Performance Testing of a 2K Joule-Thomson Closed-Cycle Cryocooler

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ABSTRACT

The Rutherford Appleton Laboratory (RAL) have developed a 2 K Joule-Thomson cooler for future space missions requiring low temperatures, such as the ESA's Athena X-ray telescope. The design, modelling and heat exchanger testing of the cooler was presented in *Cryocoolers 19*. In this follow-on paper, we describe the assembly of the compressors and ancillary panel, and we present the results of thermal and mechanical testing of this Demonstration Model cooler.

The DM cooler demonstrated 20 mW of cooling at 2 K, with a pre-cooler temperature of 12 K, and 14 mW of cooling at 2 K, with a pre-cooler temperature of 15 K. The total input power to the compressors to achieve these performances was 90 W. The compressors and ancillary panel successfully passed mechanical testing comprising a 25-g high sine test and an 11.7-gRMS random test in all axes. These units also completed thermal cycling between -20°C and +50°C (operating) and -35°C and +70°C (non-operating).

The lessons learnt from this demonstration development are summarised and we discuss how they are being applied to the Engineering Model 2K cooler that is currently under development at RAL as part of an ESA Core Technology Programme.

INTRODUCTION

The need for detectors with enhanced performance for cutting-edge space-borne science missions has led to the development of a new generation of photon detectors, such as Transition Edge Detectors and Superconducting Tunnel Junctions, which operate at temperatures less than 0.5 K. This in turn has driven the requirement for the provision of sub-Kelvin cryogenics for space applications, which has been achieved by so called cryochains which employ cooling technologies of ever decreasing temperature.

An example of such a cryochain was that used for the Planck spacecraft[1] (2009-2013), the operation of which is widely considered one of the most important technological achievements of the mission. Starting from 300 K, a passive cooling chain comprising a series of v-groove radiators was able to reach a temperature of 54 K. Continuing below that, a three stage active cooling chain, consisting of a closed cycle H_2 Sorption cooler (18 K), a closed cycle mechanical ^{4}He Joule-Thomson cooler (4 K) and finally an open cycle ^{3}He/^{4}He dilution refrigerator provided 0.1 K for the bolometric detectors of the High Frequency Instrument.

Instruments for future space science missions, such as the X-IFU instrument for Athena[2], will employ ever more complex cryochains[3] and will provide even lower temperatures. Joule-Thomson coolers can play an important part of these cryochains, forming a link between the sub-Kelvin coolers and 10-15 K

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class pre-coolers, such as Stirling or Pulse tube coolers. Their small cold tips and the provision of cooling which can be remote from the pre-cooling stages makes them ideally suited to the architectures employed in the spacecraft and the instruments of such missions.

The ^{4}He 4K-JT cooler used in the Planck cryochain was provided by RAL. The development of the 2K-JT cooler presented here follows on from the 4 K Planck cooler and builds on its successes as well as drawing on the substantial heritage of long-life closed cycle mechanical cryocooler developments at RAL. In a previous paper[4] we presented the design, analysis and testing of the heat exchangers in an open loop configuration with ^{4}He. In this paper we give an overview of the demonstration model (DM) cooler, with updated design and analysis, and we present the results of cryogenic tests in closed loop configuration with ^{3}He as the working fluid, and the results of mechanical and thermal environment tests on the compressor and ancillary panel assembly.

COOLER OVERVIEW

The DM 2K-JT cooler is shown in Figure 1. Seen on the left in the figure, the cooler utilises four reciprocating linear motor compression stages, arranged as two head-to-head pairs, to provide the required pressure ratio and mass flow for the Joule-Thomson expansion across the restriction, in this case an orifice. An arrangement of reed valves and buffer volumes inside the compressor centerplates is used to rectify the oscillating pressure swing to produce steady flow and pressure conditions. The compressors have a capacitive position sensor that allows them to be driven in a closed loop control mode; for this DM cooler a moving magnet motor architecture was used, and these were driven by a set of standard laboratory drive electronics, not shown in Figure 1.

An ancillary panel is provided for gas handling, housekeeping and health monitoring functions as well as to ensure the cleanliness of the working fluid, for which the avoidance of contaminant clogging at the orifice is critical to the long-term operation of the cooler. To simplify the test configuration for the DM cooler, the panel components were laid out together with the compressor on a common baseplate. Sensors are provided to monitor the pressure of the flow and return lines and a mass-flow meter, based on measuring the differential pressure across an orifice, is also provided. An ambient getter is included to protect against the possibility of residual impurities in the fill gas or the evolution of internally outgassed contaminants over time. Similarly, sintered filters are included to protect against the possible migration of particulate debris inside the cooler. For the DM cooler standard laboratory equipment was used to power and read out the ancillary panel sensors.

The working fluid must be pre-cooled below its inversion temperature prior to the JT expansion. Typically, this might be from a two stage Stirling or Pulse tube cooler, so two pre-cooling thermal interfaces (CHX-IF1 and CHX-IF2) are provided. To reduce the thermal load on these, there are tube-in-tube

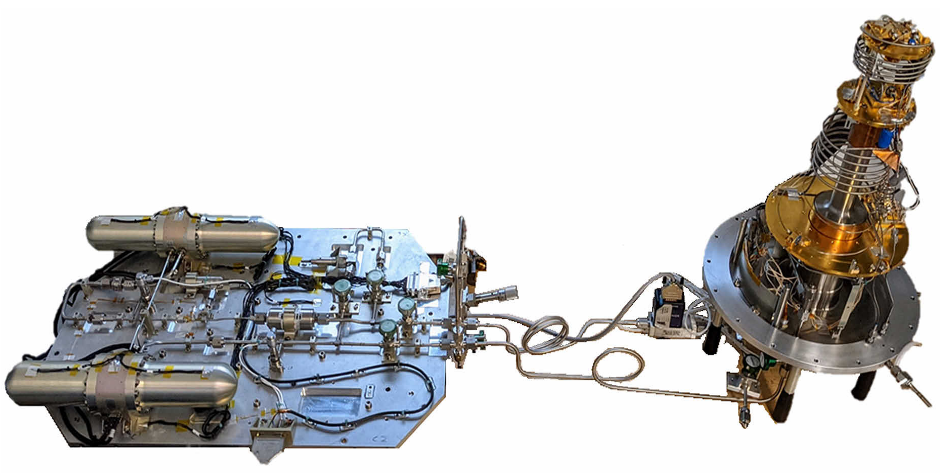

Figure 1. The Demonstration Model 2K-JT Cooler

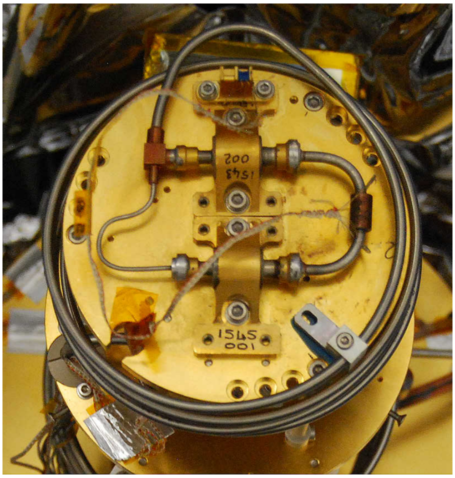

Figure 2. At the JT interface, shown here mounted on a circular test plate, the orifice, inside the copper ferule on the right, is thermally isolated from the cold tip and is equipped with a small heater to allow the possibility of raising its temperature independently of the cold tip should it become clogged. The smaller diameter high pressure flow line breaks out from the larger diameter return line before passing through a filter (bottom) A sintered reservoir (top) is provided after the orifice to retain any liquid produced during the expansion; the cooler is operated sub-critical, with the temperature being governed by the pressure in the return line above the bath.

counter flow heat exchangers between each stage and also between the first stage and ambient and between the second stage and the cold tip (CHX-CT, the JT interface). Filters installed at each thermal interface serve the dual purpose of protecting against impurities and providing good thermal contact between the fluid and the interface. The heat exchangers can be seen on the right in Figure 1 coiled up around a commercial GM-cryocooler, which was used to provide the precooling for the DM 2K-JT cooler cryogenic test campaign. A closer view of the cold tip is given in Figure 2. Calibrated Cernox sensors are provided to monitor the temperature of each interface, the cryoharness follows the routing of the heat exchanger pipework, again, for the DM cooler, these sensors were powered and read out using standard laboratory equipment.

In terms of the accommodation and integration into the instruments and spacecraft typical of space science missions such as Planck and Athena, the extended architecture of the JT cooler can be very useful. The 'warm units', comprising the compressors, ancillary panel and drive electronics, may be located some distance from the 'cold units' (heat exchangers), which minimises the possibility that mechanical and electrical disturbances from the warm units might be manifest at sensitive instrument detectors. Similarly the heat exchangers are typically around 6-8 m in total length and are easily manipulated, so can either be in a compact configuration, as shown in Figure 1, or can be routed through an extended cryogenic payload to provide remote cooling at the cold tip, with manipulation being possible after testing at unit level.

In addition the warm units may be delivered separately for integration into the spacecraft service module, whilst the cold units may be delivered independently for integration into the cryogenic part of the payload or instrument. To facilitate this, connecting lines, seen in the center of Figure 1, are provided. These may be any reasonable length, in the case of the 2K-JT cooler pressure drop in the return line is a limiting factor, but that is not always the case in general. Isolation valves (the green handles seen in Figure 1) are provided on a disconnection box at the ambient interface of the heat exchangers and also as part of the ancillary panel, so that the separate units may be delivered with their final gas fill. There are also valves provided to allow the two connecting lines to be pumped, purged and filled on the spacecraft after final installation, these can be seen located on the ancillary panel for this DM, but may also be located at the heat exchanger disconnection box.

DESIGN CONSIDERATIONS AND TRADE-OFFS

The choice of working fluid is determined by the temperature; ^{4}He is not suitable due to the superfluid transition at 2.17 K and the vapour pressure at that temperature (48 mbar) is too low to be practically accessible by reciprocating compressors that also need to generate a mass flow and a high pressure of around 8 bar. ^{3}He, on the other hand, allows the cooler to reach 2 K with a pressure of 200 mbar and 1.7 K with 110 mbar, which are accessible to reciprocating compressors, and is therefore the natural choice for the working fluid.

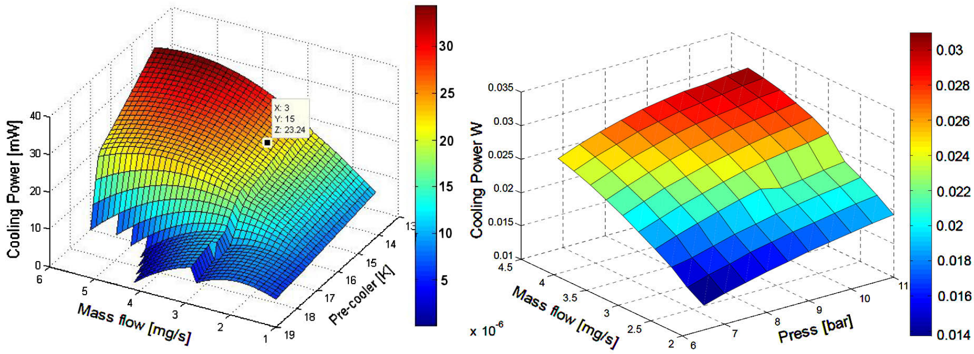

Figure 3. Cooling power as a function of mass flow, pre-cooling temperature and high pressure

Heat exchanger modeling and the selected geometry was discussed previously.[4] The cooling power as a function of mass flow and the pre-cooling temperature and high pressure prior to the expansion are given in Figure 3. It can be seen that the cooling power increases rapidly with lower pre-cooling and also increases with increasing mass flow and increasing high pressure but that the returns are diminishing as the pressure increases and becomes counter productive after approximately 10 bar.

Increasing the mass flow also has wider system impacts; the heat rejected at the pre-cooling interfaces increases in proportion to mass flow, and so to avoid overwhelming the capacity of 15 K class pre-coolers, the load rejected is limited to around 150mW at 15 K. Similarly, increasing the mass flow requires an increase in compressor capacity, in terms of size and mass as well as input power, in order to be able to maintain the low pressure and the pressure differential. In addition, the pressure drop in the return line is also increased, leading to further increases in compressor capacity. Furthermore, attempts to increase the heat exchanger efficiency to improve this situation inevitably lead to longer pipe lengths that also increase the return line pressure drop.

The compressor modelling, along with the configuration and architecture, was also discussed previously.[4] To achieve the pressures discussed above, four stages of compression are required. A full trade-off with the considerations outlined above was carried out for the DM 2K-JT cooler and has resulted in the selection of the operating conditions and parameters given in Table 1.

TEST CONFIGURATIONS

Results are presented below for several test configurations. The compressor assembly was tested, together with the ancillary panel, in a standalone configuration to evaluate the performance prior to integra-

Table 1. DM 2K-JT cooler nominal operating conditions and configuration.

High pressure (Po)	8 bar
Low pressure (Pi)	0.15 bar (0.2bar at the cold tip)
Mass flow	3 mg/s
Pre-cooling temperatures	CHX IF1 = 100 K, CHX-IF2 = 15 K
operating frequency	45-55 Hz
Fill pressure	2.3 bar
Maximum compressor stroke	13 mm pk-pk
Compressor piston diameters	S1=49 mm, S2 = 25.2 mm, S3 = 14.7 mm, S4 = 10.3 mm
Compressor mass and size	7.76kg, 401 mm x 124 mm x 96 mm (each pair)

CHX	Tube	Gauge	ID [mm]	OD [mm]	Wall [mm]	Length [m]
1	HP	17	1.10	1.47	0.185	1
	LP	8	3.28	4.04	0.380	1
2	HP	17	1.10	1.47	0.185	2.5
	LP	8	3.28	4.04	0.380	2.5
3	HP	17	1.10	1.47	0.185	2
	LP	10	2.50	3.30	0.400	2

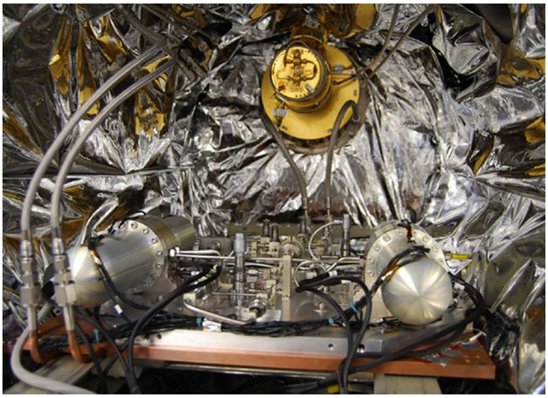

Figure 4. DM 2K-JT cooler installed in the vacuum tank for cryogenic performance tests. The heat exchanger radiation shields are removed in the picture.

tion with the heat exchangers. In this configuration a needle valve is used to adjust the mass flow under ambient conditions to be representative of the mass flow through the orifice under cryogenic conditions. Unlike a fixed orifice, the needle valve allows a variety of mass flows to be obtained for a range of compressor amplitudes.

The needle valve, which can be isolated using the fill valves, was left in-situ on the ancillary panel for the duration of the test campaign. For mechanical environment tests, which did not include the heat exchanger assembly, the needle valve was used for functional checks between each of the individual test cases.

For cryogenic performance tests the complete cooler assembly was installed into a vacuum chamber, as shown in Figure 4 with the compressors and ancillary panel mounted to a temperature controlled base-plate. The heat exchangers were coiled around a commercial GM-cryocooler as described above, with radiation shields providing a thermal environment for the interfaces at the temperature of each of the pre-cooling stages. Thermal shunts, described previously[4], were used to measure the heat rejected to each of the two pre-cooling stages.

PERFORMANCE TESTS

Standalone Compressor Tests

Figure 5 shows some results of the needle valve loop tests at the nominal fill pressure of 2.3 bar. At the start of the plot, the needle valve has been adjusted to give a near nominal mass flow with full compressor stroke at 51 Hz; a high pressure of 8.78 bar and a low pressure of 0.07 bar was achieved with a mass

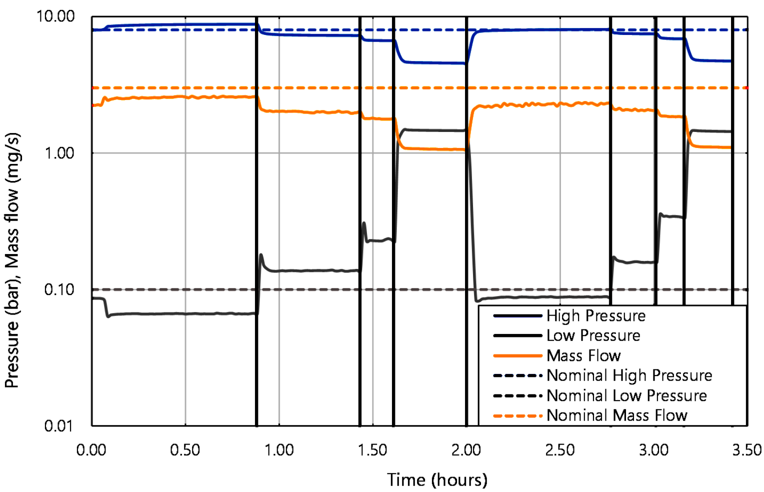

Figure 5. Standalone compressor needle valve loop test results with ^3He

Table 2. Cooler Cooling Power as a function of pre-cooler temperature and compressor stroke

Description	Full stroke				90% Stroke		80% Stroke	
High pressure (bara)	6.96	6.99	**7.23**	7.23	6.69	6.91	6.34	6.48
Low pressure (bara)	0.106	0.104	**0.105**	0.107	0.119	0.118	0.132	0.133
He-3 Mass Flow (mg/s)	2.87	2.85	**2.96**	2.98	2.77	2.85	2.61	2.67
CHX-IF1 temperature (K)	111.45	111.40	**112.02**	112.03	111.38	111.74	111.22	111.68
CHX-IF2 temperature (K)	9.96	11.99	**15.13**	18.13	12.05	15.02	12.04	15.07
CHX-CT temperature (K)	2.66	2.25	**2.20**	2.02	2.32	2.21	2.31	2.20
Heat rejected at CHX-IF1 (mW)	64.8	65.6	**79.1**	79.0	64.4	73.1	60.7	71.7
Heat rejected at CHX-IF2 (mW)	73.8	62.2	**57.1**	51.7	60.0	53.4	55.7	50.8
Cooling power (mW)	26.4	20.2	**13.8**	8.0	19.0	13.0	17.4	11.7
CPA stroke amplitude (mm)	5.77	5.78	**5.77**	5.78	5.17	5.18	4.61	4.59
CPA Power Consumption (W)	86	88	**90**	91	75	77	71	66

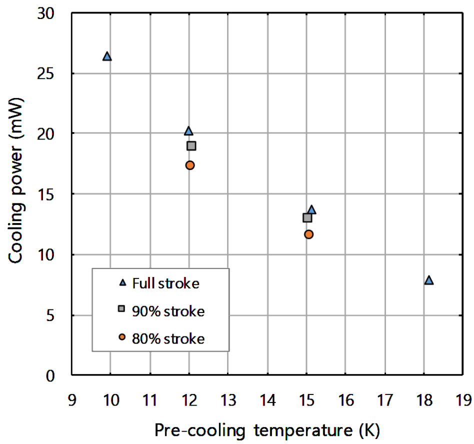

Figure 6. DM 2K-JT cooler cooling power as a function of 2nd stage pre-cooler temperature

flow of 2.6 mg/s. The plot shows several subsequent reductions in compressor stroke and is repeated for 48 Hz (after 2hours) without further valve adjustment.

Steady State Cryogenic Performance Tests

Full system tests were carried out in the vacuum chamber. Measurements were made of the steady state cooling power at several different pre-cooler (CHX-IF2) temperatures for a range of compressor stroke amplitudes at 51 Hz.

Steady state cooling power measurements are summarised in Table 2 with the nominal operating point highlighted in bold. A range of strokes are shown for a range of CHX-IF2 pre-cooling temperatures. As can be seen in the table, and in Figure 6, cooling power at the JT cold end is a strong function of the pre cooler temperature. At the nominal operating point with 15 K pre-cooling the DM 2K-JT cooler achieved a cooling power of just under 14 mW, rising to 20 mW with 12 K pre-cooling. During these steady state tests the temperature stability of the cold end was measured as being less than 10 mK/hour with no active regulation and a temperature stability of 0.16 K/hour on the pre-cooling interface.

Transient Load Cryogenic Performance Tests

Measurements were also made with sudden changes in the thermal load applied to the 2 K stage (CHX-CT) to simulate recycling of a sub-Kelvin cooler such as a sorption cooler or an ADR. The results are given in Figure 7. With the cooler at steady state without an applied heat load, the load was switched

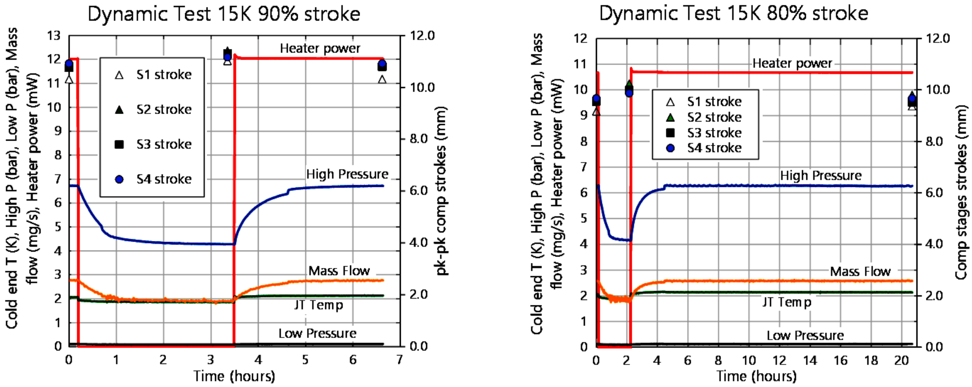

Figure 7. DM 2K-JT Cooler transient load performance tests at 15 K pre-cooler temperature

to the maximum cooling power recorded at that operating point. The behaviour of the cooler was observed, in particular the cold end temperature stability and the operation of the compressors under the sudden evolution of gas from the liquid boil-off. Data were collected at 90% and 80% of maximum stroke, and showed that the cooler operation was stable in each case.

ENVIRONMENTAL TESTS

Thermal Cycling

Thermal cycling was carried out in the vacuum environment chamber with the compressor and ancillary panel in standalone configuration. For this test the cooler was filled with ^{4}He. The temperature of the compressor and ancillary baseplate (thermal interface) was controlled and an MLI shroud was placed over the compressor and ancillary panel assembly. Three cycles were carried out including hot and cold start/stops, to the requirements given in Figure 8, which also shows the data taken during the tests. No change in performance during or after the tests was noted, and no ^{4}He leaks into the vacuum chamber were detected during the test.

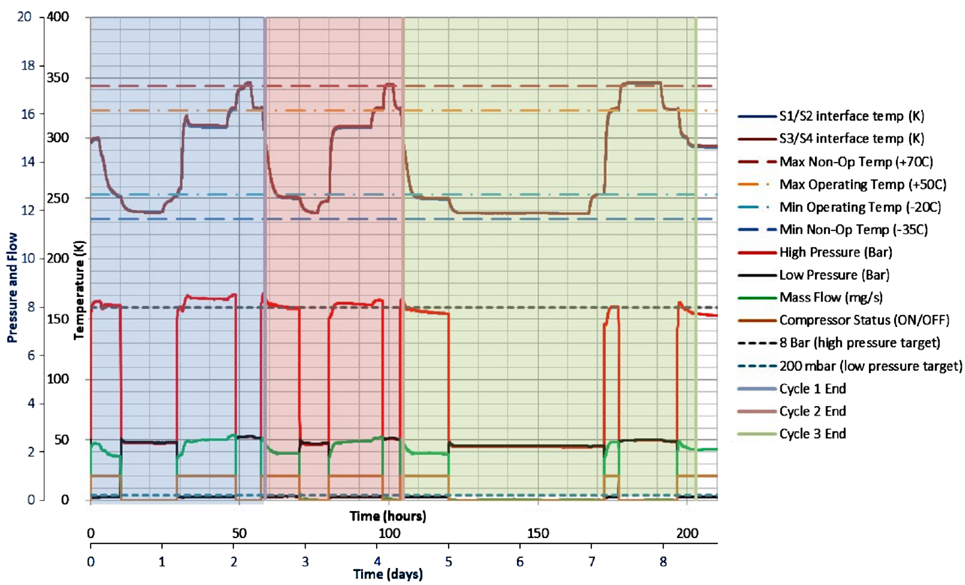

Figure 8. DM 2K-JT cooler thermal cycling

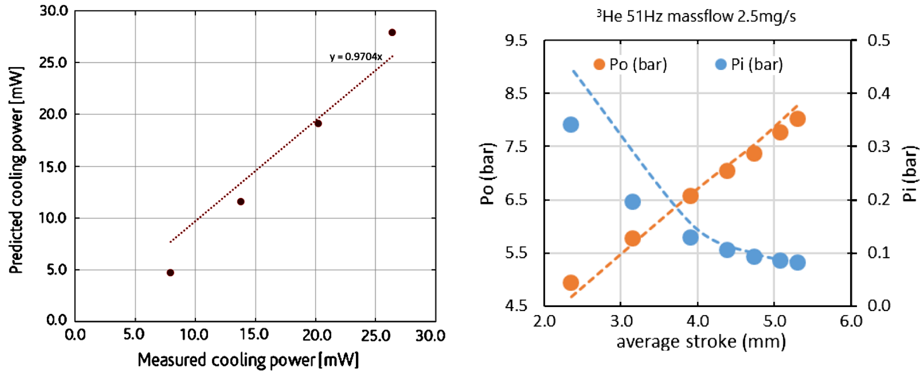

Figure 9. DM 2K-JT test setup, accelerometer locations and loads for mechanical environment tests

Mechanical Environment

Mechanical environment tests were carried out on the compressor and ancillary panel in standalone configuration. The baseplate was mounted directly to the RALSpace vibration facility shaker table. No environmental loads or representative configuration is available for the heat exchanger assembly at the present time, so it was not included in these tests.

The test setup is shown in Figure 9, together with the applied loads, which were repeated in all three axes. During each load case, the compressor coils were shorted, simulating a passive launch lock. Between all load cases, low level resonance scans, compressor stiction traces and functional run-up to nominal operation were carried out to monitor structural integrity and compressor performance. In all cases no significant changes to those tests were noted.

During the high sine sweep in the axis of the compressors, position voltage and current waveforms for each compression stage were captured. In all cases, the measured displacement was less than the maximum amplitude. The motion of the pistons results in a pressure differential across the needle valve which is similar to that in nominal operation, and this causes the pistons to pump forwards as the backshell pressure is slowly equalised through the piston clearance seal. In all cases there was no contact with the end-stops and the current and voltage maintained safe levels. A peak of 51W was dissipated during the high sine test, with nearly all of that in the two high pressure stages S3 and S4 (~20W each).

MODEL CORRELATION AND UPDATES

The models, described in the previous paper[4], that had been used to design the cooler did not initially give good agreement with the results described above. In particular the heat exchanger model, which had given good agreement with the ^{4}He open loop bottle tests presented in that paper, was poorly correlated when using ^{3}He. Several data sources[5,6,8] are available for ^{3}He, and are used in the model. Inspection of these show that the thermo-physical properties are rapidly changing below 10 K and could therefore add large errors into the model; in addition, these properties must be carefully extrapolated from the existing experimental data. Notably, large changes in specific heat capacity of the high pressure gas are strongly linked to the heat exchanger performance, and hence the cooling power achieved.

With improvements to address these issues and an increase in the model resolution at low temperatures, the agreement with the results has been significantly improved. Small improvements, mainly focusing on the reed valve dynamics, were also made to the compressor model. The updated model predictions against the experimental results are given in Figure 10, for which the agreements are significantly improved. The agreement with ^{4}He for the updated model is also still very good.

CONCLUSIONS AND NEXT STEPS – AN EM COOLER

A demonstration model 2K-JT cooler has been developed at STFC's Rutherford Appleton Laboratory. The cooler has demonstrated continuous operation at temperatures around 2 K with various applied heat loads and with pre-cooling between 10 K and 18 K. The cooling power was measured as being just under 14 mW with pre-cooling at 15 K and in excess of 26 mW at 10 K.

The compressors have demonstrated an ability to achieve pressures as low as 70 mbar with a mass-flow of 2.6 mg/s, which in principle allows temperatures below 2 K to be achieved. Operating in closed loop, the compressors have also demonstrated an ability to maintain stable operation of the cooler under worst-case conditions simulating the sudden application of a thermal load as for recycling of a sub-Kelvin cooler.

The compressors and ancillary panel have survived thermal and mechanical environmental tests without any change in performance, and a passive launch lock simulation—by shorting of the compressor coils—has been successfully demonstrated without any compressor parameters exceeding normal limits.

The models used to design the cooler have been updated and give reasonable correlation with the experimental results. As part of a follow-on contract with ESA under the Core Technology Programme, the updated models have been used to design an engineering model cooler with greater capacity, full details of which will be reported separately.

The engineering model cooler heat exchangers have been optimised to provide 20 mW of cooling with 15 K pre-cooling and the compressor capacity has been increased to produce a nominal high pressure of 10 bar at a mass flow of 4.5 mg/s. To achieve this the compressor stroke has been increased from 13 mm pk-pk to15 mm pk-pk and the largest piston diameter has been increased from 49 mm to 57 mm. Alterations to the heat exchangers have been made to reduce the pressure drop in the lines and the cold tip architecture has been modified to reduce the thermal resistance between the fluid reservoir and the interface.

The EM cooler utilises a moving coil motor architecture for which the geometry has been tailored for this application. Sensors for active vibration control are included as part of the development, as is a dedicated drive electronic unit.

ACKNOWLEDGMENTS

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