MMA MEMO No. 266

JOULE-THOMPSON CRYOCOOLER STAGE FOR SIS MIXERS: STRAWMAN DESIGNS

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Presented here are outlines of two designs for cryocoolers that might be used with the MMA/ALMA SIS mixer receivers. We concentrate mainly on the last stage of cooling, which in both cases is a Joule-Thompson expander. The first case is the current, preliminary design (further discussed in [1]), intended to provide 0.7 W of cooling at 4.0K at the mixer mounting plate. The second is intended to provide 0.7 W of cooling at 2.5K. For comparison, some data is also presented on the 4K, 1.5 W coolers now used at the NRAO 12m telescope and elsewhere.

The main point is to establish the level of difficulty involved in reaching the lower temperature. Before deciding the final specification for the telescope, this difficulty should be weighed against any improvement in receiver performance that might result.

DESIGN SUMMARY

	MMA 4.0K	MMA 2.5K	NRAO12m 4K
LHe bath temperature	3.8K	2.3K	4.2K
Capacity @ spec'd temp	0.7W	0.7W	1.5W
Precooling	13K 10atm NIST PTR2	13K 10atm NIST PTR2	12K 20atm Balzers UCH130
Heat exch. #3 efficiency	95%	95%	90%
Mass flow	0.10 g/s	0.10 g/s	0.13 g/s
Return pressure drop	0.14 atm	0.03 atm	0.15 atm
Pressure at compressor	0.51 atm	.036 atm	0.85 atm
Vol flow at compressor	8.01 m^3/h	61.5 m^3/h	3.39 m^3/h
Compressor	2stg linear CFIC 600W	3stg hybrid Vac pumps+ CFIC linear	2stg scroll 5HP/stage

For the MMA designs, the liquid helium bath temperature is set slightly below the specified mixer mounting plate temperature in order to allow for a thermal resistance of 0.3K/W between them. In practice it is difficult to do much better than this.

Much of the performance data on the existing 4K cryocooler was obtained from recent laboratory measurements, to be reported in detail separately. These measurements include the heat exchanger efficiency and pressure drop, which are critical to the design. Additional information can be found in [3].

Heat exchanger efficiency is here defined as the ratio of the actual rate of heat transfer to the rate that would occur if the temperature difference between the two streams were zero at the warm end, all else being equal.

NOTES ON MMA 4.0K DESIGN

1. The JT circuit can be driven by an oil−free, wear−free linear compressor that is commercially available. Such a compressor can operate tilted, so it can be connected by hard plumbing to the dewar on an antenna. This eliminates several of the worst causes of failure and maintenance expense in JT refrigerators: oil contamination, compressor wearout, and flexible hose leakage. The required performance can be achieved with cold components similar to those used now, except that the efficiency of the last heat exchanger should be improved from the present 90% to about 95%.

2. Precooling can be achieved by a relatively small refrigerator, here suggested to be a 2−stage pulse tube device now under development for the NRAO at NIST in Boulder. A CTI model 350 GM refrigerator is of similar capacity and could also be used. But either of these requires improved efficiency in the precooling heat exchangers compared with present practice in order to minimize the precooling load. This is believed to be achievable.

3. Additional improvements in the JT circuit are under development. These affect reliability and maintainability but not the basic performance parameters. Included are porous plug JT valves and reactive gas cleaning.

NOTES ON MMA 2.5K DESIGN

1. All notes on the 4K design are applicable here too.

2. The return flow pressure drop must be reduced substantially from the present 0.14 atm, since the pressure at the 2.3K LHe bath must be .066 atm. We assume a pressure drop of .03 atm. This must be done along with improving the efficiency of the heat exchangers, which leads to a conflict: the longer heat exchangers needed for efficiency will have larger pressure drops, all else being equal. Since the assumed drop is already nearly half of the LHe pressure, any larger drop rapidly increases the volume flow rate and thus the compressor size.

3. It is very difficult to find a pump that will handle the large volume flow of 61.5 m^3/h at .036 atm inlet pressure. For example, the 5HP Hitachi scroll pumps (model 500RHH) have a pumping rate of 12.8 m^3/h at 5.7 atm inlet (from manufacturer’s specs), and our measurements show that this drops to 8.0 m^3/h at 0.3 atm inlet. Thus, at least 8 such pumps would have to be operated in parallel as the first compressor stage. On the other hand, these pumps support a large pressure difference (5−10 atm), so maybe only one more small stage would be needed, except that the temperature rise will probably force us to at least three stages (adiabatic compression of He from .03 atm at 300K to 1.0 atm raises its temperature to 890K). We therefore choose a 3−stage arrangement and provide the following

example of a possible design:

Stage 1	4 ea. Balzers model DUO016B rotary vane vacuum pumps operating in parallel (16.2 m^3/h each) Power: 420Wx4 = 1.7 kW Volume: .025x4 = 0.10 m^3 close packed; allow 0.2 m^2 total Outlet pressure: 1.0atm
Stages 2-3	CFIC 2-stage linear compressor 3.7 m^3/h flow Total compression ratio 18 Power: ~600W Volume: .08 m^3 incl panel and heat dissipation

However, the 2nd and 3rd stages could also use oil-filled rotary vane or scroll pumps, which would be cheaper. The linear compressor is under consideration for the 4.0K design because it should produce much higher system reliability; but here we must use an oil-filled pump for the first stage, so the advantages of the linear compressor have already been largely lost.

4. Although the above compressor plan can support the flow rate, the first stage's array of pumps may not be able to dissipate the resulting heat. Special modifications may be needed for heat removal, or even more pumps may have to be used, or additional stages (with interstage cooling) may be needed.

5. Although the internal (cold) parts of the 2.5K cooler would be essentially the same as for the 4.0K cooler, the external parts (compressor) would be many times larger and would consume many times more power. This in itself will reduce reliability. In addition, the planned improvements in reliability from using a modern compressor and avoiding flexible hoses would not be possible.

6. Achievement of the specifications may prove to be impossible or impractical because of heat exchanger performance. Keeping the return pressure drop sufficiently low will be very difficult. A factor of 4.6 below that of the existing hardware is suggested here; if only a factor of 3 is achieved, the already-huge volume flow rate to the compressor would be doubled.

POTENTIAL COMPROMISES

The burden of reaching the lower temperature could be eased by relaxing either the capacity specification or the temperature specification or both. For example, reaching 3.0K at 200mW capacity is relatively easy (see, e.g., [2]).

The 0.7W capacity specification includes a 2x margin over calculated requirements [1], but some experts argue that this margin is insufficient in view of the uncertainties involved. Work is underway to reduce the uncertainties, but a large reduction in required capacity is unlikely without major changes in the planned receivers. For example, placing all receivers for <200 GHz on a separate cryocooler at 4.0K and cooling only the higher frequency receivers to 3.0K might reduce the load for the latter set to <200mW.

In that case, only one cooler could be operated at a time (due to limited space and power for compressors), leading to long cooling delays for some band changes. The overall system complexity would be increased, reducing reliability.

REFERENCES

[1] L. D'Addario, "Cryogenics," Section 6 of MMA Project Book, version 2.50 of 1999-May-02.

[2] W. Hilgerath and B. Vowinkel, "Closed cycle refrigerator for superconducting mm-wave mixers." Cryogenics, v 25, pp 573--577, 1985.

[3] D. Schroeter and J. Lamb, "Characterization and optimization of the NRAO/Balzers three stage closed cycle JT refrigerator." NRAO Electronics Div Internal Report No. 297, Nov 1994.