Development of Advanced Two-Stage Stirling Cryocooler for Next Space Missions

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ABSTRACT

An advanced two-stage Stirling (2ST) cryocooler has been developed for the cryogenic systems of future astronomical and Earth-observation missions in JAXA. The key near-term focus, SPICA, is a Japanese infrared space telescope with a primary mirror of 3.5 m diameter and an optical bench to be maintained at 4.5 K for over 5 years via radiant and mechanical cooling. The temperature of 4.5 K is achieved using a Joule-Thomson (JT) circuit combined with the 2ST cryocooler for precooling to 15-20 K; this precooling contributes strongly to the JT-circuit cooling performance. Therefore, the cooling performance and reliability of the 2ST cryocooler is key to the success of the SPICA mission. Reduction of vibration induced by the 2ST cryocoolers is another important technical issue to avoid deterioration of spatial resolution and pointing stability of optical devices. This paper describes the development status of the advanced 2ST cryocooler and discusses several technical approaches for achieving higher cooling performance, better reliability, and less vibration for upcoming space missions.

INTRODUCTION

Existing 20 K-class two-stage Stirling (2ST) cryocooler.

The existing 20 K-class 2ST cryocooler was developed for the first Japanese infrared astronomical satellite AKARI (ASTRO-F) launched on February 21, 2006 (UT), from Uchinoura Space Center.[1,2] The design of this 2ST cryocooler was based on a small two-stage Stirling-cycle cooler developed in 1991 for space applications with temperatures below 20 K.[3]

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Figure 1 shows a cross-sectional view of the existing 2ST cryocooler. It consists of a cold head (expander), a compressor, and a connecting tube. The linear compressor has two opposing pistons mounted on a drive shaft to cancel out mechanical vibration. The drive shaft is supported by two sets of linear ball bearings that maintain the piston clearance seal with low drive frequency and long piston stroke. The cold head consists of a double-staged displacer and an active balancer working as a counterweight for the displacer to reduce mechanical vibration. The displacer uses stainless steel screens in the regenerator and is supported by coil springs and contact seals. It is driven linearly by a coil and permanent magnets as well as by the gas flow from the compressor. The nominal cooling capacity is specified as 0.2 W at 20 K for 90 W power input as presented in Table 1.

AKARI uses the Focal Plane Instrument (FPI), the Infrared Camera (IRC), and the Far-Infrared Surveyor (FIS) to survey the entire sky over infrared wavelengths from 2-200 μm. These optical devices and primary mirror of 68.5 cm diameter are refrigerated to below 7 K by a hybrid cooling system consisting of 170 *l* (25 kg) of super-fluid liquid helium (HeII) cryogen plus two sets of 2ST cryocoolers for redundancy. The cold stages of the 2ST cryocoolers are connected thermally to the IVCS (Inner Vapor Cooled Shield). The parasitic heat load to the helium tank through the thermal shields and structural supports is reduced to suppress the consumption rate of cryogen to achieve an observation life of 1.5 years. During nominal operation, each 2ST cryocooler is driven with 50 W of input power to achieve 0.1 W of cooling at 20 K. If one of the two 2ST cryocoolers fails, the other one is driven with a maximum input power of 90W to achieve a cooling capacity of 0.2 W at 20 K to compensate for the lost cooling capacity.[4,5]

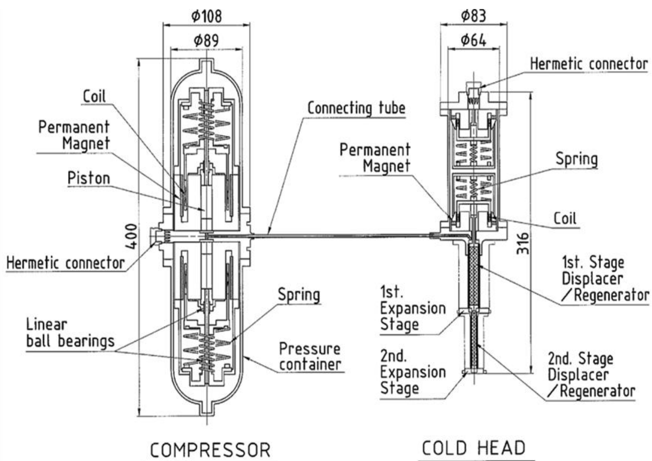


Figure 1. Cross-sectional view of the existing two-stage Stirling cryocooler (from *Advances in Cryogenic Engineering*, vol. 49, Pg. 1430)

Table 1. Specifications of two-stage Stirling cryocooler developed for AKARI

Maximum power input	AC 90 W
Cooling capacity	0.2W at 20 K
Operating frequency	15 Hz
Compressor mass	7 kg
Cold head mass	2 kg
Working gas	^{4}He of 1 MPa
Service life	> 1.5 year
Operating temperature of compressor	240 K

DEVELOPMENT OF ADVANCED TWO-STAGE STIRLING CRYOCOOLER

The mechanical cryocooler is a key technology for cooling optical devices to minimize background noise and enhance detection sensitivity; it is required to improve the performance and reliability of the cooler to keep up with the progress being made on optical device performance. Various space missions in JAXA (Japan Aerospace Exploration Agency) have already adopted or proposed to employ the Stirling cryocooler, as shown in Table 2.

The X-ray astronomical mission NEXT (New X-ray Telescope or Non-Thermal energy eXploration Telescope) planned to be launched in 2013 has a cryogenic system for Soft X-ray Spectrometer (SXS) cooling.[6,7] In the design baseline, the SXS is mounted in a cryogenic dewar and is refrigerated by an Adiabatic Demagnetization Refrigerator (ADR) heat sunk to 36 *l* of liquid helium cryogen. To achieve a cryogen orbital life of about 3 years (5 year goal), a 1K-class ^{3}He-JT circuit with two sets of modified 2ST cryocoolers for precooling provides a heat absorption of 10 mW at 1.7 K for the SXS cooling. The parasitic heat load to the liquid helium tank from the outer thermal environment is reduced by another two sets of modified 2ST cryocoolers that provide dewar shield cooling. The design concept for the cryogenic system for the Spectrum-X Calorimeter (SXC)[8] on the SRG (Spectrum-RG)[9] to be launched in 2011 is expected to be the same as that of the NEXT/SXS.

The infrared astronomical mission SPICA (Space Infrared Telescope for Cosmology and Astrophysics)[10,11] with a 3.5 m-diameter telescope maintained at 4.5 K has adopted a new concept for the cryogenic system that uses no cryogen as shown in Fig. 2. The telescope and Focal Plane Instrument (FPI) are refrigerated by a hybrid method of radiant and mechanical cooling for 5 years instead of a massive and short-lived cryogen. The desired temperature of 4.5 K is obtained by the JT circuit, combined with an advanced 2ST cryocooler for precooling to 15-20 K.[12,13] The increased cooling capacity of the 20 K-class 2ST cryocooler, which contributes strongly to the increased cooling capacity of the JT cooler, can enlarge the thermal design margin to accommodate uncertainties or incidents, although the original cooling capacity meets the SPICA requirements. Vibration reduction induced by mechanical cryocoolers is also an important issue for the SPICA mission to avoid mechanical resonance and deterioration of the spatial resolution and pointing stability of the FPI.

Several technical approaches to improve cooling capacity and reliability including vibration reduction were investigated based on this common and proven 20 K-class 2ST cryocooler developed for AKARI to ensure the success of the missions.

Table 2. Stirling cryocooler application for space missions in JAXA

Mission	Cooling object	Cooling capacity (Configuration)
SUZAKU[14*](X-ray Astronomy Satellite)	X-ray spectrometer	2.5W@100K(1-stage)
AKARI[*](Infrared Astronomy Satellite)	Telescope and detector	0.2W@20K(2-stage)
SELENE[15*](Lunar Exploration Satellite)	Gamma ray spectrometer	1.5W@80K(1-stage)
ISS/JEM/SMILES[16,17] (Earth Atmosphere Observation)	Superconductive submilimeter mixer	0.2W@20K(2-stage)
Planet-C (Venus Climate Satellite)	Near-infrared camera	1 W@65K(1-stage)
GCOM-C1/SGLI (Global Change Observation Mission)	Infrared sensor	0.5W@55K(1-stage)
Spectrum-RG[*] (Next X-ray Astronomy Satellite)	Soft X-ray Calorimeter	0.2W@20K(2-stage)
ASTRO-G[18](Radio Astronomy Satellite)	Mixer and low noise amplifier	0.2W@30K(2-stage)
NEXT[*] (Next X-ray Astronomy Satellite)	Soft X-ray Spectrometer (Microcalorimeter)	0.2W@20K(2-stage)
	Soft X-ray Imager (X-ray CCD)	1-2W@80K(1-stage)
SPICA[*] (Next Infrared Astronomy Satellite)	Telescope and detector	0.2W@20K(2-stage[**])
XEUS[*] (ESA-JAXA X-ray Mission)	Detector (TES microcalorimeter)	0.2W@20K(2-stage[**])

[*] International collaboration mission
[**] Low-vibration model

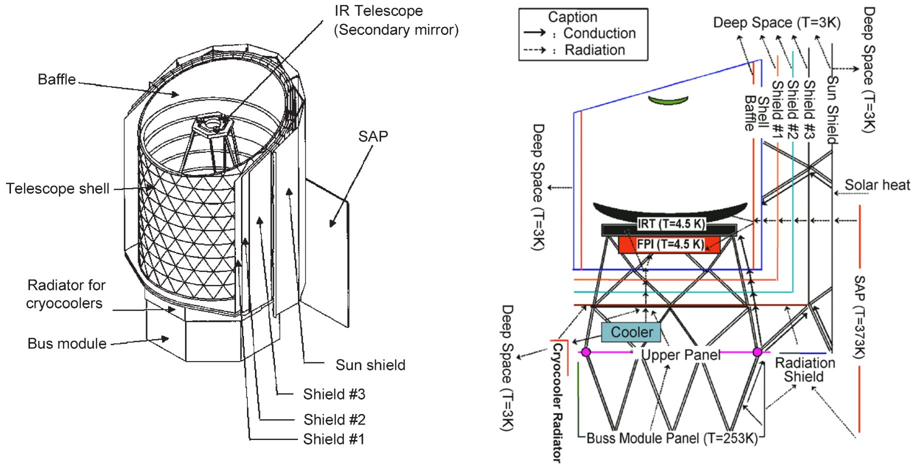


Figure 2. Configuration of the SPICA cryogenic system (from *Cryogenics* 46 Pg. 151-152)

IMPROVEMENT OF COOLING CAPACITY AND RELIABILITY

Enlarge the Second Displacer Diameter

The diameter of the second-stage cylinder of the 2ST cryocooler displacer was increased from 7 mm to 8 mm. As a result, the provided cooling capacity for a nominal input power of 90 W was increased from 200 mW to 325 mW at 20 K in the second cold stage due to the enlarged expansion space; at the same time, the first cold stage carried a heat load of 1 W at 90 K.

Evaluation of Outgas Effect to Cooling Performance

To improve the reliability of the 2ST cryocooler over its 5-year operational life, the effect of contaminated working gas on the cooling performance was investigated. Both CO_2 gas at 0-5000 ppm and N_2 gas at 0-3000 ppm in volume concentration was mixed with the helium working gas to simulate contamination from internal outgassing. The results are depicted in Fig. 3. Note that for the CO_2 case, the temperature of the first stage sharply increases when the concentration of CO_2 gas exceeds 500 ppm. It is presumed that the CO_2 gas is solidified in the stainless-steel-mesh regenerator in the displacer because the CO_2 solidification temperature of 146 K at its partial pressure equivalent to 500 ppm volume concentration inside the cooler is higher than the temperature range of the first stage displacer. This would cause the heat exchange performance of the regenerator to deteriorate causing a temperature increase of the first and second cold stages. In contrast, for the N_2 case, the temperature of both the first and second stages increases gradually as the amount of N_2 gas increases from 0 ppm to 5000 ppm. These results imply that the second stage is stably operated for cooling as long as the concentration level of CO_2 gas (N_2 gas) is maintained at less than 500 ppm (1000 ppm), and that the first stage is vulnerable to working gas contamination. It is also found that contamination of the working gas leads to irregular motion of the displacer.

The outgassing products emitted from component parts of the 2ST cryocooler were evaluated because the purity of the working gas strongly affects the cooling performance during long operation, as implied by the experimental results in Fig. 3. Accurate analyses of outgas species and amounts showed that the permanent magnets, moving coils, and adhesives were the main outgassing sources of H_2O, CO_2, CO and CH_4 inside the cryocooler. Therefore, selection of low-outgassing materials, reduction of the amounts of adhesives, and optimization of the bake-out process for degassing were investigated and verified for higher reliability.

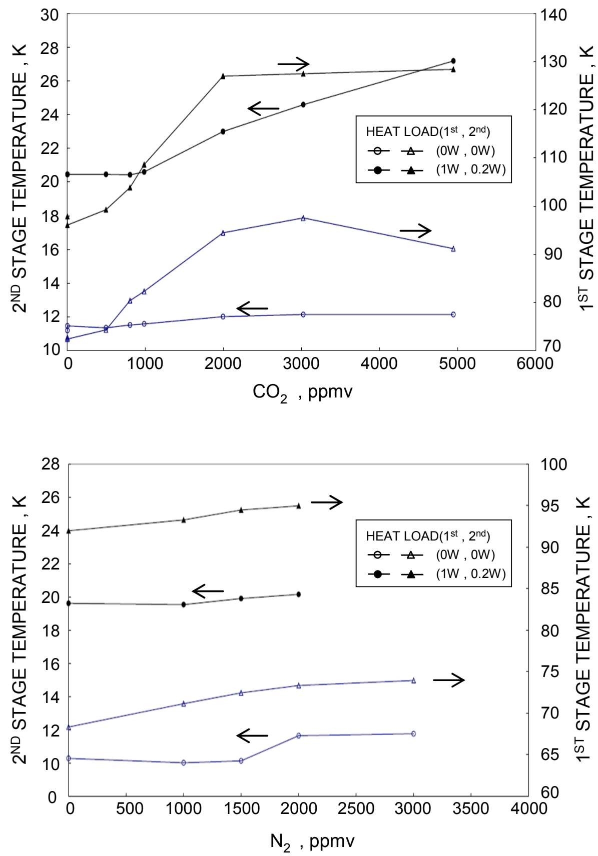
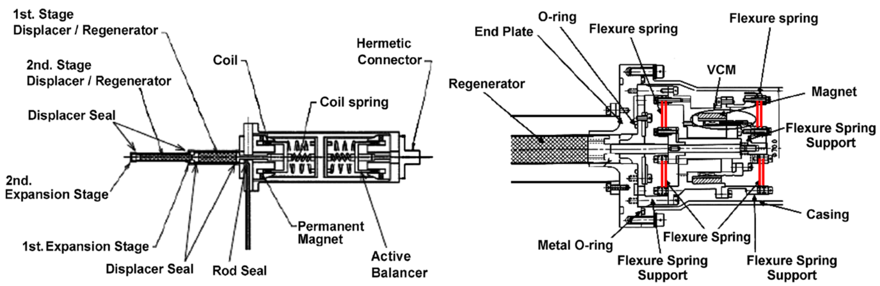


Figure 3. Cooling performance change of two-stirling cryocooler with contaminated working gas

Modification of Displacer Support in the Cold Head

The support structure of the displacer was modified to lower the risk of mechanical fatigue and abrasion from that of the existing 20 K-class 2ST cryocooler. Figure 4(a) depicts the support structure of the displacer in the cold head of the existing 2ST cryocooler. The displacer is supported by coil springs and contact seals that cannot avoid sliding contact with the inner surface of the cylinder. Displacer abrasion by sliding contact may increase the displacer clearance gap and cause working gas contamination after running for many years. Figure 4(b) shows that, to realize a no-contact structure between the displacer and the cylinder, flexure springs that have been adopted already in the proven single-stage Stirling cryocooler[19] operated on board SUZAKU and SELENE were installed. The flexure springs maintain alignment of the displacer attached on the driving shaft, and thus control the small clearance between the cylinder and the displacer that seals the expansion space. Although the contact area is reduced due to this support configuration, at the top of the second displacer, sliding contact is hardly avoidable because the first and second displacers are connected by pinning with slight motion perpendicular to the driving axis. However, the cooling performance test results reveal that the clearance of the second displacer may be increased to 2.5 times the current design value to reduce abrasion risk while maintaining the same cooling capacity. Higher cooling performance with the second cold stage temperature close to 16 K was obtained using this support modification as shown in Fig. 5.



(a) Original design of cold head (b) Modified design of cold head

Figure 4. Schematic view of two-stage Stirling cryocooler cold head.

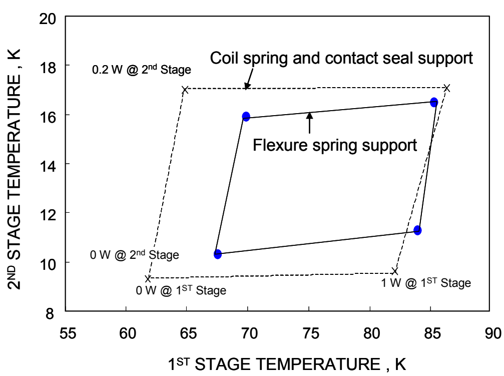


Figure 5. Improved cooling capacity due to flexure spring support for 80 W power input

Further adjustment of the driving conditions was evaluated and was found to improve the cooling capacity of the modified 2ST cryocooler. The achieved performance was 0.2 W at 16.0 K at the second cold stage and 1 W at 83.6 K at the first cold stage, for an AC input power of 90 W, a working gas pressure of 0.9 MPa (Nominal 1.0 MPa), a driving frequency of 15 Hz, and a voltage phase difference between the compressor and the displacer in the cold head of 170 degrees (nominal 180 deg).

An upgraded 2ST cryocooler engineering model is under development and fabrication based on these modifications and investigations to achieve both higher reliability and higher cooling capacity. A life test is planned to start in 2008 for 5 years or more.

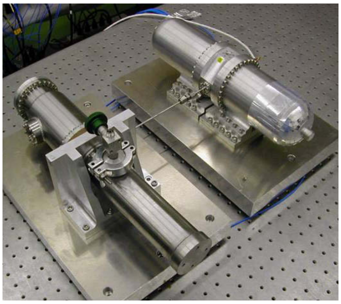
VIBRATION REDUCTION

Reduction of the vibration disturbance induced by the mechanical cryocooler is one of the important technical issues to avoid deterioration of spatial resolution and pointing stability of optical devices in observation missions. The level of dynamic vibration force induced by the 2ST cryocooler and means of reducing it by active balancer control are also under evaluation.

Figure 6 shows the construction of the vibration disturbance measurement system. Three sets of three-axis piezoelectric dynamic force sensors PCB-M260M with specific resolution of 0.001N-rms are installed underneath each of two cryocooler mounting plates with the dimension of 400(L) × 250(W) × 40(H) mm to detect the dynamic disturbance force transferred from the cryocooler. An inertial shaker is also introduced to calibrate the measurement system as a reference vibration source. Figure 7 shows the calibration test configuration. The PCB force sensors are shaken by an inertial shaker through the cryocooler mounting plate; the excitation force can be measured with a shaker force gauge on the connecting point. Consistency between the excitation force by an inertial shaker and the measurement value of the PCB force sensor is confirmed as a frequency response function and the effective measuring frequency range is specified as less than 200 Hz.

The 2ST cryocooler has an active balancer opposed to the displacer in the cold head; it works as a counterweight that is synchronized with the driving frequency and cancels out the dynamic vibration of the displacer oscillation as shown in Fig. 1. Since the displacer motion is largely governed by the inner working gas flow while it is driven simultaneously by the electromagnetic force, the synchronized voltage input to both the active balancer and the displacer is not enough to reduce the disturbance force substantially. Fine tuning of the driving voltage amplitude and phase difference for the active balancer is required to minimize the disturbance force induced by the cold head. Table 3 shows the disturbance force amplitude in the driving axis at the 15 Hz drive frequency of the cold head and the compressor for 50 W power input. The dynamic force in the direction of the driving axis can be reduced to approximately 1/50 in the case of fine-tuned active balancer operation; this can be achieved through careful adjustment of the driving voltage, compared to the case of no-active balancer. Simultaneous reduction of vibration force induced by the compressor is presumed to be the results of reduction of the vibration transferred from the cold head through the connecting transfer tube.

Driving the active balancer by a voltage signal with higher harmonic frequency elements was attempted to reduce the higher harmonic disturbance forces. Measurements shown in Fig. 8 indicate that the vibration level of the cold head is reduced to less than 0.1 N-rms in the region of 15-150 Hz when a voltage signal that includes independently predetermined higher harmonic frequency elements is input.



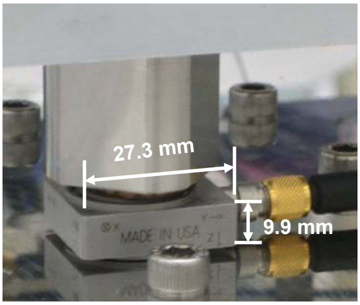


Figure 6. Appearance of measurement configuration for mechanical vibration force induced by two-stage Stirling cryocooler on the mounting plate (left) and piezoelectric three-axis force sensor (right)

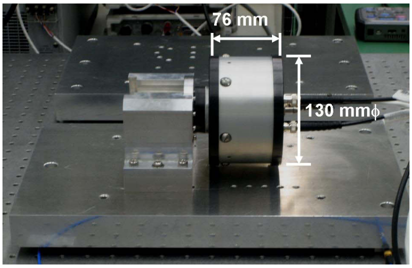


Figure 7. Calibration test configuration with an inertial shaker exciting the force sensors underneath the cryocooler mounting plate

Table 3. Comparison of vibration force at driving frequency 15Hz for 50 W power input

Driving mode	Cold head (N-rms)	Compressor (N-rms)
w/o active balancer	2.64	0.37
w active balancer fine-tuned	0.06	0.14

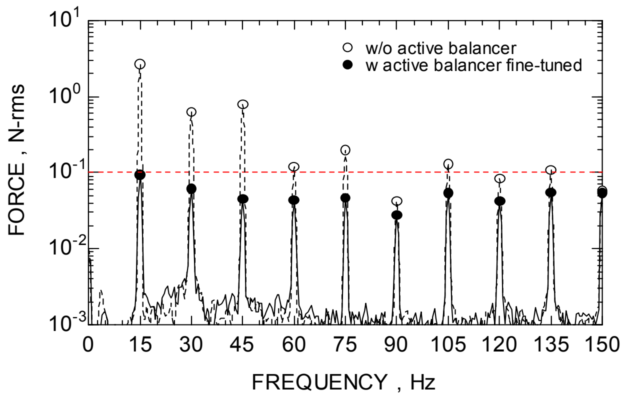


Figure 8. Spectrum of disturbance force amplitude induced by cold head

CONCLUSIONS

This paper describes the improvements in cooling performance and reliability achieved with the development of an advanced two-stage Stirling cryocooler addressed to the requirements of future space missions. The advanced cooler was based on an existing cryocooler for the infrared astronomical satellite AKARI. Specific findings included:

1) Stable cooling performance is achieved when the concentration of CO_2 and N_2 gases contaminating the working gas are kept less than 500 ppm and 1000 ppm of volume concentration, respectively. Outgassing analyses were used to verify the higher reliability.
2) Higher cooling capacity due to avoidance of displacer slide contact was achieved by supporting the displacer on flexure springs, and by increasing the diameter of the second displacer. Additional tuning of the driving condition provided a cooling capacity of 0.2 W at 16 K for 90 W power input.
3) The vibration forces induced by the two-stage Stirling cryocooler were evaluated with a measurement system of piezoelectric force sensors that was calibrated using an inertial shaker as the reference vibration source. The vibration force level of the cold head was reduced to approximately 1/50 of the no-active-balancer case. Using an input voltage signal combined with predetermined higher harmonic frequency elements was able to reduce the vibration forces to less than 0.1 N-rms in the region of 15-150 Hz.

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