DEMONSTRATION OF A 10 K STIRLING COOLER FOR SPACE APPLICATIONS

G.R. Baker, D. Féger, A.S. Gibson, A. Little
AstriumUK Ltd. - Gunnels Wood Road, Stevenage, Hertfordshire, SG1 2AS, UK
Tel (44) 1438 77 4461, Fax (44) 1438 77 4344, andrew.gibson@astrium-space.com

T.W. Bradshaw, M.R. Crook, A.H. Orlowska
Rutherford Appleton Laboratory -Chilton, Didcot, OX11 0QX, UK

B.J. Tomlinson
Air Force Research Laboratories, Kirtland Air Force Base, New Mexico, USA

A. Sargeant
Cubic Applications Inc. - 4055 Hancock Street, Suite 115, San Diego, California 92110-510 USA

ABSTRACT

The performance capability of a two-stage Stirling cycle cooler intended for space applications in the range of 10 K has been demonstrated and characterised over a range of interface temperatures. To achieve the desired performance, several design improvements were made to an existing qualified cooler, originally intended for application in the 20-50 K range. The modifications have extended the ability of the cooler to provide 45 mW of heat lift at 10.4 K, with a base temperature at 9.4 K. Significant advances have been made despite the inherent limitations of regenerator material heat capacity and the constraints of Carnot efficiency at these low cryogenic temperatures.

Background information is provided on the closed cycle mechanical cooler with an outline of the design, including a summary of the developments to improve regenerator performance, compressor capacity and reduction of parasitic heat losses in the system. Results of recent characterisation testing are presented herein. These results are compared with calculated performance from modeling, and reviewed in light of design improvements that are proposed to further reduce the base temperature and increase heat lift of the cooler accordingly.

INTRODUCTION

The 10K two-stage cooler has been developed as an extension of an existing qualified Astrium/RAL 20-50 K cooler, for the purpose of cooling of arsenic-doped silicon sensors for Very Long Wave Infrared (VLWIR) detection.

The original cooling requirement originated from the US Air Force Research Laboratory (AFRL) specification of 500 mW heat lift at 35 K. These requirements were updated in 1997 to reflect the operational temperature and heat lift requirements of Si:As detectors, having re-established the load point target to be 45 mW of heat lift at 10.3 K.

Figure 1. 10 K Closed Cycle Stirling Cooler System

It was proposed that the revised AFRL specification be addressed using a modified two-stage Stirling cooler using rare-earth regenerator materials and an increased swept volume.[1] Astrium pursued this approach in collaboration with RAL, utilising the combined experience gained in the design and testing of coolers developed for the 20-50 K temperature range under development contract with the European Space Agency.

10 K COOLER HERITAGE

The basis of Astrium's cooler technology is the development by Oxford University and the Rutherford Appleton Laboratory (RAL) of an 80 K cooler design that was space-qualified for the Improved Stratospheric

Mesospheric Sounder (ISAMS) instrument. The 50-80 K Astrium cooler was derived directly from this 80 K cooler, with minimal design changes.[2] This technology was later carried over for use in the 20-50 K coolers with an enlarged compressor piston and two-stage displacer design being introduced, as described by Scull.[3] The same design heritage for spring suspension, linear drive and clearance seals from these coolers has also been implemented, with only minor changes, in the 10 K two-stage Stirling cycle cooler.

REGENERATOR MATERIALSELECTION

The primary design challenge of the regenerator is the restricted heat capacity of materials at low cryogenic temperatures. As the temperature falls, the density of the working gas passing through the regenerator increases, and the resulting heat flow into the regenerator increases accordingly. Unfortunately, the heat capacity of most conventional regenerator materials decreases along with temperature. The regenerator material selected must be porous with a high surface area to volume ratio, allowing for efficient gas flow and an effective exchange of heat.

Initial work to improve the 20-50 K cooler was focussed largely on selection of regenerator materials to optimise the regenerator performance.[4] Improvements in regenerators have been possible with use of specific materials that undergo magnetic transitions at sufficiently low temperature, with a large magnetic contribution to the specific heat below 20 K where more traditional materials become limited.

A trade-off of materials was performed early in the project, taking into account the associated thermal and fluidic performance. The trade-off resulted in the selection of specific rare-earth materials, over more traditional materials (including lead wire, lead-plated mesh, gold wire and stainless steel mesh). With the field narrowed to a few configurations involving rare-earth materials, these options were put through a qualification process with regard to their mechanical, thermal and chemical compatibility prior to direct comparison by means of test. Manufacturability, compatibility with launch vibration and temperature cycling effects were considered in the selection process. The regenerator material configuration was finalised following a series of tests performed to compare the final two candidate materials, as tested sequentially in the same 10 K prototype cooler.

DISPLACER OPTIMISATION

The other key design challenge presented is that of low Carnot coefficient of performance (COP) at low temperature. Note that the maximum COP achievable at 10 K is 3.4%. Therefore, considerable attention was taken in design of the displacer geometry. In order to obtain optimal cold-tip temperature. A design study was performed involving sensitivity analyses for geometric parameters of diameter and length of each cold stage, as well as varying stroke and phase parameters to optimise performance.[5] In addition, this study involved detailed modeling of pressure drops through each stage of the regenerator, with validation provided by test results from a prototype 10 K cooler. High frequency pressure sensors were incorporated in the prototype model to allow for measurement of pressure at each stage of the displacer, providing comparison with modeled results.

The limitation of Carnot COP is compounded by the typical efficiency that is achievable by mechanical means. As a result, compressor power input was expected to exceed 150 W. The design must be able to manage the heat dissipation from the compressors employed. Calculated COP values for the cooler are further discussed below in relation to test data obtained recently. Note that it was not necessary within the scope of work to fully optimise the compressor system, as the objective simply to demonstrate the capability of the cooler with new regenerator material and geometry.

RESULTS OF PAST TESTING

Following prototype testing, the results of pressure drop analysis and compressor system redesign were implemented along with the modified regenerator geometry. This first prototype reached a 9.9 K base temperature in November 1998.

Following completion of the engineering model 10 K cooler, a program was carried out to determine basic sensitivity of the cooler to operation at various strokes, fill pressures (8-11 bar gauge) and frequencies (24-32 Hz). The paper by Baker (et al)[6] provided an extensive account of the test program leading up to delivery of the coolers to AFRL. The cooler performance was proven during these tests, performed in 2000 at Astrium, to reach a base temperature of 9.4 K and to lift 45 mW at 10.4 K. All of the tests to this point were performed in a clean room laboratory, using convective cooling to maintain stable interface temperatures.

COOLER SYSTEM DESIGN OVERVIEW

COOLER SYSTEM CONFIGURATION

A four-compressor system was selected to provide the cyclic variations of pressure to the single displacer mechanism. An increase in capacity was required to obtain adequate pressure swing, which was beyond the capability of a pair of standard 50-80 K compressors, used on the 20-50 K coolers. Refer to Figure 2. For reasons of cost, schedule and risk, the four-compressor approach was adopted, using the standard compressors, rather than to design a larger two-compressor.

The compressors are arranged in back-to-back pairs for vibration cancellation. A momentum balancer mechanism is fitted to the displacer, to minimise vibration at the cold-tip interface. Heat exchange takes place in a displacer unit in which the working fluid is displaced through a heat exchange matrix. The compressor piston and the displacer are each driven sinusoidally in a fixed phase relationship by linear drive motors powered by an external electronics unit.

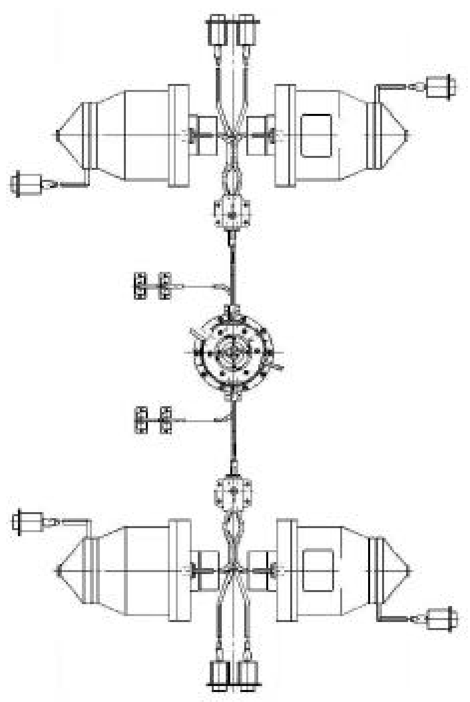

Figure 2. Schematic of 10 K Cooler System showing 4 Compressors and Displacer in Centre (Top View)

MECHANICAL AND THERMAL DESIGN

Each mechanism (compressor, displacer) incorporates a shaft mounted on diaphragm springs. The mounting system is rigid radially, allowing only translational movement along the shaft. This is necessary since both the piston in the compressor and the heat exchanger in the displacer depend on close tolerance clearance seals. The compressors are based on the design of the standard Astrium 50-80 K compressors produced for, MOPITT[7], ODIN, MIPAS, AATSR and INTEGRAL, with piston diameter scaled up by ~10%. Note that the former two have been successfully launched into orbit, with the latter three scheduled for launch within the next year.

The displacer launch support tube was removed for final tests at Astrium and AFRL as no launch vibration requirements were specified. Note that the design is readily upgradeable to a flight configuration by means of a launch support tube similar to that used on the 20-50K cooler.

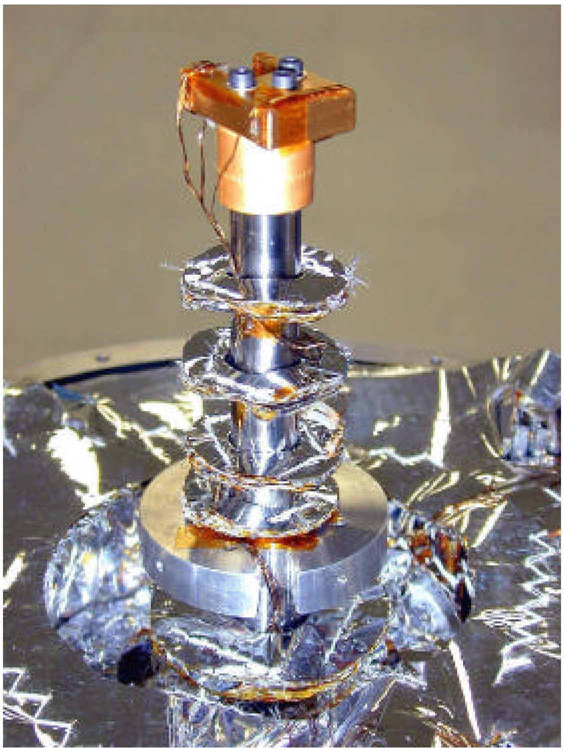

Figure 3. 10 K Cooler Cold-Tip with Temperature Monitor

The compressor and displacer are mounted in an aluminium frame for effective heat rejection by means of conduction.

THERMAL CHARACTERISATION TESTS

Following delivery of the coolers to AFRL initial tests were performed to confirm acceptability of the coolers after transit. Following these functional tests, the cooler was tested in thermal vacuum after a considerable number of cryo-pumping cycles to remove cold-tip contamination. In total, over 4000 hours have been logged with the cooler system since delivery.

A series of thermal vacuum tests were performed with the cooler running inside a 36" vacuum chamber with cables and chiller lines passed in using feedthroughs. The cooler base plate was firmly situated on two 9" round copper cold-plates with support struts. The chiller line ran through both the plates on continuous basis to maintain a steady interface temperature. The interface temperature was calculated as the average temperature of the two compressor support flanges.

HEAT LIFT PERFORMANCE VS. TEMPERATURE

A series of tests were performed at AFRL to assess the thermal sensitivity of the cooler. Interface temperatures were varied over the range from 275 K to 295 K, while maintaining constant compressor and displacer stroke, constant drive frequency and phase. The four compressors were operated near to the maximum rated amplitude (~96% of maximum swept volume). The results of the testing were entirely consistent with past measurements performed at Astrium, indicating that the performance was repeatable.

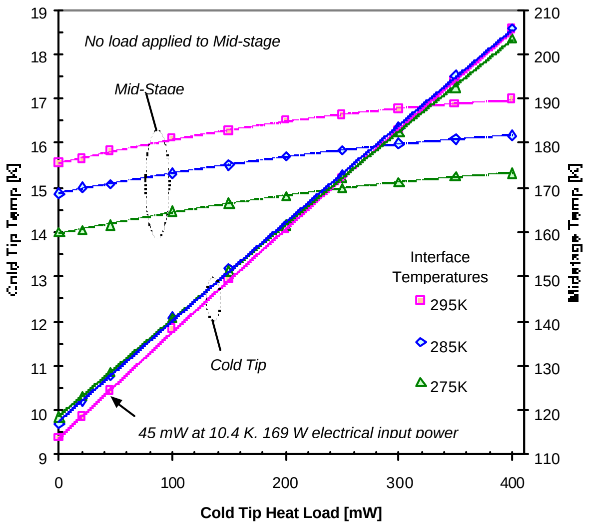

Figure 4. Cold-Tip Temperature vs. Heat Load at Various Interface Temperatures

In what would initially seem to be a counter-intuitive trend, the lowest cold-tip temperature was achieved at the highest interface temperature. However, noting that the test was performed at constant stroke, the increase in cold-tip temperature for a drop of 20 K at the interface is accompanied by a significant decrease in input power (~27% reduction at 275 K). This clearly demonstrates that the cooler is limited in stroke at lower interface temperatures.

The Carnot COP has been calculated including the electrical I^2R losses included in the input power term, as this is likely to be the figure of merit of interest to most users. It is noted that the COP, in the range of 10 K, is considerably higher at the higher interface temperature, indicating that there is actually room to increase the interface temperature further and reach a lower base temperature with higher input power. Also note that AFRL have yet to perform optimisation tests that could yield further minor performance improvements.

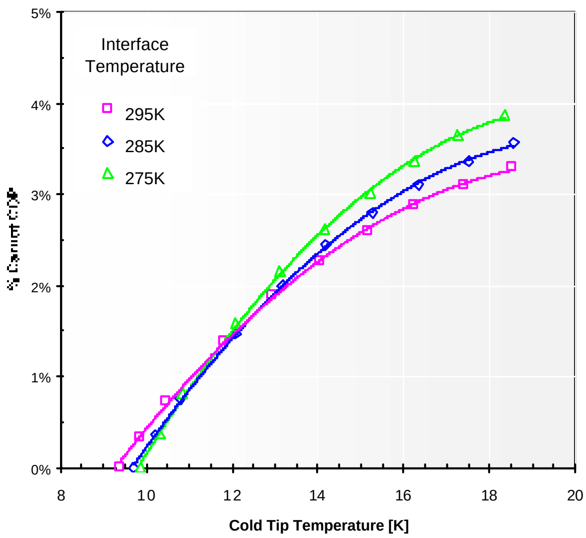

Figure 5. Coefficient of Performance at Constant Stroke (based on heat lift divided by electrical input power)

COMPARISON WITH MODELED RESULTS

The modeling of the 10 K cooler was based on RAL's thermodynamic model of the two-stage cooler, which had been developed in detail elsewhere[8]. A comparison of the modeled and measured data is shown below.

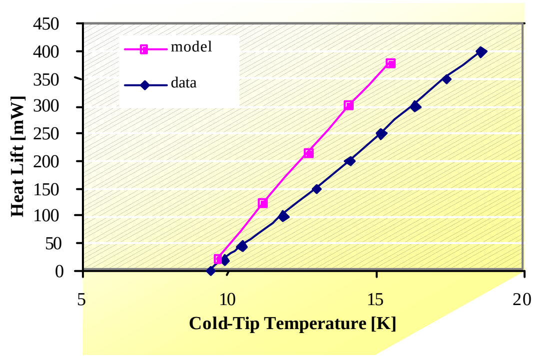

Figure 6. Comparison of Results with Model

It is noted that the agreement between the predicted and measured data is reasonable at the lowest temperatures, but there is an increasing overestimate in the modeled data as the temperature rises. Several factors conspire to cause this discrepancy:

- The model is known to consistently underestimate the pressure drop through the regenerator. This is being addressed with the use of empirical data but it is clearly an area in which the performance of the cooler could be improved.
- The model predicted a lower mid-stage temperature than achieved. This difference is greatest for the higher heat lifts and would clearly have an impact on all the calculated losses.
- As the model first computes the gross cooling power, adjusts this for the predicted pressure drop, then determines the losses, relatively small errors in any of these calculations can cause a much larger error in the net cooling predicted.

The model and data imply that the pressure drops inside the machine limit the heat lift of the cooler. The fact that the best performances have been achieved at the highest heat rejection temperature is an indication that the pressure swing in the cooler needs to be increased, to improve the base temperature and increase heat lift accordingly. In fact, the increase observed in heat lift and reduction in base temperature gained by using four compressors are critical, but improvements to the design that reduce pressure drop could give similar gains without the necessity for high input powers. The lower heat lift at reduced interface temperature may also be related to lower mean fill pressure, implying that it may be diligent to optimise fill pressure according to the anticipated temperature conditions.

The base temperature is still limited by the regenerator material. Further optimisation of the regenerator structure could yield significant heat lift improvements, while there are many potential regenerator materials that should improve the base temperature. This may be achieved by implementing a regenerator mesh, which is less dense.

Agreement with mid-stage temperature prediction may be increased by improving heat rejection at the displacer, by improving the thermal path by means of material changes or by addition of heat straps.

CONCLUSIONS

Testing at AFRL has confirmed the capability of the cooler to provide 45 mW cooling at 10.4 K while operating in a vacuum environment, consistent with past test results performed in the laboratory at Astrium. This performance result has come extremely close to the target cold-tip temperature of 10.3 K for this heat load.

Characterisation of the 10 K cooler over a range of interface temperatures indicated that the cold-tip is capable of further improvement in performance, given a greater pressure swing. The coefficient of performance calculated near the base temperature has been shown to be higher at higher interface temperatures. This is consistent with proposals to implement a compressor system with increased capacity to further improve performance.

The comparison of measured and modeled data also showed room for improvement, with results deviating from the model at higher cold-tip temperatures. As the model underestimated pressure drops, this indicates that the regenerator design can be improved. The model did, however, predict performance quite accurately in the range of 10 K.

In summary, several improvements including compressor capacity, regenerator modifications and displacer heat rejection properties appear to be feasible methods to extend the success of the design to reach lower temperatures and increased heat lift, with emphasis on improved machine efficiency.

ACKNOWLEDGMENTS

The authors would like to recognise the contributions of all manufacturing, test and engineering support personnel from Astrium, RAL and AFRL who contributed to this effort.

REFERENCES

1,5. A.H. Orlowska, T.W. Bradshaw, S. Scull, "Progress Towards the Development of a 10 K Closed Cycle Cooler for Space Use", Cryocoolers 10, R.G. Ross, Jr. – Editor, Kluwer Academic/Plenum Publishers, New York, 1999.

2. Jones, B.G., "Development for Space Use of BAe's Improved Single-Stage Stirling Cycle Cooler for Applications in the Range 50-80 K", Cryocoolers 8, R.G. Ross, Jr. – Editor, Plenum Press, New York, 1995.

3. Scull, S.R., "Design and Development of a 20 K Stirling Cooler for FIRST", Cryocoolers 9, R.G. Ross, Jr. – Editor, Plenum Press, New York, 1995.

4. A.H. Orlowska, T.W. Bradshaw, C. Jewell, B.G. Jones, S. Scull, "Improvements to the Cooling Power of a Space Qualified Two-Stage Stirling Cycle Cooler", Cryocoolers 9, R.G. Ross, Jr. - Editor, Plenum Press, New York, 1997.

6. G.S. Mand, J.R. Drummond, D. Henry, J.Hackett, "MOPITT On-Orbit Stirling Cycle Cooler Performance", Proceedings of the 11th International Cryocooler Conference, Keystone, Colorado, June 2000.

7. G. Baker, D. Féger, A. Little, A.H. Orlowska, T.W. Bradshaw, M. Crook, B.J. Tomlinson, A. Sargeant, "The Development of a 10 K Closed Cycle Stirling Cooler for Space Use", Proceedings of the 11th International Cryocooler Conference, Keystone, Colorado, June 2000.

8. T W Bradshaw, A H Orlowska, J Hieatt, "Computer Modeling of Stirling Cycle Coolers", Proceedings of the 7th International Cryocooler Conference, 17-19th November 1992, Santa Fe, New Mexico, US, PL-CP--93-1001, p621.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

AATSR	Advanced Along Track Scanning Radiometer
AFRL	Air Force Research Laboratory
BMDO	Ballistic Missile Defense Organization
DoD	Department of Defense
RAL	Rutherford Appleton Laboratory
MIPAS	Michelson Interferometer for Passive Atmospheric Sounding
MOPITT	Measurements of Pollution in the Troposphere – Canadian IR Instrument on NASA Terra Platform
ODIN	Swedish IR instrument
CDE	Cooler Drive Electronics
COP	Coefficient of Performance
VLWIR	Very Long Wave Infrared