Characterization testing of Lockheed Martin Micro1-2 cryocoolers for the Mapping Imaging Spectrometer for Europa (MISE) instrument

I M McKinley, M A Mok, C D Hummel, D L Johnson and J I Rodriguez
Jet Propulsion Laboratory, California Institute of Technology, Pasadena, CA 91109
USA

Email: ian.m.mckinley@jpl.nasa.gov

Abstract. The Mapping Imaging Spectrometer for Europa (MISE) instrument on the Europa Clipper mission will use a Lockheed Martin “high-power” Micro1-2 pulse tube cryocooler with a heat rejection temperature below 250 K. This paper describes the performance testing and results of Lockheed Martin Micro1-2 coolers optimized for these conditions. The heat reject temperature varied between 170 K and 260 K for different helium fill pressures. The coolers were driven with input powers ranging from 5 W to 40 W and drive frequency between 125 Hz and 150 Hz. For all conditions measured, the heat flow from the compressor was between 54% and 58% of the total heat and the compressor temperature was between 4 K and 6 K warmer than the expander temperature. In addition, another Micro1-2 cooler optimized for 300 K environment was subjected to a life test at reject temperatures below 250 K. The cooler performance and helium leak rate did not change over the duration of the life test. Moreover, a burst test was performed on a unit of this model of cooler that was without the internal components. Finally, a method for developing analytical expressions to curve fit cooler performance data is presented. These expressions are provided and predict cooler performance to within 15% for a fixed drive frequency over a range of cold tip temperature, heat reject temperature, and input power.

1. Introduction
The Jet Propulsion Laboratory (JPL) has chosen the Lockheed Martin Micro1-2 cryocooler to provide active cooling on the Mapping Imaging Spectrometer for Europa (MISE) instrument on NASA’s Europa Clipper spacecraft. The Micro1-2 coaxial pulse tube microcryocooler can be driven with up to 60 W and weighs 450 grams including the compressor pedestal mount and is slightly larger than the 350 gram, 25 W standard version (Micro1-1) that has been thoroughly characterized previously [1, 2, 3, 4, 5]. These coolers can be optimized for various heat rejection environments, cold tip temperatures, and drive frequencies [6, 7]. The performance of various Micro1-2 coolers was previously reported [6, 7, 8, 9, 10, 11]. The Micro1-2 cooler was qualified to Technology Readiness Level (TRL) of six for Earth orbiting missions in 2016 [8] as well as for the harsher Europa environment in 2017 [11]. The MISE instrument intends to limit power consumption of the cooler by taking advantage of its functionality at heat rejection temperatures below 250 K. This led to the development of MISE prototype coolers optimized for 220 K heat rejection temperature to provide 0.75 W of cooling at 80 K while operating at 135 Hz [7]. The left photograph of Figure 1 shows one of the two Micro1-2 MISE prototype coolers as it was delivered to JPL. Previous work reported on the characterization testing performed MISE prototype units discussing performance, exported forces, off-state conductance, random vibration, and magnetics testing [7]. The helium fill pressures for the coolers in previous work [7] was 800 psi and 750 psi for the first (Proto1) and second (Proto2) prototype coolers, respectively. This follow-on work presents further performance data from these two coolers as well as results from a burst test and life test performed on different coolers.
2. Thermal vacuum tests on MISE prototype coolers

2.1. Test setup and procedure

Figure 1 (right photograph) shows a MISE prototype cooler in the TVAC chamber at JPL. This test setup was identical to that previously used in Ref. [7] and similar to that described in Ref. [5]. The cooler was mounted to an aluminum plate that was connected to a CTI 350 coldhead by means of a copper bar. All of the cold surfaces including the microcooler cold finger were wrapped in multiple layers of aluminized mylar (MLI) for insulation. The cold tip temperature was measured by a Lakeshore DT-670 diode and controlled by a Lakeshore 340 temperature controller powering a resistive element. Both the heater and sensor were attached to a copper block that was clamped to the cold tip and made use of four-wire measurements. The microcooler was powered using a Chroma 61602 AC source supplying between 5 W and 40 W at frequencies between 125 Hz and 145 Hz. The heat rejection temperature was defined as that of the expander mounting flange of the microcooler. It varied from 170 K to 260 K while the cold tip temperature varied from 80 K to 250 K. Finally, Lockheed Martin supplied and equation describing the recommended maximum drive voltage based on the motor characteristics as a function of frequency, compressor temperature, and current. This maximum recommended voltage was not exceeded during testing.

2.2. Effect of fill pressure

Due to the possibility of the heat rejection environment in the MISE instrument increasing from 220 K to 240 K, the first prototype cooler helium fill pressure was reduced from 800 psi to 600 psi to adjust for the higher heat rejection temperature. Figure 2 shows (a) the specific power of Proto1 and (b) the motor efficiency vs. drive frequency for 15 W input power and 80 K cold tip for different expander temperatures and the two different fill pressures. The specific power was defined as the compressor input power divided by the cooling power. The motor efficiency was defined as the quantity of the losses due to joule heating ($i^2R$) in the coils subtracted from the compressor input power divided by the compressor input power [12]. In this case, the coil resistance was taken to be 5.4 ohms corresponding that at room temperature (RT) obtained with a 4-wire measurement. The compressor temperature was less than 5 K warmer than the expander temperature for all nominal performance measurements falling within the range that has been shown have no effect on performance [7]. It is evident that for a fixed expander temperature, the optimal drive frequency decreased with decreasing fill pressure. The specific power and motor efficiency respectively increase and decrease with decreasing fill pressure. Figure 3 shows the compressor input power vs. cooling load for Proto1 at 80 K cold tip for various expander temperatures and fill pressures. For a fixed drive frequency of 135 Hz and 140 Hz at expander temperature 220 K and 230 K, respectively, the performance was worse at 600 psi than at 800 psi. In addition, for the optimal drive frequency at these expander temperatures with 600 psi fill pressure, the performance was still worse than the cooler operating at its optimal drive frequency with 800 psi fill pressure.
2.3. Effect of heat rejection temperature and drive frequency

Figure 4a shows the specific power of Proto1 filled to 600 psi and Figure 4b shows the motor efficiency vs. drive frequency for 15 W input power and 80 K cold tip for different expander temperatures. It is evident that the minimum specific power and the maximum motor efficiency both depended on the heat reject temperature and that the specific power increased while motor efficiency decreased for increasing expander temperature. In addition, the optimal drive frequency at 15 W and 80 K cold tip increased with increasing expander temperature. Furthermore, as the expander temperature increased, the minimum in specific power vs. frequency became narrower while the width of the peak in motor efficiency did not change. Figure 4c shows the compressor input power vs. cooling load for Proto1 filled to 600 psi at 80 K cold tip for various heat rejection temperatures when driven at the optimal drive frequencies determined from the minimum specific power shown in Figure 4a. The cooling power for a given compressor input power decreased nearly linearly in this expander temperature range. Figure 4d-f shows the same data as Figure 4a-c except for Proto2 filled to 750 psi. The second prototype cooler exhibited the same trends as the first.

(a) $\frac{P_{in}}{Q_{out}}$ vs. Drive frequency (Hz)
(b) $\eta_{motor} = 100 \left[ \frac{P_{in} - \frac{1}{2} R_{coil} f^2 (T_{comp}/T_{RT})}{P_{in}} \right]$

Figure 2. (a) Specific power and (b) motor efficiency vs. drive frequency for Proto1 for different heat rejection temperatures and fill pressures with the compressor input power at 15 W and the cold tip at 80 K.

Figure 3. Compressor input power vs. cooling load for Proto1 at 80 K cold tip for various expander temperatures and fill pressures.
Figure 4. (a) Specific power and (b) motor efficiency vs. drive frequency for Proto1 filled to 600 psi for different heat rejection temperatures with the compressor input power at 15 W and the cold tip at 80 K. (c) Compressor input power vs. cooling load for Proto1 filled to 600 psi at 80 K cold tip for various expander temperatures with the cooler driven at its optimal frequency. (d-f) Same as (a-c) except for Proto2.
2.4. Heat flow distribution

The heat flow distribution between the compressor and expander of Proto2 was measured for various cold tip temperature, compressor input power, and expander temperature. Heat flow meters were built from stainless steel plates and installed beneath the expander and compressor. They each were instrumented with two DT-670 diodes bonded into slots on each of their sides. They were calibrated for fixed mounting plate temperature between 180 K and 240 K by adding constant heat loads between 0 W and 20 W while measuring the temperature difference across the diodes. Heat flow vs. temperature difference was linearly fit for each mounting plate temperature, then the linear coefficients were linearly fit as a function of mounting plate temperature similar to the approach used in Ref. [13].

The heat flow distribution was measured as a function of compressor input power with the cooler driven at its optimal frequency for an expander temperature ranging from 220 K to 250 K with a cold tip temperature of 80 K. In addition, it was measured as a function of cold tip temperature for an expander temperature of 240 K and an input power of 15 W at 140 Hz. The heat flow distribution was independent of cold tip temperature. Finally, the heat flow distribution was measured as a function of drive frequency for expander temperature ranging from 220 K to 250 K with a cold tip temperature of 80 K and 15 W compressor input power. For all conditions measured, (i) between 54\% and 58\% of the heat was dissipated at the compressor and (ii) the compressor temperature was between 4 K and 6 K warmer than the expander temperature.

2.5. Performance fitting

Figure 5 shows the measured performance of Proto1 filled to 600 psi and driven at 135 Hz for various expander and cold tip temperatures. The measured data was curve fit to generate a polynomial with 27 coefficients that predicts the cooling load for compressor input powers between 5 W and 30 W, expander temperatures between 170 K and 250 K, and cold tip temperatures from 80 K up to the expander temperature. This polynomial was generated by fitting cooling load $Q_{\text{cool}}$ vs. cold tip temperature $T_{\text{CT}}$ for fixed compressor input power $P_{\text{comp}}$ for each measured expander temperature $T_{\text{exp}}$. Then the coefficients of these polynomials were fit as a function of $P_{\text{comp}}$. Then the resulting coefficients were used as a function of $T_{\text{exp}}$. Combining all of these expressions yields Equation (1) for which the coefficients are defined in Table 1.

The dashed black lines in Figure 5 show the predicted cooling loads for the measured data. It is evident that Equation (1) accurately predicts the cooler performance. In fact, the maximum relative error in predicted cooling load was 13\% and occurred only for a single measured point. The error of all other predictions were below 6\%. In addition, the maximum absolute error in cooling load was less than 60 mW. Finally, Equation (2) can be solved for $P_{\text{comp}}$ in terms of $Q_{\text{cool}}$, $T_{\text{CT}}$, and $T_{\text{exp}}$ and the result used to predict the compressor input power. Together these expressions for $P_{\text{comp}}$ and $Q_{\text{cool}}$ can be used alternatingly in a transient thermal model to accurately model the cool down of an instrument such as MISE.

\[ Q_{\text{cool}}(P_{\text{comp}}, T_{\text{CT}}, T_{\text{exp}}) = AT_{\text{CT}}^2 + BT_{\text{CT}} + C \quad (1) \]

\[
A = DP_{\text{comp}}^2 + EP_{\text{comp}} + L \\
B = FP_{\text{comp}}^2 + GP_{\text{comp}} + H \\
C = IP_{\text{comp}}^2 + JP_{\text{comp}} + K \\
D = C_{01}T_{\text{exp}}^2 + C_{02}T_{\text{exp}} + C_{03} \\
E = C_{04}T_{\text{exp}}^2 + C_{05}T_{\text{exp}} + C_{06} \\
F = C_{07}T_{\text{exp}}^2 + C_{08}T_{\text{exp}} + C_{09} \\
G = C_{10}T_{\text{exp}}^2 + C_{11}T_{\text{exp}} + C_{12} \\
H = C_{13}T_{\text{exp}}^2 + C_{14}T_{\text{exp}} + C_{15} \\
I = C_{16}T_{\text{exp}}^2 + C_{17}T_{\text{exp}} + C_{18} \\
J = C_{19}T_{\text{exp}}^2 + C_{20}T_{\text{exp}} + C_{21} \\
K = C_{22}T_{\text{exp}}^2 + C_{23}T_{\text{exp}} + C_{24} \\
L = C_{25}T_{\text{exp}}^2 + C_{26}T_{\text{exp}} + C_{27} \\
C_{01} = -1.0821E-10 \\
C_{02} = -5.1745E-06 \\
C_{03} = 2.6016E-09 \\
C_{04} = -1.1439E-06 \\
C_{05} = 1.2187E-04 \\
C_{06} = 2.6664E-08 \\
C_{07} = -1.1850E-05 \\
C_{08} = 0.001287659 \\
C_{09} = 1.0821E-10 \\
C_{10} = -5.1745E-06 \\
C_{11} = 2.6016E-09 \\
C_{12} = -1.1439E-06 \\
C_{13} = 1.2187E-04 \\
C_{14} = 2.6664E-08 \\
C_{15} = -1.1850E-05 \\
C_{16} = 0.001287659 \\
C_{17} = -5.1745E-06 \\
C_{18} = 2.6016E-09 \\
C_{19} = -1.1439E-06 \\
C_{20} = 1.2187E-04 \\
C_{21} = 2.6664E-08 \\
C_{22} = -1.1850E-05 \\
C_{23} = 0.001287659 \\
C_{24} = -5.1745E-06 \\
C_{25} = 2.6016E-09 \\
C_{26} = -1.1439E-06 \\
C_{27} = 1.2187E-04 \\
\]

Table 1. Coefficients for Equation (1).
Figure 5. Measured and predicted compressor input power vs. cooling load for various cold tip temperature and heat reject temperature for Proto1 filled to 600 psi and driven at 135 Hz.

3. Life test

The same Micro1-2 unit that was used for TRL testing [8, 11] completed a life test at JPL that lasted from October 2, 2017 to February 22, 2019. This unit had previously accumulated 7,700 hours of operation at Lockheed Martin at 300 K heat rejection temperature operating with 60 W of compressor input power at 140 Hz [6]. Figure 6a shows the desired and measured expander temperature of the cooler during the JPL life test as a function of time. It also shows the measured mounting block and compressor temperatures. The cooler operated at 220 K, 185 K, 250 K, and 220 K for 10.65 hours each then it turned off and was cooled to 125 K over 9 hours. It was then heated to 140 K at which point it turned on at low power and continued to warm up to 220 K. This temperature profile repeated every 55 hours. The cold tip was controlled to 80 K and the cooling load and cooler input power were continuously measured while the cooler operated at 122 Hz. In addition, a helium leak check was performed weekly. Including the operation at Lockheed Martin, the flexure bearings in this unit accumulated at least 8.16x10^9 cycles. In addition, the cooler accumulated 240 start/stops at temperatures below 250 K and underwent greater than 210 thermal cycles down to 125 K.

Figure 6b shows the steady-state specific power over the course of the life test at the three different expander temperatures. Steady-state was declared after the temperature rate of change of both the expander and cold tip was less than 0.1 K/min for greater than 360 minutes when the cooler was on. This definition corresponded to one steady-state point per temperature dwell. Figure 6b also shows the linear fit for each expander temperature along with ±2 standard deviations of the fit. It is evident that the fits capture the trend of the data well. According to the linear fits, the specific power increased by 0.21 W/W, 0.18 W/W, and decreased by 0.18 W/W over 500 days for expander temperatures of 185 K, 220 K, and 250 K, respectively. The corresponding change was less than 1% for all expander temperatures. In addition, the helium leak rate was measured nearly 60 times and never exceeded 1.1x10^-9 mbar·L/s. Note that a leak rate of 7.6x10^-7 mbar·L/s corresponded to a 10% drop in helium fill pressure over 18 years. The helium leak rate was measured with the cooler both on and off for expander temperatures ranging from 160 K to 250 K and compressor temperature ranging from 160 K to 270 K. There was no correlation between leak rate and cryocooler temperature.
Figure 6. (a) Life test temperature profile and (b) steady-state specific power vs. time.

4. Burst test

Figure 7 shows a photograph of the test setup used to perform the hydrostatic burst test. The test was done hydrostatically to prevent the large release of energy upon burst that occurs if a compressible gas is used. The unit used for the burst test did not have any of the internal, non-pressure vessel components. A custom transfer line was fabricated and attached to the cooler. It featured a T-joint with a valve. The helium fill port on the reservoir volume was also implemented with a valve. In preparation for the test, the cooler was pressurized with helium to check for leaks. No leaks were detected. Then the cooler was connected to a vacuum pump that pulled vacuum on the cooler for more than 24 hours. Next, the cooler was backfilled with water through the transfer line valve. Then the reservoir volume valve was opened with it pointed up so that it was the highest point of the cooler and flow of water was observed. This indicated that there was no air trapped in the cooler. The reservoir volume valve was closed and the cooler was placed in a containment chamber. The water pressure in the cooler was slowly increased until the cooler burst at 1750 psi at the reservoir volume weld. The location is indicated by a black arrow in Figure 7.

Figure 7. Photograph of the hydrostatic burst test setup.
5. Conclusion

The performance of the MISE prototype cryocoolers was presented for various heat rejection temperature, cold tip temperature, helium fill pressure, and drive frequency. For all conditions measured, the heat flow from the compressor was between 54% and 58% of the total heat and the compressor temperature was between 4 K and 6 K warmer than the expander temperature. In addition, this model of cooler passed a life test based on performance and helium leak rate metrics. Moreover, a burst test was performed on a unit of this model of cooler that was without the internal components. Finally, a method for developing analytical expressions to curve fit cooler performance data was presented that was able to predict performance to within 15% for a fixed drive frequency over a range of cold tip temperature, heat reject temperature, and input power.

6. References

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