# CRYOGENICS OPTIONS FOR THE MMA

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## REQUIREMENTS

We begin by assuming that the required cooling at each antenna is 1.0W at 4.0K, 5W at 20K and 20W at 50K. I believe that it should be a major goal of the thermal design of the receiver to limit the requirements to no more than this. However, the actual requirements will not be known until the design is much further developed, so these numbers merely serve as a common basis of comparison for various approaches, with a belief that they are more or less in the right ballpark. Changing the required power at each temperature by a factor of 2 in either direction should not change the qualitative conclusions, and the systems should scale accordingly. Reducing the lowest temperature to 3.5K or 3.0K causes a large reduction in efficiency for some approaches, so this could affect the choice.

The design goal is to achieve substantially better overall efficiency (power input required) compared to systems of similar performance used at the NRAO in the past, while improving (or at least maintaining) reliability.

IDEAL EFFICIENCY

It can be shown that the thermodynamic efficiency of a cooling engine can never exceed that of the Carnot cycle, which gives:

$$Q/W = T_c / (T_h - T_c)$$

where $Q$ is the rate of heat removal from the load, $W$ is the input power, $T_c$ is the cold (load) temperature and $T_h$ is the temperature at which the heat is dumped to the environment. For our three load temperatures, taking $T_h = 290K$ in all cases, we get:

$Q/W\ (4K) = .014$, $W(4K) = 71.5$ W;

$Q/W\ (20K) = .074$, $W(20K) = 67.5$ W;

$Q/W\ (50K) = .208$, $W(50K) = 96$ W;

This gives a total power of 235 W. In practice, the Carnot efficiency is never achieved, nor even approached. As shown in Fig 1, efficiencies range from a few percent to about 20 percent of the Carnot value, with the higher efficiencies being achieved only in large systems (Q>1kW) at the higher temperatures (>30K). NRAO systems with similar performance use 7 to 10 kW, which implies 2.3% to 3.3% of Carnot efficiency.

To achieve highest efficiency, it is necessary that both the expansion of the working fluid at the cold end of the cycle and its compression at the hot end be isothermal. That is, heat must be absorbed and dissipated at constant temperature. This cannot be done in practice. In addition, high efficiency requires that losses due to friction, undesired thermal paths, and dead volume (parts of the engine containing working fluid that does not contribute to a segment of the thermodynamic cycle) be minimized. Various practical systems achieve this to different degrees.

Fig 1: (Walker 1983, Fig 1.4.) Efficiency as a function of refrigerating capacity and temperature.

## EXPERIENCE AT THE NRAO

For cooling to 50K and 20K, we have used commercial Gifford-McMahon (G-M) refrigerators. These are available from several manufacturers; most of our experience is with CTI and Balzers. Essentially all of the input power goes into a compressor that is mounted remotely from the refrigerator and operates asynchronously. Its job is to compress the helium working fluid from about 5 atm to about 15 atm and then cool it to room temperature, while maintaining sufficient flow rate for the refrigerator. The refrigerator manufacturers also supply the compressors, but at the NRAO we have usually assembled our own compressors from major components purchased from third parties. Because power consumption has heretofore not been a major concern, and to limit the number of different compressor configurations that must be supported, those built by both the NRAO and the refrigerator manufacturers have tended to be over-designed. This has produced efficiencies lower than what is feasible. Some data is given in Table 1 (refrigerators) and Table 2 (compressors).

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Table 1: G-M Refrigerators at the NRAO

| Mfgr | Model | 77K load | 20K load | Min T |
|---|---|---|---|---|
| CTI | 22 | 10 W | 1 W | 12 K |
| CTI | 350 | 20 | 3 | 12 |
| CTI | 1020 | 50 | 10 | 12 |
| Balzers | UCH130 | 115 | 19 | 6.5 |

Note: Load on each stage is for zero load on the other stage.

Minimum temperature is at second stage with no external loads.

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Table 2: Helium Compressors

| Mfgr/Model | Type | Pressures | Flow | Power In |
|---|---|---|---|---|
| CTI/8200 | Rotary | 17, 5.5 atm | 29 Nm^3/h | 2.2 kW |
| CTI/9600 | Rotary | 17, 5.5 | 75 | 5.5 |

| | | | | |
|---|---|---|---|---|
| NRAO/2.5hp | Scroll[1] | 20, 5.5 | 42 | 2.0? |
| NRAO/5hp | Scroll[2] | 20, 5.5 | 88 | 4.0? |
| NRAO/JT | Scroll[3] | 20, 4, 0.5 | 85, 1.7 | 7.5? |

[1] Using Hitachi 250RHH pump.

[2] Using Hitachi 500RHH pump.

[3] Two-stage, Hitachi 250RHH followed by 500RHH. Low flow into 1st stage for J-T circuit.

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For cooling to the 3-4K range, all of our systems have used a Joule-Thompson (J-T) expander with pre-cooling of the helium working fluid by a two-stage G-M refrigerator. This is usually thought of as a three-stage system. The same G-M stages also provide cooling for other components (including radiation shields) at the higher temperatures. Again, all of the power for the J-T stage goes into a remote compressor, which must pump from about 0.5 atm to about 20 atm at a flow rate of about 1 SCFM (1.7 Nm^3/hr) per watt of cooling at 4K. Due to the fairly high compression ratio, a two-stage compressor is usually required. Traditionally, all components of the J-T stage have been built at the NRAO, including recuperative heat exchangers and compressors. Compressors have been assembled from readily available components, leading to over-design and imbalance between stages, and producing lower than attainable efficiency.

THEORETICAL CONSIDERATIONS AND KNOWN TRADEOFFS

Among the different thermodynamic cycles, the efficiency of practical systems decreases in the order Stirling (integral), split Stirling, Brayton, Gifford-McMahon, Joule-Thompson (see Fig. 2). (I'm not yet sure where the pulse tube fits into this list; see later discussion.) For this reason, Stirling refrigerators have been subjected to intense development in recent years for space applications. Their main disadvantage is that the compression must be cyclic and synchronized with the expansion (displacer motion). It's therefore best to mount the compressor and expander together and drive them from a single motor, but this leads to vibration and to a need to dissipate large amounts of heat near the cold head. The compressor can be separated by a pipe and driven from a separate motor (split Stirling), but then the efficiency is degraded because the pipe represents dead volume for the working fluid, and propagation delay through the pipe may constrain the cycle period.

Fig. 2: (Walker, 1983, Fig 2.7.) Comparison of efficiency of different cycles as a function of temperature.

The J-T cycle is inherently quite inefficient, but it is rarely used by itself for cooling to 4K. Other cycles, typically G-M, are used to pre-cool the J-T working fluid. (A pure J-T system for 4K would use several stages, with different working fluids, since He is needed for the last stage but its inversion temperature is about 40K.) Nevertheless, a large amount of energy is typically used by the final stage. (Cooling from 300K to 30K has the same Carnot efficiency as cooling from 30K to 3K. If the latter uses a less efficient cycle, than its power input will dominate.)

An noted earlier (Fig. 1), larger systems tend to achieve higher efficiency. This seems to hold across all types of thermodynamic cycles. It is probably due to small systems being dominated by parasitic losses (friction, conduction leaks, imperfect seals, etc.) whose size is roughly proportional to the perimeters of pipes and cylinders, or the areas of mating surfaces; whereas the desired heat pumping ability is proportional to the flow rate and volume of the working fluid. This means that the pumping ability scales with a larger exponent than the losses, favoring the larger systems.

Finally, note that for some cycles (including G-M), the efficiency depends strongly on the operating conditions (Fig. 3). As the ratio of maximum to minimum pressure over the cycle increases, the efficiency falls. This is because the main inherent inefficiency of such cycles comes from void volume in the regenerator and incomplete expansion of the working fluid. In a G-M engine, expansion is achieved by letting the gas expand into the low-pressure side of the compressor, not by any change in the volume of the expansion space. Furthermore, gas in the regenerator is both compressed and expanded, and so does not contribute to refrigeration. At higher pressure ratios, the mass of gas that fails to refrigerate for these reasons becomes larger, yet that gas still must pass through the compressor, consuming power. Neglecting losses, the Carnot efficiency is approached as the pressure ratio approaches unity. Nevertheless, practical G-M refrigerators tend to use pressure ratios of 3 to 4 because for a given size (and hence cost) of machine, the total cooling power increases with pressure ratio (and hence total mass flow), albeit slower than the increase in power consumed.

Fig. 3: Efficiency vs. pressure ratio (Walker 1983, Fig 5.16).

## OPTIONS AVAILABLE FOR THE MMA

A. Construct three-stage G-M/J-T systems along the lines of our past experience. This is the most conservative approach. We avoid the need for much development, and we have considerable confidence that it will work correctly and that we understand how to operate and maintain it. But we give up the possibility of improved reliability and reduced power consumption, which would have a substantial effect on operating cost.

B. Like A, but expend a substantial development effort on just the compressor part in an attempt to optimize its design. At first glance, it looks feasible to reduce power consumption by a factor of 2.

C. Purchase commercial 2-stage G-M systems that achieve 1W at 4K. These have been available for about 5 years, and are currently produced by at least three manufacturers (see Table 3), but they have not been used at the NRAO. The advantage would be simplicity and (presumably) better reliability than A or B, but probably higher initial cost. Power consumption would probably be slightly better than A but worse than B. Experiments should be done in which we adjust the pressure ratio and cycle frequency in an attempt to maximize efficiency at some sacrifice in cooling power.

D. Investigate substantially different approaches that promise much higher efficiency but for which no commercially-available products exist yet. These include Stirling cycle, reverse-Brayton cycle, and pulse-tube coolers. There are also some interesting new developments in optical cooling (crystals which cool to cryogenic temperature when pumped by a laser). Of these, the pulse tube technology looks most promising for our purposes. There is a substantial literature on and considerable laboratory experience with such devices. We could not select one of them for the MMA without a lot of in-house experience, which would take an investment of manpower and time to acquire. If the final MMA choice can be deferred for two years, then this investment could be worthwhile.

Table 3: Commercial 4K G-M Cryocoolers

| Mfgr | Model | Load at 4.2K | Min Temp | System Price | Input Power |
|---|---|---|---|---|---|
| Daikin[1] | CSW210 | 0.8 W | 3.0 K | $35k | 6.7 kW |
| Sumitomo[2] | SRDK408 | 1.0 | 3.1 | 39k | 7.5 |
| Sumitomo[2] | SRDK405 | 0.5 | 3.1 | 24k | 2.5 |
| Leybold | 4.2GM | 0.5 | 3.4 | 37.5k | 6.5 |

Notes:

[1] Represented in U.S. by APD Cryogenics, Inc.

[2] Represented in U.S. by Janis. Purchased by Princeton U. Physics; see memo on visit, 27 June 1997.

## REFERENCE