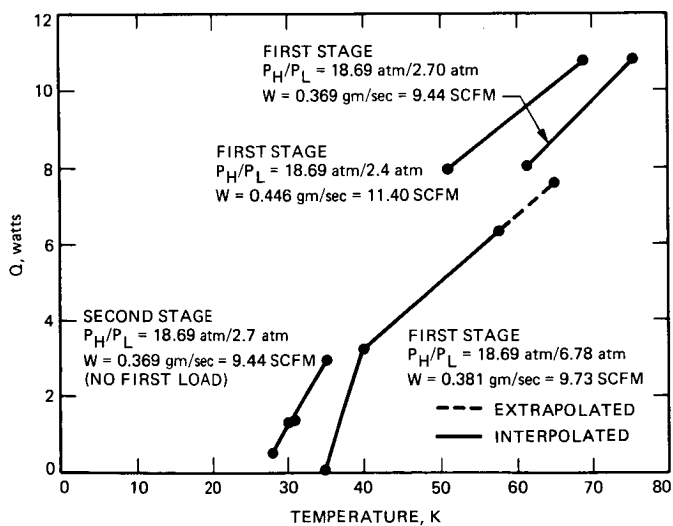**Fig. 1. CTI model 21 G–M performance with helium**

P_H/P_L = 18.0 atm/6.8 atm
W = 0.587 gm/sec = 7.5 SCFM
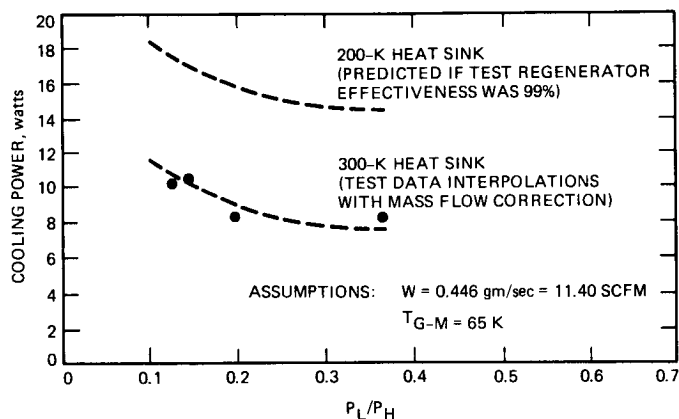
FIRST STAGE
NO SECOND-STAGE LOAD

SECOND STAGE
NO FIRST-STAGE LOAD

Fig. 3. Hydrogen G–M cooling powers for varying P_L/P_H

200-K HEAT SINK
(PREDICTED IF TEST REGENERATOR
EFFECTIVENESS WAS 99%)

300-K HEAT SINK
(TEST DATA INTERPOLATIONS
WITH MASS FLOW CORRECTION)

ASSUMPTIONS: W = 0.446 gm/sec = 11.40 SCFM

T_{G-M} = 65 K

Fig. 2. CTI model 21 G–M performance with hydrogen

FIRST STAGE
P_H/P_L = 18.69 atm/2.70 atm
W = 0.369 gm/sec = 9.44 SCFM

FIRST STAGE
P_H/P_L = 18.69 atm/2.4 atm
W = 0.446 gm/sec = 11.40 SCFM

SECOND STAGE
P_H/P_L = 18.69 atm/2.7 atm
W = 0.369 gm/sec = 9.44 SCFM
(NO FIRST LOAD)

FIRST STAGE
P_H/P_L = 18.69 atm/6.78 atm
W = 0.381 gm/sec = 9.73 SCFM

- - - EXTRAPOLATED
—— INTERPOLATED