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PT410 Cold Head dynamic characterization

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Abstract. In the design of cryostats that use cryocoolers, the characterization of the cryocooler is crucial for vibration-sensitive applications. This study focuses on the PT410 pulse tube cryocooler from Cryomech, using modal and microvibration testing. Modal testing examined the natural frequencies and vibration modes of the cold head. Microvibration testing determined the interface forces caused by the cryocooler's operation on the mechanical mounting interface and the accelerations induced on the thermal interface of the cold stages. The first bending modes of the cryocooler were identified through modal testing. Forces were directly measured using triaxial load cells, and accelerations of the cryocooler stages were recorded with accelerometers. The characterization was performed on a test bench designed for precise measurements. The cold head of the cryocooler is supplied by a helium flow pulse, which introduces vibration. Consequently, an additional support was added for the supply line elements. This study characterizes the dynamic response and mechanical stability of the PT410 cold head based on measured data and testing results. These results are essential for the designing and developing of cryogenic systems where controlling vibrations is critical.

1. Introduction and objectives

INTA was the responsible for the design, manufacturing and validation of the ground-based cryostat for the X-IFU instrument (ESA's L-Type mission ATHENA)[1]. This cryostat would provide the adequate environmental conditions for the calibration of X-IFU during months. The dynamic characterization of cryocoolers is critical for vibration-sensitive applications. For our cryostat, the microvibration levels were one of the main requirements. The objectives of the proposed tests were to measure the interface forces of the coldhead and the microvibrations during its operation, since the cryocooler manufacturers did not provide this information. These tests were also relevant to gain experience and knowledge for the integration and the determination of the involved constrains.

2. Modal test

Initially, a modal test was conducted on a cryocooler suspended from a crane via a sling and springs to simulate free-free conditions. This setup was used to approximate the cold head natural frequencies and incorporate them into the simulation, thereby enabling a more accurate representation during microvibration testing. This approach ensured a better understanding of the cold head dynamic behavior, leading to a more realistic and effective analysis in the complete system context. The test was performed using an impact hammer to apply forces in the X and Y directions on the copper cold head spacer. The data was collected through measurements from the 3 accelerometers placed by means of X60 adhesive and Kapton in 3 different positions: cold head, 1st stage, and 2nd stage.



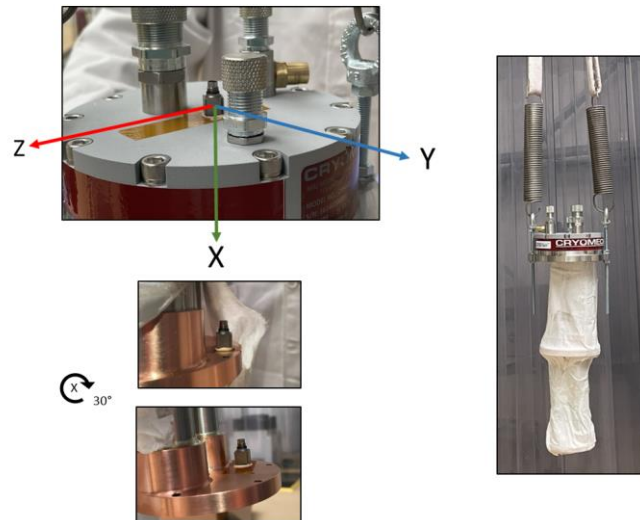


Figure 1. Free-free condition and accelerometers location.

An impact hammer, equipped with a force sensor at its tip, was used to strike the cryocooler. The hammer provides a controlled, measured force input to the structure. The location and angle of impact were carefully chosen to excite a wide range of frequencies and modes.

When the impact hammer strikes the cryocooler, it generates a force impulse, and the accelerometers measure the resulting vibrations. The force input and vibration response were recorded simultaneously. From the recorded data, Frequency Response Functions (FRFs) were calculated. These functions represent the relationship between the input force and the output response in the frequency domain. FRFs helped in identifying the natural frequencies (peaks in the FRF), damping ratios, and mode shapes of the cold head.

Table 1. Modal results measured vs simulation.

Mode	Modal type	Freq. measured (Hz)	Freq. simulated (Hz)	Error (%)	Damping (%)
1	1 st Bending XY	245.8	241.2	1.91	0.30
2	1 st Bending XZ	352.2	352.5	0.09	0.21

The frequencies closely aligned with those calculated in the simulation, thus validating their reliability. They served as inputs for the subsequent microvibration test, ensuring a representative experimental setup with controlled conditions and reliable instrumentation. The damping values remained below 1%, indicating low damping capacity within the cold head. Consequently, they could not be relied upon for damping purposes in future endeavors or alternative experimental settings.

2.1 Validation

From the measured Frequency Response Functions (FRFs), a modal model was derived. Using this modal model, additional FRFs were synthesized. This section visually demonstrates the adequacy of the model's modal fit to the experimental data. To achieve this, the cumulative measured FRFs were presented and juxtaposed with the cumulative FRFs synthesized from the modal model. The errors obtained are below the $\pm 5\%$ conservative margin assumed according to the calibration certificate of the load cells (± 1) and the error induced by the mounting.

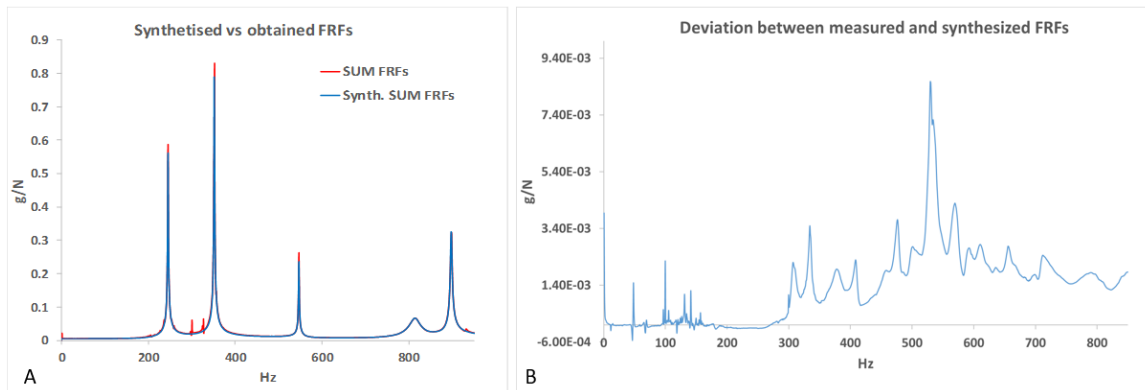


Figure 2. A. Measured FRF (red) and synthesized (blue) B. Deviation between the curves.

3. Test bench modal test

A test bench was designed to enhance the rigidity and to accurately determine the interface forces. [2] This bench was constructed and designed by INTA structural personnel, providing a cost-effective and efficient solution. The same method of the cold head modal test was followed. By accelerometers measuring the impact of the hammer in specific positions, the vibration response of the test bench has been obtained.

During the tests, the floor introduced effects that influenced the determination of the bench’s natural frequencies. This interference underscored the need for further refinement in the test setup. By addressing these floor-related issues, the accuracy of frequency measurements and interface force determinations can be significantly improved, leading to more reliable data for future analyses and applications.

3.1. Validation

In this section, the graphic illustrates how well the modal model fitted the data obtained from the test. To achieve this, the sum of the measured Frequency Response Functions (FRFs) was compared with the sum of the FRFs synthesized from the modal model. The floor modes begin at 40 Hz, limiting the validity of the interface force measurements to frequencies up to this value. The errors obtained are below the $\pm 5\%$ conservative margin assumed according to the calibration certificate of the load cells (± 1) and the error induced by the mounting.

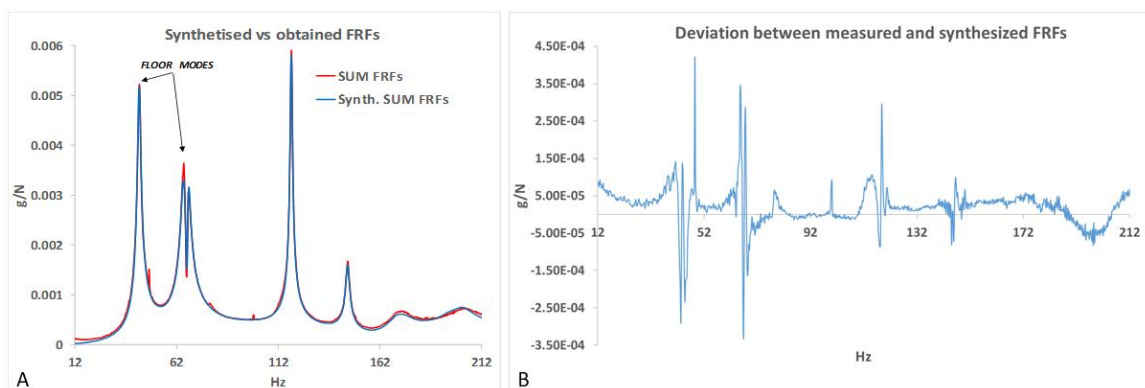


Figure 3. A. FRFs synthesized (blue) and measured (red). B. Deviation between the curves.

4. Microvibration test

The microvibration test was conducted to measure and analyze the small-scale vibrations generated by the cold head during operation. By performing a microvibration test, the dynamic behavior of the cryocooler can be characterized, understanding how it interacts with other components and the

mounting structure. The data obtained from microvibration testing helped in optimizing the design of the cryocooler and its integration into a larger cryostat system. The design of isolation systems can effectively dampen or isolate vibrations, protecting sensitive equipment and allowing the measurement of the specimen's performance inside the cryostat.

4.1. Set-up configurations studied

One important question for accurate microvibration measurement was understanding the influence of the reservoirs, remote motor and helium pipes allocation respect to the cold head in terms of the force transferred. Different configurations have been studied:

- *Standard 1.* Remote motor and reservoirs positioned in a plane perpendicular to the Z-axis of the cold head, with the pipe connecting the cold head to the remote motor at a steeper angle.
- *Standard 2.* Remote motor and reservoirs positioned in a plane perpendicular to the Z-axis of the cold head, with the pipe connecting the cold head to the remote motor at near right angles.
- *Standard 2 + bellows.* Same configuration as standard 2 but with dampers installed. The height of the cold head with respect to the remote motor changes.
- *Vertical 1.* Remote motor and reservoirs in the Z-axis, vertically over the cold head. The remote motor has a pendulum degree of freedom.
- *Vertical 2.* Remote motor and reservoirs in the Z-axis, vertically over the cold head. The remote motor pendulum degree of freedom is restricted.

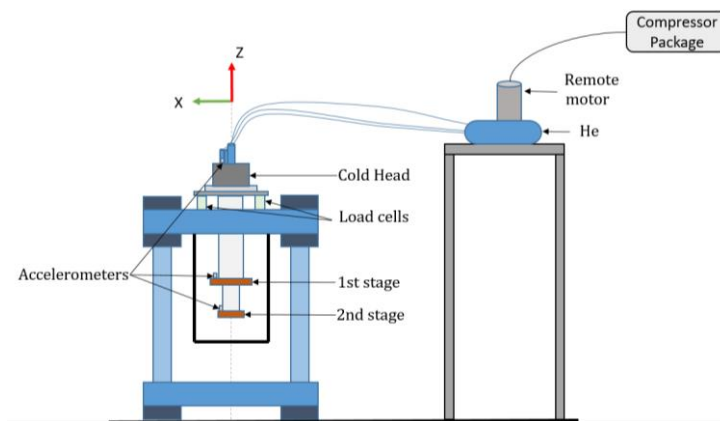


Figure 4. Test bench set-up for standard positions.

The vertical configurations induced large values for the Z component of forces on the cold head, exceeding the design goals, leading to the rejection of these configurations. Therefore, only the results of the desirable configurations were analysed.

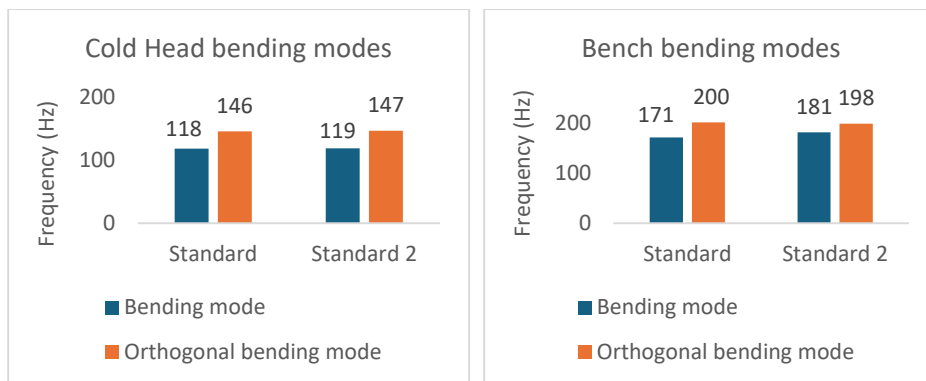


Figure 5. Bending modes.

As can be observed, there was not much variation in the modal frequencies between the two positions, Standard and Standard 2, the curvature of the flexlines had little effect. The obtained data according to the bending and flexible modes of each element are:

- First bending modes of the cold head: 115 Hz and 150 Hz.
- First flexible modes of the test bench: 180 Hz and 200 Hz.

4.2. Forces and moments

The forces and moments for the characterization of the cold head have been obtained. It is important know the moments induced, which is often a missing information about cryocoolers. Based on the literature [3], an RMS value of 0.2 is considered appropriate for the predesign phase and simulations. However, the data obtained differ from this value because it involves a flight cryocooler rather than the ground-based one studied.

Table 2. RMS values between 0.5-30 Hz

Conf.	Fx (N)	Fy (N)	Fz (N)	Mx (Nm)	My (Nm)	Mz (Nm)
S	0.273	0.863	1.300	0.172	0.453	0.235
S2	0.548	0.373	1.736	0.211	0.473	0.240

The next figure shows the forces and moments obtained in the standard 2 configuration which is similar to the standard 1 configuration.

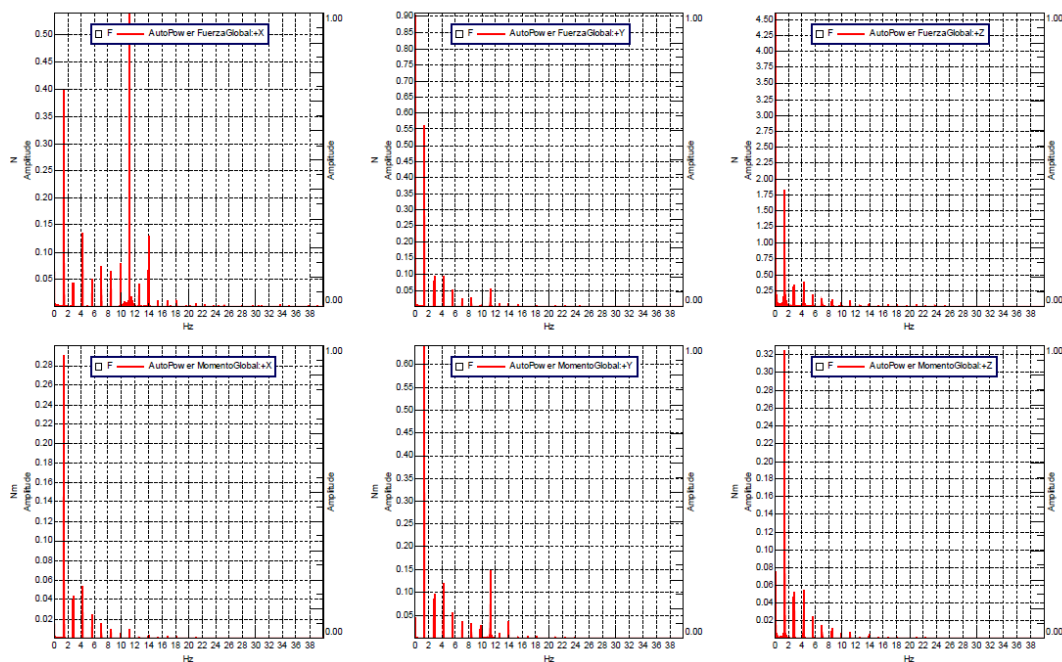


Figure 6. Forces and moments in standard 2 configuration.

Beyond the 30 Hz threshold, it is noticed that amplitudes remained relatively subdued, suggesting minimal perturbation to the test conditions in accordance with referenced literature [4]. However, a noteworthy observation arises at 12 Hz, where a potential coupling between the resonant mode and either the ground or the diverse mounting accessories becomes apparent. The same information can be represented in the time domain, the moments and forces in the cold head during operation.

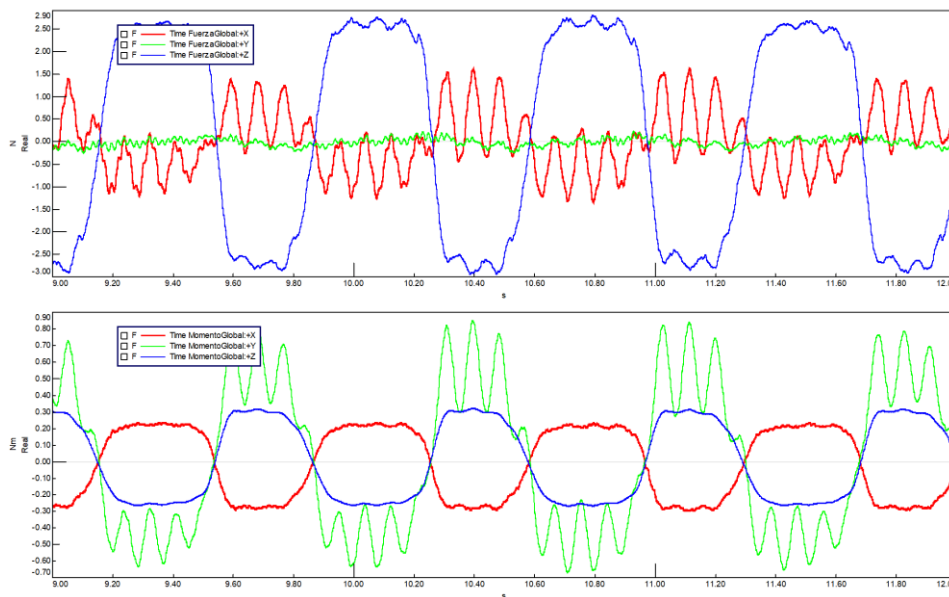


Figure 7. Forces (upper) and moments (down) in standard 2 configuration.

5. Conclusions

The response forces of the cold head, including the moments, which is usually hard information to find manufacturers' specifications, have been obtained. From the modal test, the first bending modes of the cold head obtained are 245.8 Hz in XY direction and 352.2 Hz in XZ direction which were in accordance with the simulation.

The forces generated by the cold head during operation on the test bench have been measured with an accuracy of $\pm 5\%$. The significant components of these forces are observed within the first 40 Hz. The primary modes of the "floor-test bench-cryocooler" assembly have been determined being the first bending modes of the cryocooler 115 Hz and 150 Hz.

The vertical acceleration levels were measured to be low in all cases. In the X and Y directions, the levels at Stage 2 were high enough to observe the low-frequency harmonic components in all scenarios. For future analyses and measurements, a more rigid foundation or negative stiffness isolators could improve accuracy.

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