Qualification Status of a New 100 Nms Reaction Wheel Flywheel Mass with Integrated Damping System

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Abstract

With increasing demand for heavier satellites, an improved reaction wheel with larger flywheel mass storing up to 100 Nms was developed while maintaining the decade-proven TELDIX™ bearing design. An active damping system was designed to reduce loads caused by the heavier mass to the bearings.

This paper describes the verification effort for integrating a resonant, multi damper system into the flywheel mass. The patented damper system minimizes the loads generated at the bearing level. The paper also describes the performance impact of the extension of the flywheel mass from 68 Nms to 100 Nms. The major challenge was the fine-tuning of the quality-factor to allow reuse of the heritage ball bearing unit. This resulted in a flywheel mass design with an active damping system, combining over 80 mechanical parts. The latest phase of the development included extensive testing and analysis to identify the adjustment parameters needed to handle such a complex system and verify full reproducibility for large quantity manufacturing. The qualification test campaign of the RSI 100-220/60 is scheduled for completion in May 2026.

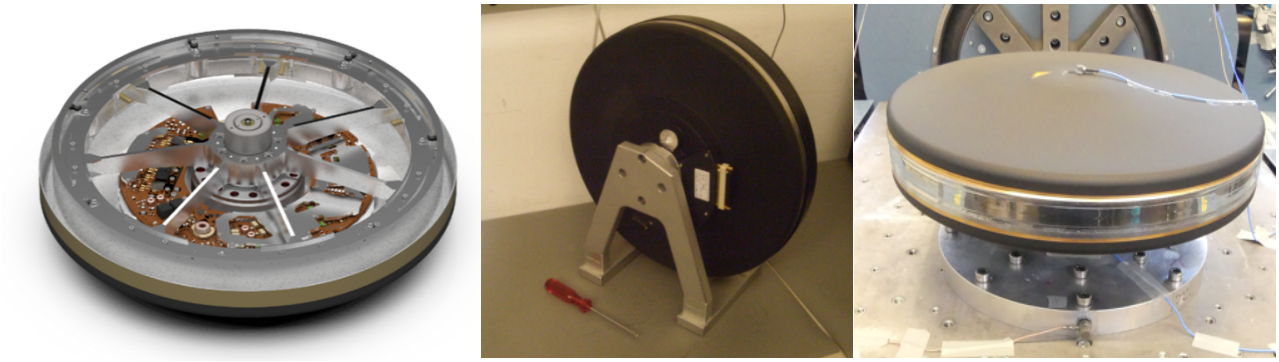

Figure 1. TELDIX™ reaction wheel RSI 100-220/60. CAD model without top cover (left), Engineering Qual Model (EQM) mounted on test fixture (middle) and on shaker during vibration testing (right).

Introduction

The market demand for large reaction wheels with an angular momentum storage of 100 Nms and more to maneuver heavy satellites for telecommunications up to 7000 kg has risen sharply in recent years. At the same time, the requirements of high vibration loads are increasing, especially during the launch or separation of the satellite from the spacecraft. The factor limiting the load capability of reaction wheels is the resulting forces on the ball bearings. FEM analysis confirms that the primary influence on the bearing load is the first axial resonance of the flywheel mass.
With the intention to further employ the decade-proven TELDIX™ bearing unit with heavier reaction wheels, Rockwell Collins Deutschland GmbH has developed a highly efficient damping system for structural resonances of flywheel masses to keep the launch loads compatible with the load capability of the TELDIX™ bearing unit design. This damping concept is based on vibration absorbers and complies with the same vibration level requirements as the existing 68 Nms reaction wheel assembly which was designed in 1972 and collecting in-orbit heritage since.

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The design goal for the new reaction wheel assembly was a momentum storage capacity of 100 Nms at ±6000 rpm with a motor torque of 220 mNm over the full speed range for both directions, hence the name RSI 100-220/60†. The main dimensions of the reaction wheel assembly are a diameter of 420 mm and a height of 135.6 mm. The total mass is less than 12 kg. This reaction wheel contains a novel 100 V wheel drive electronics unit including a powerful feedback mode to recuperate the stored energy.

In the early development state of the mechanical design, preliminary finite element method (FEM) simulations were performed to investigate the eigenfrequencies and eigenmodes of the novel reaction wheel. Moreover, special consideration was given to the bearing forces, which serve as the defining constraints for the permissible vibration level. The flywheel mass significantly impacts bearing loads, with the first axial resonance mode posing the greatest stress; therefore, targeted damping of this mode is essential. The quality-factor (Q-factor) is a primary indicator of damping performance, representing the ratio between energy storage and dissipation within the system. Preliminary FEM analysis for estimating the expected bearing loads showed that a Q-factor of the flywheel mass of less than 15 must be achieved as a design goal to fulfill the specified requirements. With over 80 individual components, the assembly process is highly complex, making a reproducible Q-factor challenging to achieve.

In order to not reduce the requirements on the vibration loads and consequently remain an attractive manufacturer of reaction wheels on the market, Rockwell Collins Deutschland GmbH has developed a highly efficient damping system for structural resonances of flywheel masses in reaction wheels, which is based on a vibration absorber. With this damping concept, it is possible to continue the usage of the well-known reliable TELDIX™ ball bearing unit that has been used successfully for many years even with larger and heavier rotating masses.

Requirements & Design Specifications

All reaction wheels manufactured by Rockwell Collins Deutschland GmbH have been operated with a well proven TELDIX™ ball bearing technology for many years. The maximum allowable force for the axial and transversal load has been determined by detailed theoretical and empirical analysis. This limit results from a change of geometry caused by the load. With increasing load on the ball bearing unit, the contact angle of the balls is changed. Due to this change, when reaching a certain load, the contact ellipse is cut off at the borders of the toroidal races. Higher peak forces are induced into the balls leading to a nonlinear stress increase. The calculation of the ellipse cut-off with very conservative assumptions gives the limits of the bearing force.

For a predominant share of reaction wheel units, analysis confirms that the bearing load is the limiting factor for the allowed dynamic loads. The bearing load calculation methodology has been iteratively refined over several years and validated against empirical data derived from extensive customer qualification campaigns.

The reaction wheel's dynamic behavior is characterized by four primary modes [Figure 2], where the flywheel's mass serves as the dominant influence on the resulting resonance frequencies:

- Rp – parallel rocking mode of the flywheel
- Ra – anti-parallel rocking mode of the flywheel
- Ax1 – first axial translational mode of the flywheel
- Ax2 – second axial translational mode

† RSI = German abbreviation of "reaction wheel with integrated electronics"

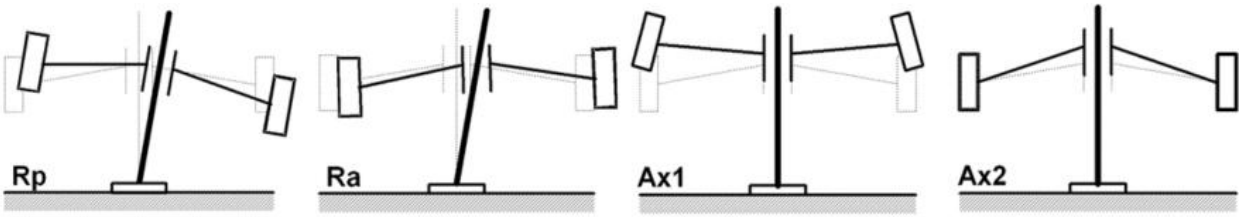

Figure 2. Primary Eigenmodes of TELDIXTM Reaction Wheels

The bearing loads in heavy reaction wheels are primarily driven by the Q-factor of the first axial (Ax1) resonance mode. The Q-factor is defined as the ratio of resonant amplification to the input excitation level (and is thus inversely proportional to the damping ratio). It serves as a critical measure of energy dissipation. Systems with a high Q-factor exhibit weak damping. Consequently, achieving a low Q-factor for heavy flywheels is essential to minimize bearing stress and ensure structural integrity under specified vibration loads.

For the new development of the RSI 100 wheel type the lowest possible Q-factor should be achieved to withstand the specified vibration levels. Initial estimations of the maximum bearing forces were derived from a FEM-based assessment. For this analysis each random vibration power spectral density as well as each shock response spectrum is transformed to a transient signal in the time domain. The duration of these signals is equivalent to the duration of vibration that the wheel assembly will experience during testing at the equivalent level.

Using this time dependent signal, a response analysis of the resulting forces on the balls of the bearings (or rather the springs representing the balls) is carried out with the Siemens NX CAD-System. From the output of this response analysis, a histogram of the resulting forces is then calculated by means of a MATLAB® script. For the sinusoidal loads the resulting peak force spectra are directly calculated by the CAD software (computer aided design).

Mechanism Architecture

The concept of tuned mass damper utilizes a secondary small mass to oscillate with a phase shift relative to the larger main mass, thus creating a relative motion within the combined system which then is slowed by friction and dissipates the excess energy (Figure 3).

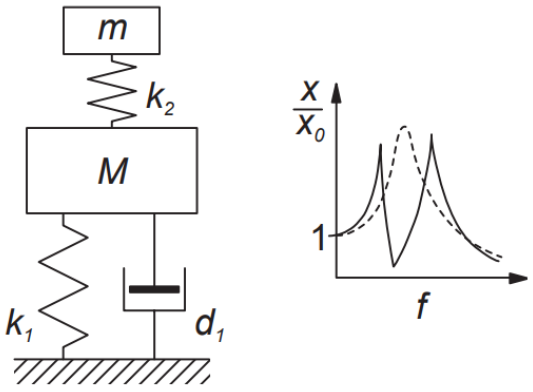

Figure 3. Tuned Mass Damper Concept with 2 Masses

A significant challenge in the system design is the precise parametric definition required for consistent functionality across the complete operational envelope. This involves frequency tuning to align with the target vibrational mode (Ax1) while simultaneously addressing the higher acceleration levels, which corresponds to the system's response to increased g-forces. The parametric definition must satisfy both frequency and load requirements to ensure the system meets the desired performance criteria.

For the reaction wheel application, this system incorporates a main flywheel mass “M” with specific properties (notably Ax1) and the tuned mass dampers, each consisting of an auxiliary mass element “m” (tilger mass) attached to the flywheel, with a damping ring that takes the function of the spring k2, and intentional friction elements for the purpose of energy dissipation and vibration damping.

The objective is to achieve effective damping at a specific frequency (Ax1). For this to happen, the damper masses must disengage from the flywheel within a precise frequency band around the critical Ax1 mode during excitation.

There are two critical timing issues:

1. If the mass dampers uncouple too early (very loose coupling):
   They should remain coupled until the target frequency Ax1 is reached. When disengaging too early, the system behaves as if the damper masses are independent, and damping is ineffective at Ax1. The system exhibits two separate resonances, leading to poor overall damping performance at the desired Ax1 frequency.

2. If the mass dampers uncouple too late (tight coupling):
   They should detach before the system reaches its natural resonance frequency. If they remain coupled too long, the mass dampers effectively become part of the main mass (M) and move in-sync. While this lowers the system's natural frequency (Eigenfrequency) slightly, there is hardly any damping effect because the mass damper components are not dissipating energy at the target frequency.

To achieve optimal damping, the uncoupling of the mass dampers from the flywheel mass within the targeted frequency range is crucial. This precise timing directly influences the system's natural frequency (Eigenfrequency) and determines whether the damping occurs effectively at the target frequency Ax1.

Development Process

The development of the 100 Nms flywheel mass was based on long-standing experience with smaller reaction wheels. From their successful application, a framework for the initial technical goals and the construction of the first test vibration mass was established. However, the results of the early development phase were disappointing: The increased flywheel mass and dimensions led to unexpectedly poor damping behavior, characterized by very high Q-factors.

To move the development forward, the engineers returned to fundamental analysis methods, including CAD and particularly FEM. These FEM analyses, which were not available at the time of the heritage flywheel mass development, enabled a detailed study of the relative movement between the flywheel mass and the damping system. Based on this technical analysis, the first significant improvement regarding the Q-factors was achieved. By changing the material and modifying the damping rings, the Q-factor was improved, although still not in the desired range. At the same time, the entire tuned mass damper system was better understood, forming the basis for further optimizations.

Initial optimizations of the mass dampers to realize lower Q-factor proved unusable, as they resulted in unacceptable high unbalance values. Nevertheless, the Q-factors were already very close to the desired target values at that time. Further adjustments allowed both the unbalance to be minimized and the required radial preload to be set. This was achieved by integrating the centering rollers directly into the auxiliary masses, enabling more precise alignment.

Finally, setting the axial preload represented the last milestone. Due to the larger diameter compared to the heritage models and the modified damper rings, the position and number of spacer sleeves were re-selected. This resulted in the final, fully optimized version of the vibration mass, fulfilling all required performance characteristics.

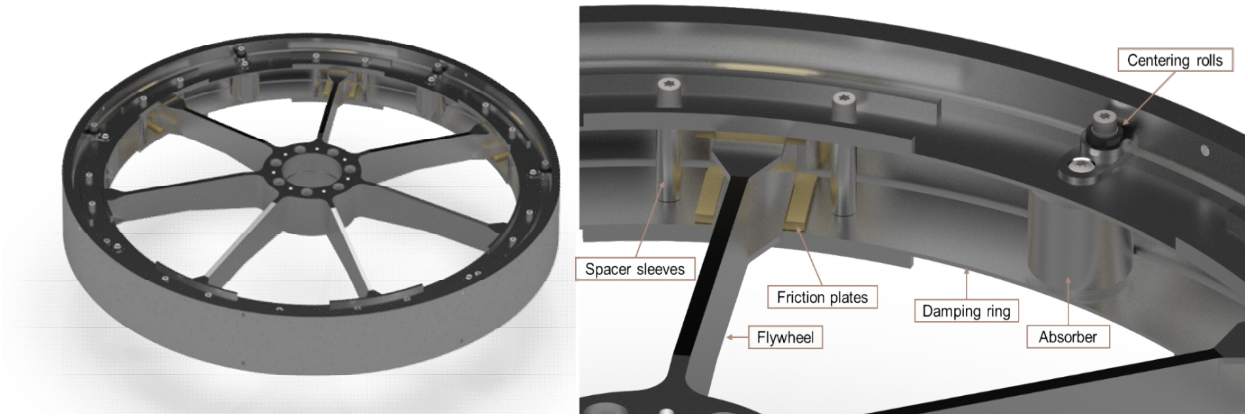

Figure 4. Assembled Flywheel Mass with Damping System (Left), Axial Damping System (Right)

Assembly

To achieve the precise timing and the desired eigenfrequency, two preloads (the radial preload and the axial preload) are essential. Both preloads influence the critical parameters, but they are not independent; adjusting one affects the other. This interdependence necessitates a precise calculation and adjustment of both. The complexity of this system is further amplified by the large number of components. For example, the flywheel mass assembly comprises over 80 individual parts. Each part has its own function and presents specific manufacturing challenges.

The following single parts are particularly noteworthy (see Figure 4):

- Flywheel
  The flywheel, comprising a central hub, seven spokes, and an outer mass ring, is manufactured as a single piece from forged stainless steel. This ensures dimensional accuracy and minimizes unbalances. The critical points for surface finish and dimensional accuracy are the wheel spokes, the wheel friction surfaces, and the centering grooves of the damping system. The axial and radial preload of the damping system is adjusted at these interfaces.

- Damping ring
  The components utilize custom-matched material for their vibration properties. Precise manufacturing of wall thickness is critical and represents a key manufacturing challenge, as it directly influences the system's vibration characteristics. The damping rings function as spring elements and directly influence the damping of the system.

- Mass damper (absorber)
  To effectively counteract or absorb vibrations at a specific frequency, the resonant frequency of the tuned mass damper itself must be carefully matched. Consequently, the mass of the absorber is directly defined by the targeted frequency of the Ax1 system, ensuring optimal resonance and thus maximal damping of vibrations. However, the precision required for this mass is extremely high; manufacturing tolerances and any deviations from the calculated value can significantly impair the performance, potentially leading to ineffective vibration control.

- Spacer sleeves
  The spacer sleeves are used to adjust the axial preload between the friction surfaces. Their precise length is essential for the axial preload.

- Friction plates
  The friction plates, together with the damper, are the primary energy dissipators during damping. The flatness of the friction plates surface must match the counter-surface of the flywheel.

Insufficient flatness or a non-matching profile can lead to uneven wear, unwanted sliding, and consequently, reduced damping performance.

- Centering rolls
  Insufficient radial preload or misalignment of the Centering Rolls can lead to improper radial positioning. This can cause uneven loading on the damper elements, unwanted sliding between the absorber and the flywheel, and ultimately reduced damping effectiveness, as well as potential issues with detachability.

The entire flywheel assembly contains 80 components with critical interdependencies for proper functionality of the damping system. When combined with typical tolerances and processes, the resulting assembly becomes immensely complex, and properties are almost non-predictable. Therefore, all manufacturing and assembly steps were scrutinized to minimize tolerances and variation with the goal to eliminate variables, achieve the desired damping properties and overall performance, and allow mass production.

For reliable axial and radial preload, precise preparatory work is crucial for successfully executing the two main assembly steps:

- Axial preload is achieved using spacer sleeves. To precisely calculate the length of the sleeves in the system, the height of the friction plates is crucial. The adhesive bonding of the damping elements to the damper ring must be uniform, even after the adhesive cures.

- Radial preload is set using friction rolls. It is achieved through precision milling and adjusting the friction rolls to the flywheel mass.

An analysis of the CAD data reveals the relevant tolerance chains that influence the setting and value of the preloads. For the axial preload and the radial preload, there is a respective sequence of components whose tolerances affect the final preload value. The tolerance chain for the axial preload extends from the flywheel mass, through friction plates and their adhesive layer, over the damping ring thickness, and up to the spacer sleeves (Figure 4). Modification of these preloads and the relevant components can directly impact the mass distribution. So later microvibration measurements must confirm compliance to low unbalance requirements.

Test Campaign

The tests at the flywheel level were initially started in 2020 with mixed results. During some tests, it was clearly visible that the damping effect was as expected, while repetitive tests showed that in some cases the damping effect did not occur at all. Overall, the testing was hard to reproduce and the Q-factor inconsistent.

To trace the cause, the previous approach for testing the flywheel needed to be redefined. Through various settings and tests, it was determined that it is essential to identify when the absorbers decouple from the flywheel mass and that the frequency band of the detachment is crucial. Furthermore, the number of absorbers that decouple from the flywheel mass is important. For accurate assessment, the flywheel has been equipped with the maximum possible number of sensors available to the test system. With these insights, it was possible to optimize the application of the required radial and axial pre-tension during the assembly process of the flywheel. The following plots show the results of the testing at 4g excitation.

Our understanding of the testing process was greatly enhanced by examining key vibration test diagrams related to Q-Factor determination, shown in Figure 5. The relevant data is shown by the green line (the acceleration of the flywheel) which represents the loads the bearing unit needs to withstand. The other colors show acceleration data on tuned mass absorbers.
Figure 5 (top): When the mass dampers are coupled too loosely to the flywheel and thus disengage prematurely at low frequency, two distinct resonance frequencies are seen, as all damping masses move in sync and independent from the flywheel mass. No energy is removed during the critical resonance phase

of the flywheel, leading to poor overall damping performance. The system vibrates with much higher amplitude than expected, directly reflecting in a poor Q-Factor result of 33.4.

Figure 5 (middle): The opposite problem is presented. The absorbers are not detaching at all or, often long after the vibrations were at their most intense. While seemingly attached for longer, this delay means the dampers are ineffective at precisely when they were needed most (during the high-amplitude oscillations). As a result, the system fails to dampen effectively, again yielding a high Q-Factor and confirming the poor damping. The Q-Factor of the flywheel mass configuration was 20.1 (better than in the previous test but also higher than desired).

These distinct behavior modes (triggered by improperly tuned mass damper uncoupling) underscore the critical importance of achieving the correct preloading of the dampers. Extensive testing, focusing on fine-tuning the pre-tension settings, was undertaken. This involved not only adjusting the forces but also verifying the resulting absorber detachment behavior. Through this iterative process, a reliable procedure was established. This procedure enabled the precise adjustment of the pre-tension to achieve the ideal release timing, as is seen in Figure 5 (bottom). Consequently, it became possible to consistently reproduce the desired damping effect, achieving a Q-Factor of 7.3 (well below the design threshold of 15), indicating highly effective vibration control. The nearly perfectly tuned flywheel and mass damper absorbers work in concert and the damper masses disengage at the targeted frequency band.

Before any tests, the assumption was that the kinetic energy of the flywheel mass would be dissipated through friction at the contact points between the spokes and the damping ring. Over the test campaign, the importance of fine-tuned absorber decoupling is the critical factor to achieve effective damping of Ax1 of the flywheel mass.

Q-Factor results on full reaction wheel assembly:
With the newly implemented assembly process, the first flight models achieved an average Q-factor of 10.6. This value is lower than the previously assumed Q-factor threshold of 15.

Successful integration with ball bearing unit:
Tests were conducted where the flight models were integrated with the ball bearing unit. These tests were all successful and passed, as demonstrated in the next sections.

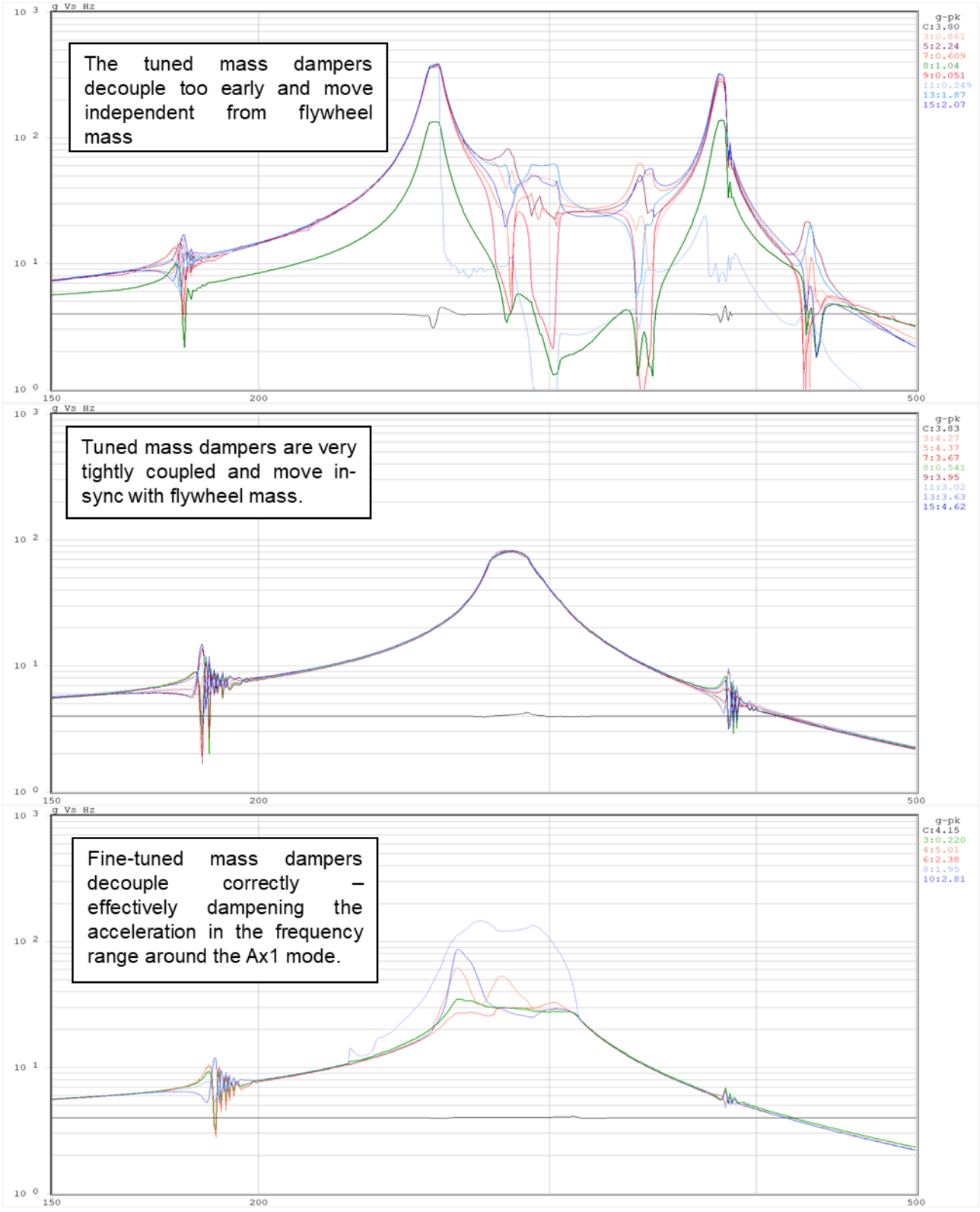

Figure 5. Acceleration measurements (color-coded for various sensors) during vibrational excitation. When adjusted correctly (bottom), inverse movement of the mass dampers results in significant broadening and reduction of the flywheel acceleration (green), effectively dissipating energy from the resonance peak.

Microvibration results during qualification campaign

Any mechanism transmits undesired forces and moments to the satellite. Knowing the microvibration signature (particularly for continuously operated, high-speed reaction wheels) is therefore crucial for every satellite mission. Especially for rotating devices, the force/momentum signature exhibits various design-specific features and hence is a great tool for (stability) analysis and health monitoring. During the RSI 100 flightworthiness qualification campaign, microvibration measurements have been performed at the following stages:

a) Initial measurement
b) After the vibration test campaign (standard sine, random, quasistatic and shock levels)
c) After the thermal vacuum test (TVAC)
d) After electromagnetic compatibility testing (ongoing at time of paper submission)

Each microvibration measurement yields a total of 6 waterfall diagrams (see Figure *6*), showing the correlation of wheel speed, frequency, and amplitude of the generated force (momentum) for each cartesian axis. The test is performed for all four quadrants of the reaction wheel, but here only the counterclockwise rotation in acceleration mode is presented.

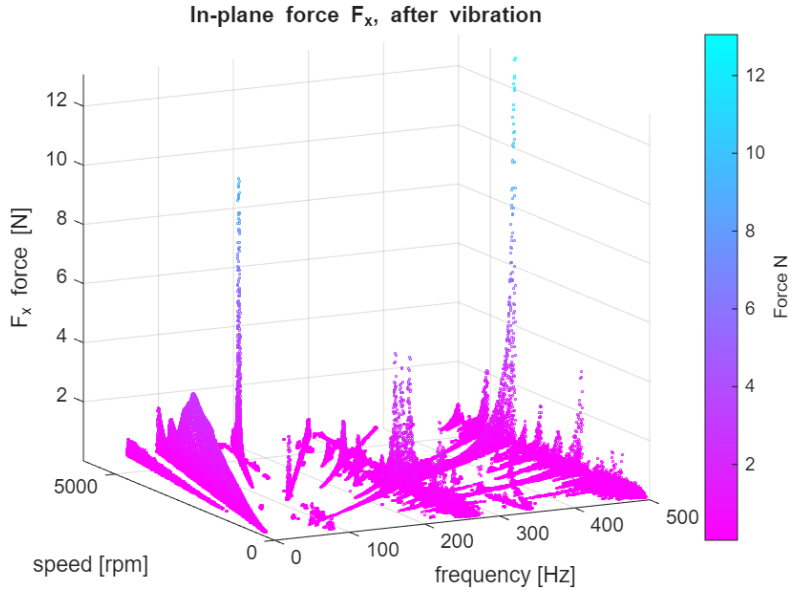

Figure 6. Waterfall Diagram of Fx (=in-plane), Color-Coded for Force Amplitude

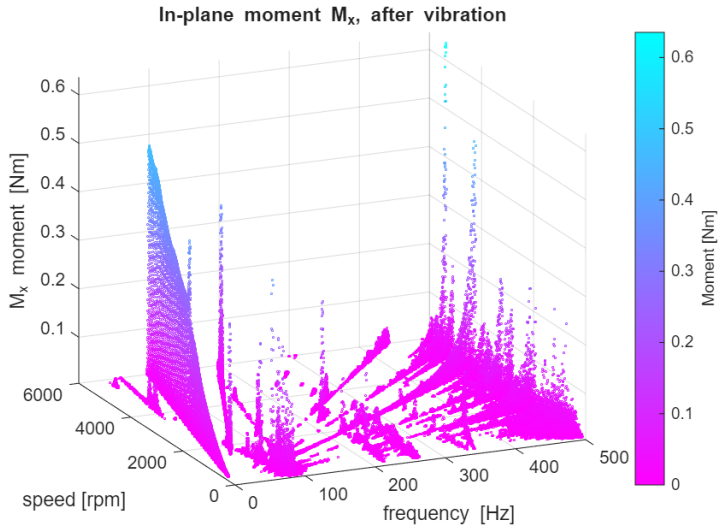

Figure 7. Waterfall Diagram of Mx (=in-plane) color-coded for moment amplitude

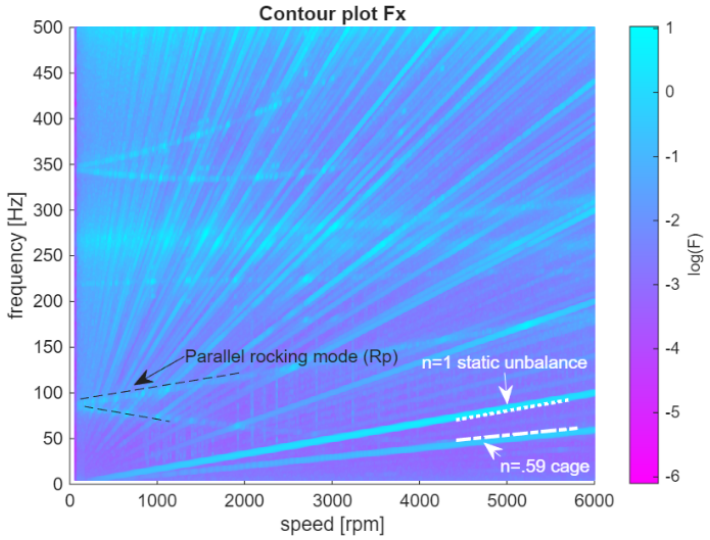

Figure 8. Contour diagram (top view of above Fx waterfall diagram), showing exemplary harmonic forces (white lines) and geometric design resonance frequencies (black lines).

Due to the wheel's symmetric design around the rotational axis, x and y data are presented in a common "in-plane" diagram. The out-of-plane diagrams are omitted here, as the disturbance generated by the wheel in the z axis is by magnitudes smaller than the in-plane forces / moments and therefore of less interest to the satellite designer / operator.

The contour diagram ("Campbell plot", Figure 8) exhibits the systematic design features of the reaction wheel. For example, the forces generated by static unbalances of the rotating mass (n=1) and the cage (n=0.59) can be seen as harmonics with respect to the rotational frequency of the wheel (Figure 8, white lines). Geometric design resonances are expressed at constant frequencies or (during excitation) split up into branches, e.g. the rocking mode of the rotating parts (black lines, 83 Hz @ 0 rpm).

When viewed from the side, a "peak hold" diagram (Figure 9) was used to better assess the evolution of the wheel's behavior over the test campaign. From an operator's perspective, the forces / moments shown in the broad 0-100 Hz range are the most critical, as these are caused by the unbalance and are thus speed dependent. In contrast, peaks at higher frequencies are driven by structural resonances, so they exhibit at distinct and narrow frequency ranges (thus enabling easy mitigation by employing specific damping

mounting structures during satellite design). When analyzing the evolution of peak hold data, only minor shifts in frequency and amplitude are observed on the initial curve (blue). Between vibration and TVAC testing (red, green), no further alterations are observed.

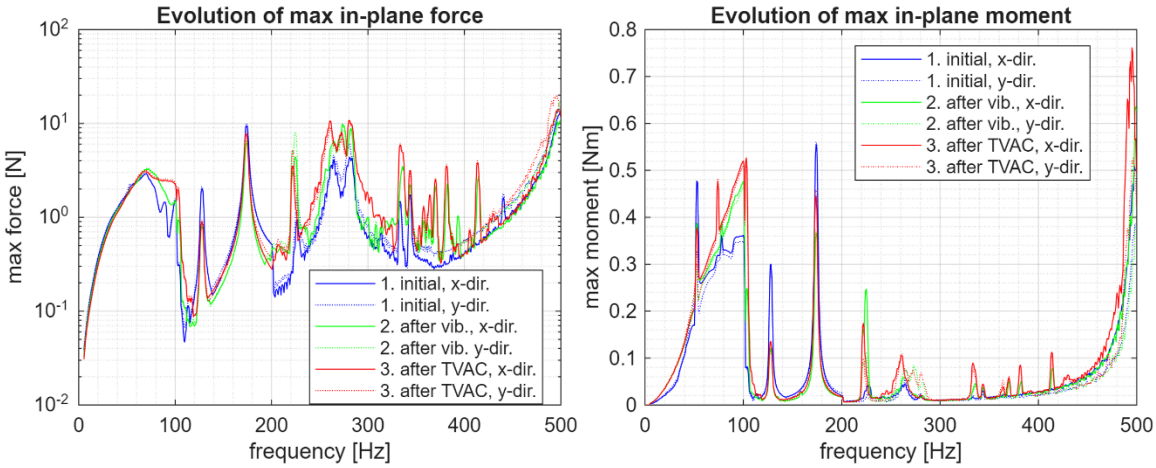

Figure 9. Evolution of peak hold forces (left) and moments (right) in rotational plane. Data in the 0-100 Hz range are directly caused by unbalance effects of the rotating mass.

Typically, the static and dynamic unbalance are the predominant forces and moments impacting the disturbances induced by the reaction wheel to the satellite. Hence, great effort is taken to realize small unbalance values and monitor their evolution as well as to present these values to the satellite designer / operator.

The static unbalance is well below the limit of 5 g*cm over the entire speed range. For a spoked wheel design (as used for the RSI 100) the unbalance shows a speed dependency, as the centrifugal forces will increase the diameter of the rotating mass rim. Although the effect is microscopic, it can be observed in the slope change of the static unbalance for speeds greater than 4000 rpm (Figure 10, top left).

Especially in the dynamic unbalance diagram (Figure 10, top right), a resonance effect at 3200 rpm is visible. In the original moment data at the n=1 harmonic (Figure 10, bottom right), this resonance is emphasized even more. During rotation, the unbalance causes moments / forces at the n=1 harmonic. These moments / forces then can excite structural vibration. If coinciding with a structural resonance, significant higher (or lower) signals are detected by the sensors, depending on constructive (or destructive) interference. So, while the measured force (or moment) is real, the mathematical conversion to static (or dynamic) unbalance at these resonances can yield artifacts.

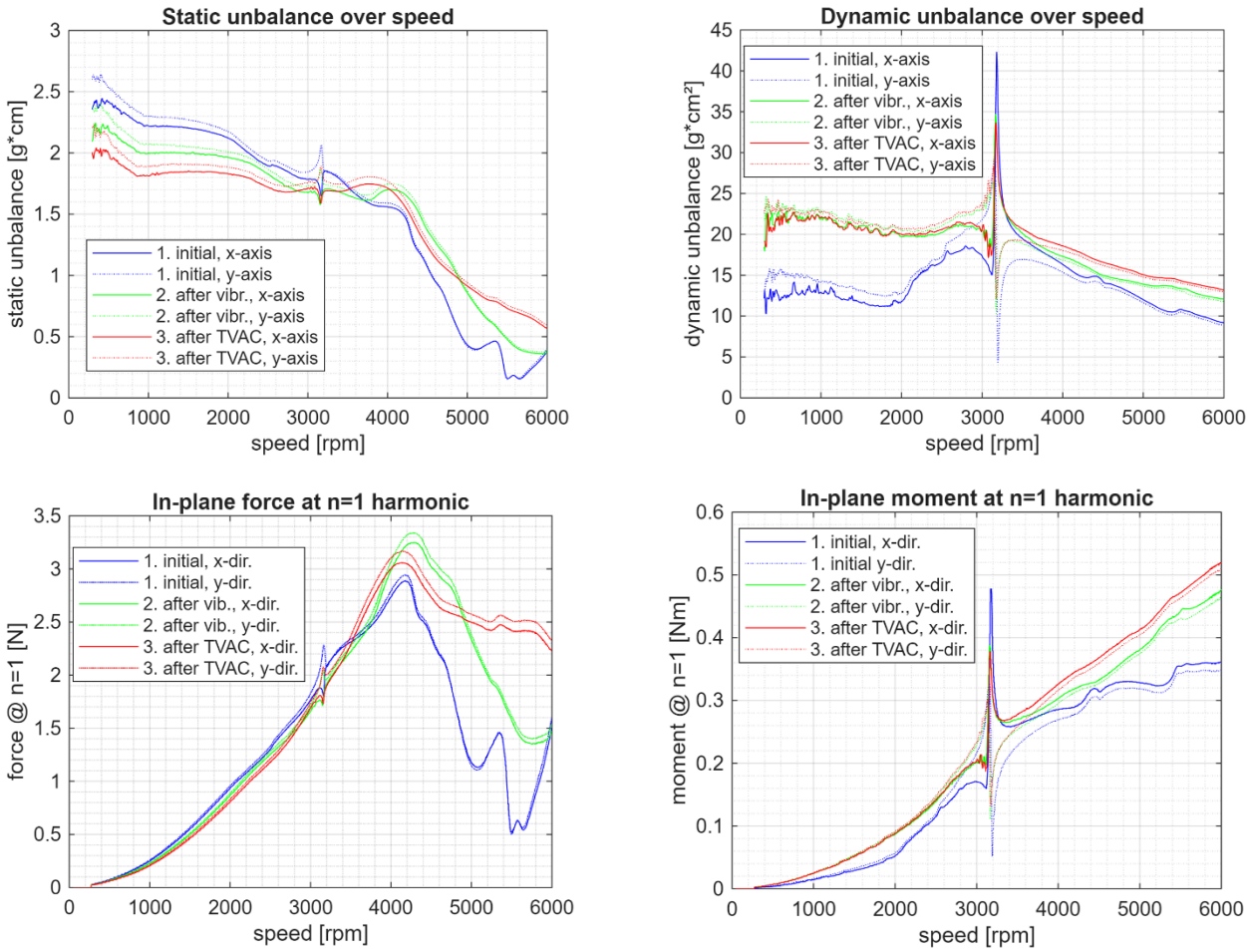

*Figure 10. Evolution of static (top left) and dynamic (top right) unbalance over speed, values are well below the limit of 5 g*cm and 30 g*cm². These unbalances are derived from the n=1 harmonic force (bottom left) and moment data (bottom right). See the text for cause of the resonance peak at 3200 rpm.*

With this background, the artifactual peak in the dynamic unbalance can be ignored and the same observation as for the static unbalance can be drawn: During vibration testing, the dynamic unbalance increases, but then remains unchanged for the following TVAC test. The dynamic unbalance complies with the limit of 30 g*cm².

The microvibration measurements over the various stages of the qualification campaign confirmed the RSI 100's ability to withstand the stress and loads applied during the qualification campaign. The EQM microvibration characteristics obtained are as expected and well aligned with the design and mechanical analysis. The wheel's unbalances are well below the allowed limits. As the test campaign was based on typical mission requirements, the confidence is high that this new reaction wheel will meet the requirements of future space missions.

Conclusion & Lessons Learned

The approach of developing a large 100 Nms reaction wheel while keeping the proven existing bearing unit was challenging but rewarding. The newly designed and patented flywheel mass with integrated damping system adds complexity to the system which can only be controlled with precise manufacturing and assembly techniques.

A major lesson learned over this journey was the importance of well-defined and reproducible manufacturing and test steps to realize the outcome promised by simulations and analyses. Successful development was only possible by the open and honest collaboration across the involved departments with a common intent to reduce variation (and thus complexity) to an otherwise unmanageable, highly unpredictable mechanical system.

When these modifications are implemented correctly, the required load reductions on the bearings are realized. Once the effects were understood and the assembly steps fine-tuned and controlled, repeatable manufacturing with an identical outcome was possible. This was confirmed by the qualification campaign. The qualification campaign consisted of thermal vacuum cycling, vibration, and shock test, as well as electromagnetic compatibility tests (among others) with intermittent microvibration measurements.

The successful completion of each individual test setup concluded that the new RSI 100-220/60 with the active damping concept is suited for spaceflight in all the relevant environments from LEO to interplanetary missions. This achieved the goal of developing a flywheel mass with 100 Nms around a 30-year-old bearing unit design which was never intended for such large flywheel masses while keeping the well proven performance and longevity of the bearing unit unaltered.

In combination with the new, integrated damping system of the 100 Nms rotating mass, the RSI 100-220/60 utilizes the decade-proven and trusted design of the bearing unit while providing a larger angular momentum storage capacity needed for future large satellites.