Improving E-ELT M1 prototype hard position actuators with active damping

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ABSTRACT

During the advanced design phase of the European Extremely Large Telescope (E-ELT) several critical components have been prototyped. In particular, a representative section of the E-ELT primary mirror has been assembled. It is equipped with complete prototype segment subunits including prototype edge sensors and position actuators. One purpose is to test various control strategies and evaluate the achievable performance. One of the segment subunits is equipped with three prototype hard actuators, i.e. typically piezo actuators being passively stiff. To improve such type of position actuators various active damping techniques were implemented and tested. In this paper an active damping strategy based on positive position feedback is introduced. The theoretical and implementation test results are presented. This also includes results related to vibration transmission.

Keywords: E-ELT, primary segmented mirror, active damping, control, vibration, position actuators

1. INTRODUCTION

The European Extremely Large Telescope (E-ELT) is a programme led by ESO for a next generation optical and near-infrared, ground-based telescope. Its optical design is based on a three-mirror anastigmat with two folding flat mirrors sending the beam to either of the two Nasmyth foci along the elevation axis of the telescope. The elliptical primary mirror has a diameter of 39m and consists of 798 quasi-hexagonal aspherical segments with a circumscribed diameter of 1.4 m and 50 mm thickness. Each segment is connected to the back-structure by means of a segment support system, which is actively positioned by three position actuators (PACT). Figure 1 shows the E-ELT with a cut-out of 4 prototype segments including back-structure and segment support systems. The scheme in the lower right corner illustrates the role of the position actuators in the segment support system. Three PACT are used in each segment support system to control the out of plane degrees of freedom (piston-tip-tilt) of the segments.

The main purpose of the active dynamic control of M1 is to limit wavefront aberrations by compensating back-structure deformations originating from changes in telescope elevation, temperature and wind forces, but also limiting the effect of vibration transmission to segments and thus degradation of the wavefront.

During the design phase of the E-ELT programme two types of position actuators were investigated and prototyped, soft and hard actuators. Both are two-stage actuators, consisting of a coarse stage to reach the necessary stroke and a fine stage to reach nano meter tracking error. The soft actuators are characterized by low open-loop stiffness of the fine stage typically based on a suspended voice coil system. The necessary dynamical stiffness for wind force rejection is generated by closed loop control. Hard actuators are already stiff in open-loop typically based on preloaded piezo or hydraulic systems. Closed-loop control for them serves mainly to position them accurately. Because of their passive stiffness for many aspects, especially earthquake safety, hard actuators are more convenient than soft actuators. However, mounted in the segment support they exhibit a resonance with low damping in the frequency region of around 50Hz, which could be excited by vibration sources when left undamped.

Figure 2 shows the open loop transfer functions of a soft and hard PACT in a segment assembly, 10Hz resonance for soft, 50Hz resonance for hard PACT. For the soft PACT the resonance falls within the closed-loop

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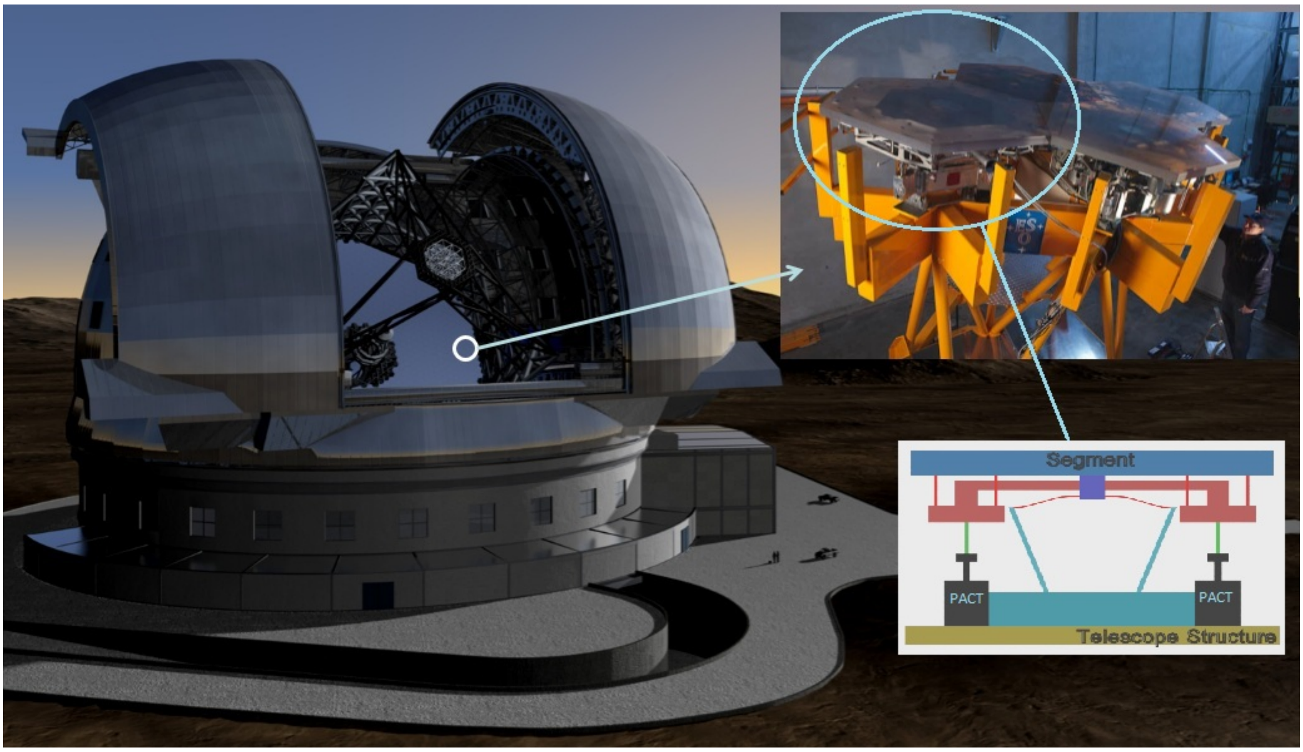

Figure 1. Schematic view of the 39m E-ELT primary mirror with a cut-out of 4 prototype segments incl. segment supports and back structure. Schematically the role of the PACT is illustrated when moving a segment in piston direction.

bandwidth and thus can be damped by standard PID control with position feedback. However, for the hard PACT the 50Hz resonance is well above the achievable closed loop bandwidth, and thus standard PID control cannot considerably damp such resonance. The amplification of vibrations at such resonance frequencies is considered as a risk for wavefront control and thus there is a strong motivation to introduce active damping in order to reduce vibration transmission.

It has been shown that damping could be achieved by introducing additional nested feedback loops based on additional sensors, especially acceleration or force sensors, e.g. for the E-ELT M5 system.[1] However, seen the large number of PACT in the M1 system (roughly 2400 pieces) this would lead to additional complexity (more components, more cabling) reflected in an increase of the number of failure modes, increased cost and a more difficult obsolescence strategy. Therefore, an alternative approach has been explored, called positive position feedback, which uses the existing position feedback to damp the critical resonances, and thus simplifying the plant, before closing an ordinary position loop. The approach has been published many years ago,[2] but has been rarely used, mainly because it is restricted to a specific class of systems. As will be shown in this paper the main benefit of positive position feedback is to improve hard PACT damping without adding any additional sensor using a measurement-based tuning procedure with few tuning parameters. This has been demonstrated in simulation and in tests in the M1 Test Facility, a 4-segment prototype mock-up of the E-ELT M1 system.[3]

Section 2 will introduce a more detailed motivation for the damping of hard PACT. Section 3 will then give an overview and classification of different damping approaches considered originally. Then in Section 4 several aspects of the positive position feedback approach are discussed. This includes a detailed description of the implementation and tests done with an existing hard PACT prototype. The necessary modifications are described in detail, so that they could be used for similar systems. Finally, Section 5 concludes the paper with a perspective of using positive position feedback for M1 control.

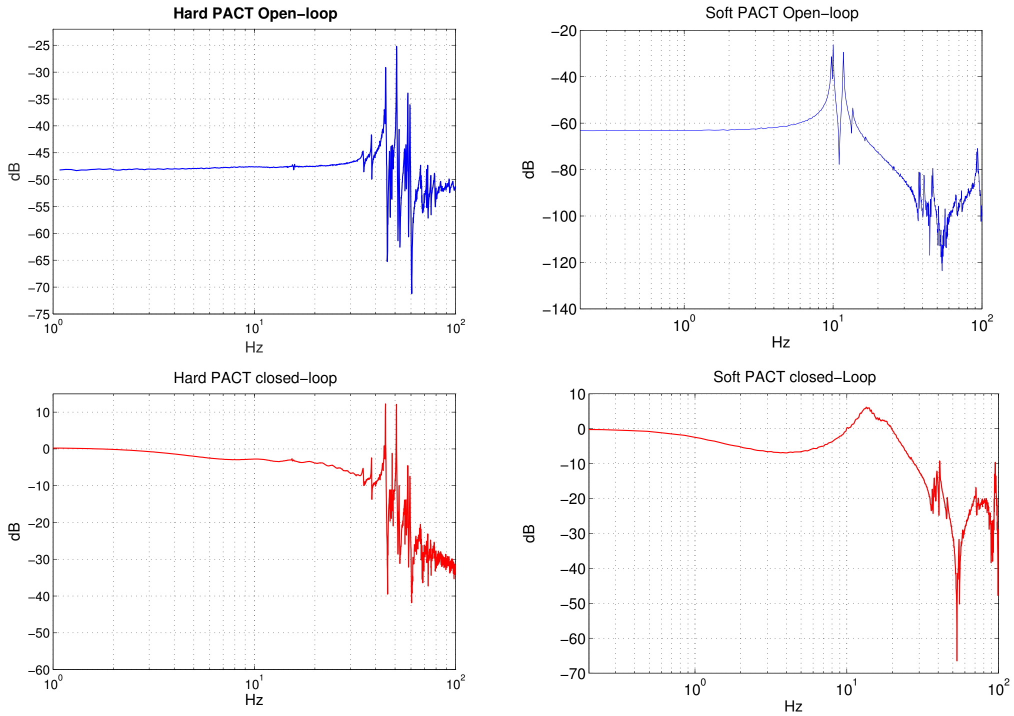

Figure 2. Measured open and closed-loop frequency responses of soft and hard PACT in a segment assembly. In case of soft PACT the resonance at 10 Hz is damped by control, in case of hard PACT the resonances around 50Hz remain even in closed loop control. If using notch filters they could be suppressed in the reference closed-loop response but not in the vibration transmission towards the segment

2. MOTIVATION FOR ACTIVE DAMPING IN THE CASE OF PACT

In the context of E-ELT M1 segmented mirror control the closed-loop PACT dynamic control of a segment support has four main requirements:

1. Provide enough dynamic stiffness for wind force rejection
2. Provide enough closed-loop bandwidth for M1 figure control
3. Limit vibration transmission
4. Use the same controller for all segments in all telescope elevations

Reaching enough dynamic stiffness (point 1) for hard PACT is not an issue, because of the large passive stiffness of these devices. This is one of the main motivations to use hard PACT for segment control.

For M1 figure control the position actuators are part of a relatively slow (low frequency) servo system following figure references of the central control system and correcting for deformation of the M1 cell due to the gravity sag, temperature variation and wind. Even though the figure loop is relatively slow, the demanding requirements on small M1 figure residuals (tens of nanometers after adaptive optics corrections) require a position actuator closed-loop bandwidth of roughly 10Hz (point 2). In order to reach such bandwidth of 10Hz in presence of low-damped resonant modes of the M1 segment subunit starting from 45Hz, as it is the case for the hard PACT, it is common practice to apply appropriate filtering in the actuator controller. Typically notch filters are used to cancel the effect of the resonant modes in the open loop transfer function. One disadvantage of such approach is that the exact location of problematic resonant modes needs to be known in advance and needs to be invariant. In case of a limited number of the servo loops this might be sufficient and feasible, but in case of M1 control with roughly 2400 actuator servo loops, tuning of notch filters becomes practically inefficient, very sensitive and most important not robust to the variations and changes of the interesting resonant modes from one segment unit to the other, and especially for different telescope elevations (point 4). In addition to the concerns about robustness of the filtering approach it should be noted that notch filtering used in PACT controllers would not improve the vibration transmission, i.e. even in closed-loop PACT control resonance peaks in the vibration transmission between structure and segments would remain unchanged (point 3).

One possible solution to address the concerns of the last three points with a common approach is to add damping to the resonant modes by means of feedback control, so called active damping. Active damping approaches are similar to any feedback loop concept. Some of them do not require the exact knowledge of the system under control (e.g. resonant modes frequency). However, they are based on feedback between sensors and actuators and thus they have to cope with the limitations of any closed-loop control system that are stability/robustness issues. The main motivation for using active damping in the context of hard PACT control is twofold: i) all 2400 PACT servo-loops could be implemented with exactly the same control parameters robustly without requiring a segment by segment or case by case tuning of servo loop filters, ii) the amplitude of the vibration transmission to the segments could be reduced and hence the overall performance of the M1 mirror due to the observatory equipment vibrations is improved.

In our case of hard PACT, the idea was to use the actuator itself as a integrated damping device containing actuation and sensing. Different sensing devices and possible active damping control algorithms were studied and one out of them, not requiring any additional sensor, has been implemented in the existing hard PACT. All studied approaches provide damping, but they differ substantially once it comes to implementation. In the next section, a brief overview of the main active damping strategies is introduced.

3. OVERVIEW OF ACTIVE DAMPING APPROACHES

The active damping strategies are classified based on the type of the sensors, their physical location and the type of the implemented algorithm.[4] Here, three main potential approaches on the basis of different sensing systems namely, i) accelerometer, ii) force sensor, and iii) position sensor are overviewed.

1. **Acceleration Feedback**

   Accelerometers can be attached to the tip of the position actuator. Then the strategy consists on either to integrate the acceleration signal a to obtain the velocity and then by a direct gain controller K_a introduce damping, i.e.

   $$u = C_{damp} * a = -\frac{K_a}{s} * \frac{s}{s+\omega_b} * a = -\frac{K_a}{s+\omega_b} * a \tag{1}$$

   or by passing the acceleration signal through a second order filter and by generating a force proportional to the output of the filter, i.e.

   $$u = C_{damp} * a = -\frac{K_a}{s^2 + 2\xi_f \omega_f s + \omega_f^2} * \frac{s}{s+\omega_b} * a \tag{2}$$

   where damping factor ξ_f at a selected frequency ω_f are chosen by the designer.

   In practice, acceleration signals at low frequencies are noisy and thus a high-pass filters are included in the controllers. The high-pass filter corner frequency is presented by ω_b.

2. **Force feedback**

   Force sensors need to be put into the load path between PACT and segment support, e.g. being an integral part of the tip of the PACT. The approach is more or less equivalent to the control strategies based on acceleration feedback as long as the acceleration of the base of the segment support is zero. However, there are some advantages with force feedback, i) force sensors have often better sensitivity than accelerometers, ii) the stability properties of the force feedback are often better than the acceleration feedback, because of the differential force measurement the collocation is in general better.[4] Let K_{act} be the axial stiffness of each actuator, then the force sensor y measures the force at the interface of the actuator and the mirror, i.e. $y = -K_{pact}z + u_{act}$, where u_{act} is the active control force of an actuator (see Figure 3). The controller is the integral of the measured force, i.e.

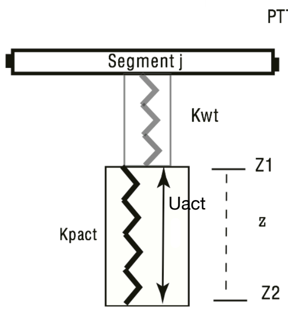

Figure 3. A schema of a segment, PACT and its stiffness and segment support and it stiffness

$$u = C_{damp} * y = -\frac{K_f}{s} * y \tag{3}$$

In practice often high-pass filters are added to the control to remove possible offsets and low frequency perturbations.

3. **Positive position feedback**

   The approach consist of filtering the position signal (from the position sensor) with a second order filter and feed it back to generate a force in a positive feedback constellation. This is not an intuitive approach while in general most of the control loops are negative feedback loops. The effect and damping capabilities are more or less like the acceleration based active damping strategies with an additional property that the control adds -40dB/decade roll-off at higher frequencies.[4] The filter has the frequency ω_f and the damping ξ_f as design parameters:

$$u = C_{damp} * z = +\frac{K_z}{s^2 + 2\xi_f\omega_f s + \omega_f^2} * z \tag{4}$$

   where z is the position signal. One advantage of the approach is that in addition to damping the roll-off of the system under control is also improved, allowing higher frequency gain stabilizations. The approach has also its limitations: since the stability limit is defined at static or zero frequency of the system, i.e. the open-loop static gain shall not be equal to 1, the approach cannot be implemented for systems with open-loop integral behavior such as motors and drives for axis control. These issues will be discussed and demonstrated in detail in Section 4.

From the three most common approaches explained earlier, two approaches require additional sensors: accelerometer and force sensors. Even though force feedback is probably the most powerful method in reaching large damping without major stability risk, hardware wise it is the most complex to implement. Force sensors with the right sensitivity need to be mounted into the load path matching the geometry, not changing the interface stiffness and withstanding the survival loads. In addition, they require sensitive evaluation electronics, which need to be integrated into the PACT and to be maintained over the 30 years of E-ELT lifetime (obsolescence issues). Therefore, in case of PACT, the third approach using the positive position feedback is very attractive since the existing internal position encoder can also be used for the active damping. In this case the segment position actuator itself can be used as an active damping device, reducing the costs considerably. The approach was analyzed and implemented successfully for the M5 unit actuators.[1,5]

It was decided to implement and test the positive position feedback damping schema on the hard position actuators under the segment support system of the M1 test stand. The design approach, its implementation and measurement results are presented in Section 4.

4. PACT WITH POSITIVE POSITION FEEDBACK: DESIGN, IMPLEMENTATION AND RESULTS

4.1 Design and implementation

The original control software of the actuators were modified such that the damping algorithm scheme can be implemented and tested under a segment subunit in the M1 test stand. Figure 4 shows the general concept of the PACT servo-loop with the additional nested loop for the active damping using the positive position feedback. In the same figure the necessary modification of the software algorithm is schematically indicated by a different color.

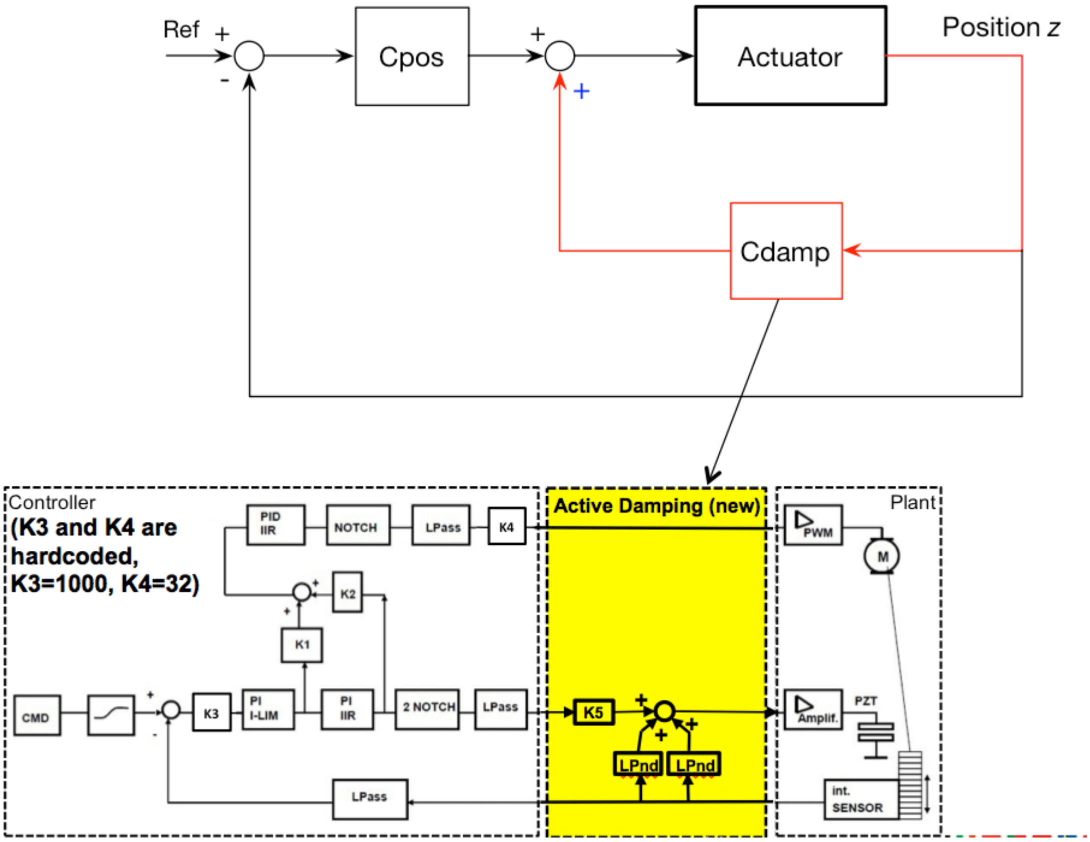

Figure 4. PACT control scheme, including the newly implemented active damping

The design of the active damping controller, its associated parameters, implementation and the verification of the damping loop is followed by the following main steps:

i) The open loop frequency responses from the actuator drive inputs to the position encoder of each actuator under a segment subunit using white noise or sine sweep excitation were derived. Figure 5 shows the excitation signal to one actuator in form of a sine sweep, its encoder response reading and the resulting frequency response (open-loop $G_{pact}(j\omega)$).

ii) The open-loop frequency response ($G_{pact}(j\omega)$ complex numbers as a function of frequency) of the actuator, and the frequency response of the damping controller $C_{damp}(j\omega)$ were used to estimate the closed-loop response and necessary stability response criteria. For the damping filter

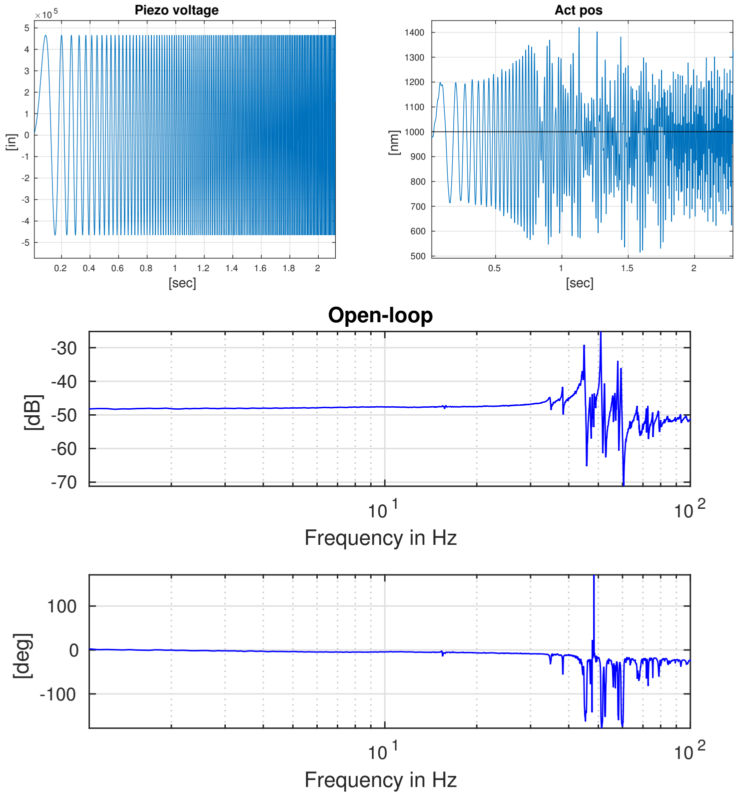

Figure 5. Measured open-loop frequency response of an actuator under a segment subunit in the test stand

$$C_{damp}(s = j\omega) = \frac{K_z}{s^2 + 2\xi_f \omega_f s + \omega_f^2} \tag{5}$$

the parameter ω_f was selected to be close to the first important mode to be damped (45Hz), i.e. $\omega_f = 2\pi 45$. Experience, shows that the damping parameter ξ_f in the range of 0.25 to 0.4 lead to satisfactory results. Using a loop shaping method for the loop transfer response $-C_{damp}(j\omega)G_{pact}(j\omega)$, the damping controller gain K_z is varied until a robust closed-loop response is obtained. Figure 6 shows the Nyquist response and the sensitivity function $S(j\omega)$ for a design with following controller parameters: $K_z = 110$, $\omega_f = 2\pi \times 45$, and $\xi_f = 0.25$.

The robustness of the internal loop is verified by the modulus margin or the largest amplitude of the sensitivity function $\max_\omega |S(j\omega)| < 6dB$ (see Figure 6 right). One should note that the overall stability and robustness shall be verified as well including the outer-loop or the position loop controller C_{pos} (e.g. a PID controller). This is done by verifying stability and robustness criteria for the loop transfer response $-C_{damp}(j\omega)G_{pact}(j\omega) + C_{pos}(j\omega)G_{pact}(j\omega)$.

The estimated response of actuator with the selected control parameters is compared with the open-loop response in Figure 7 where the damping effect on at least three mechanical modes can be observed.

iii) The satisfactory design parameters are implemented in the control software and the damping loop is closed. Using the sine sweep excitations, the frequency response of the actuator with the active damping loop closed is measured. Figure 8 compares the measured frequency response of the actuator when closing the active damping loop with the estimated response from the design. The good accordance of the responses confirms the correctness of the design, and the implementation of the algorithm in the control system. In addition, it shows the advantage of such a measurement (model) based design where in principle changes and optimization of the

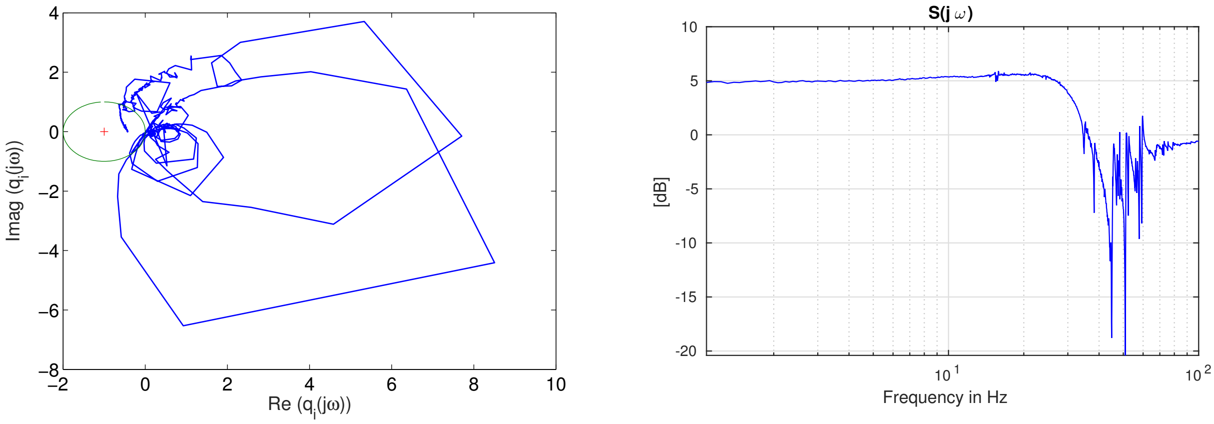

Figure 6. Stability and robustness for the damping loop design: Nyquist curve (left), sensitivity function $S(j\omega)$ and the modulus margin < 6 dB (right)

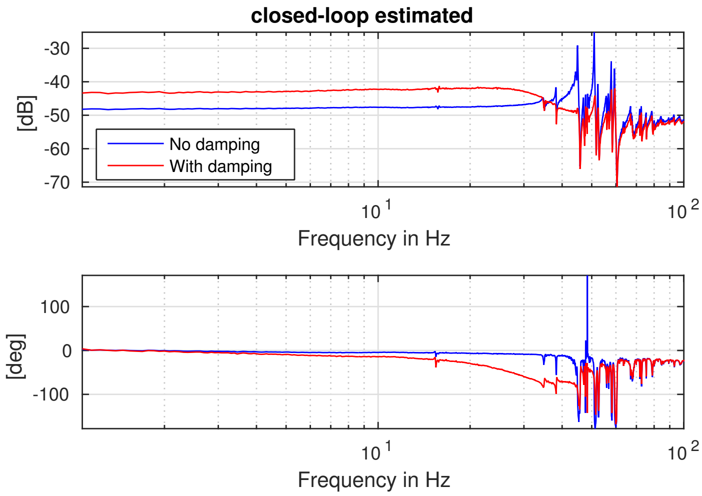

Figure 7. Estimated frequency response of the actuator with the active damping loop compared to the open-loop response

control parameters do not require a trial and error manipulation on the hardware setup, hence reducing the risk of wrong manipulation/implementation of control parameters.

iv) As it is observed from the responses in Figure 8 the static gain from the position loop controller output to the damping loop reference input needs to be scaled (parameter K_5 implemented in control scheme Figure 4). By doing so another advantage of the approach shines and becomes apparent: the damping loop also adds some roll-off in the response seen by the position loop controller (see Figure 9). The immediate observation comparing both responses is that the position loop controller can be designed much easier (simpler structure and less sensitive to knowledge of mechanical modes) without requiring notch filters. Consequently, if required a closed-loop with a larger bandwidth can be achieved (see Figure 9 right).

4.2 Actuator and segment motion due to vibration: damping loop effect

After implementing the active damping loop on each actuator under the segment subunit and verifying the stability and efficiency of the damping loop through frequency response measurements, the effects of the environment vibration on the motions was verified. The open-loop frequency response from actuator input commands to its encoder signal is already a good indicator for introduction of some level of damping to the system. However, the

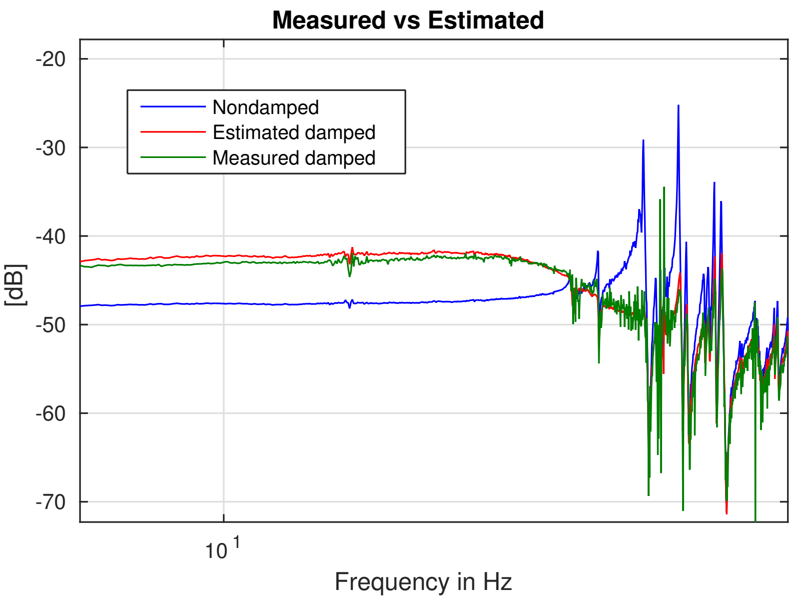

Figure 8. Measured vs. estimated frequency response of the actuator after closing the active damping loop

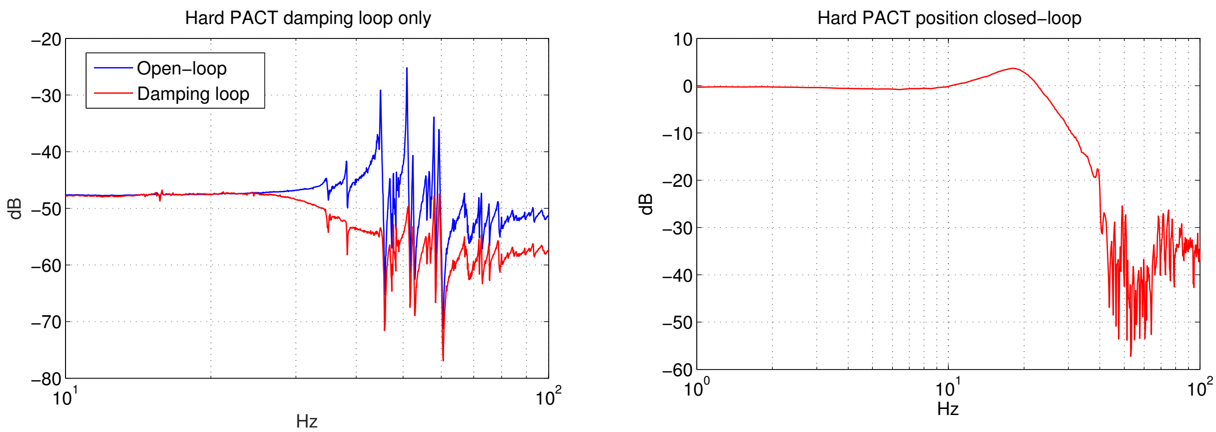

Figure 9. Open-loop vs. damping loop frequency response of the actuator including the static-gain scaling: Damping + roll-off (left), position closed-loop frequency response (right)

damping seen from the actuator command to the encoder measuring the relative elongation of the actuator does not necessarily imply the same level of damping at the segment level motion. Various measurement tests have been performed on the system to verify the efficiency of the damping loop also from the vibration transmission aspects.

The first test performed was reading the encoder of the actuators in presence of the environment vibrations (during this test an instrument with its cooling system and electronic cabinets was operational in the lab) once with active damping loop off and once turning it on. Figure 10 shows the Power Spectral Density (PSD) and the cumulative error value of the actuator internal encoder as a function of frequency for two cases of damping loop off and on.

To test the effect of the damping loop on the segment level, three accelerometers were installed on the top of the segment subunit equipped with the actuators with the damping loop. The accelerometers were placed just above the location of each actuator (see Figure 11). This way using the three acceleration signals piston, tip and

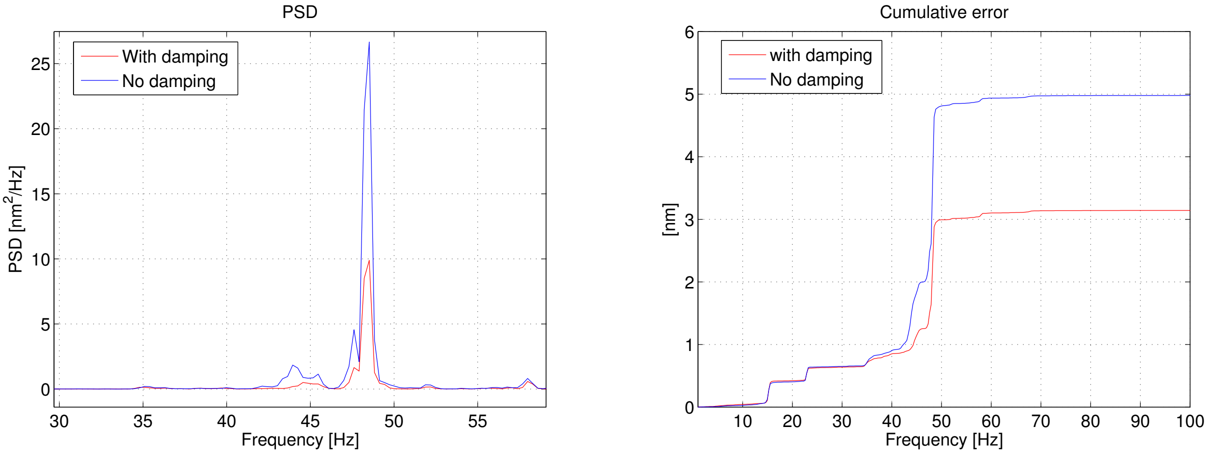

Figure 10. Damping loop: measured PSD (left) and the cumulative error (right) of the position actuator encoder (stand still) due to the lab vibrations

tilt motion of the segment could be reconstructed.

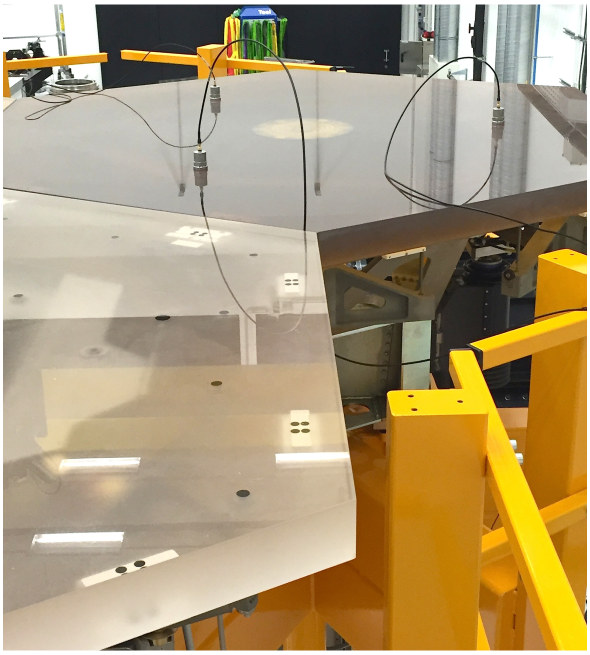

Figure 11. Accelerometers on the top of a segment in the M1 test facility with actuators using the active damping loop

During this measurement period there was no equipment (e.g. instrument with its cooling system) with potential vibration source present in the laboratory. It was decided to use the 'neighboring' segment actuators as a shaker. White noise excitations with a controlled voltage amplitude was generated, and introduced to the test stand. From the accelerometer reading, and filtering the signals the piston/tip and tilt motion of the segment were reconstructed and the PSD and the rms motion in different frequency intervals were estimated. The test was performed and repeated for two cases once with active damping loop off and once on. Figure 12 shows the segment tilt acceleration PSD and the estimated rms tilt motion in different frequency intervals. A reduction of segment motion by a factor two in the frequency intervals in the region of 40Hz to 50Hz can be observed.

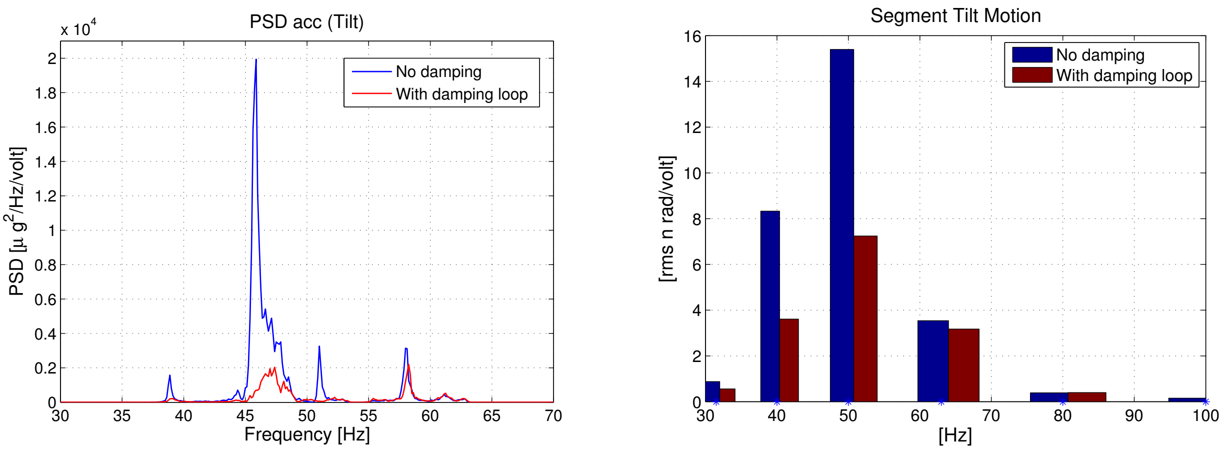

Figure 12. Damping loop: measured PSD (left) of the segment tilt acceleration, estimated segment rms tilt motion (right) for different frequency intervals due to a controlled excitation generated from the neighboring segment (white noise)

The results are in accordance with the vibration analysis and simulation performed for the primary segmented mirror,[6] where similar reduction and efficiency of damping in terms of performance is observed. Using the finite element model (FEM) of the telescope structure and mirror units including the M1 segments, the optical sensitivity and knowledge of the wavefront control correction capability, the transmission of vibration from various potential sources to the wavefront error can be estimated.

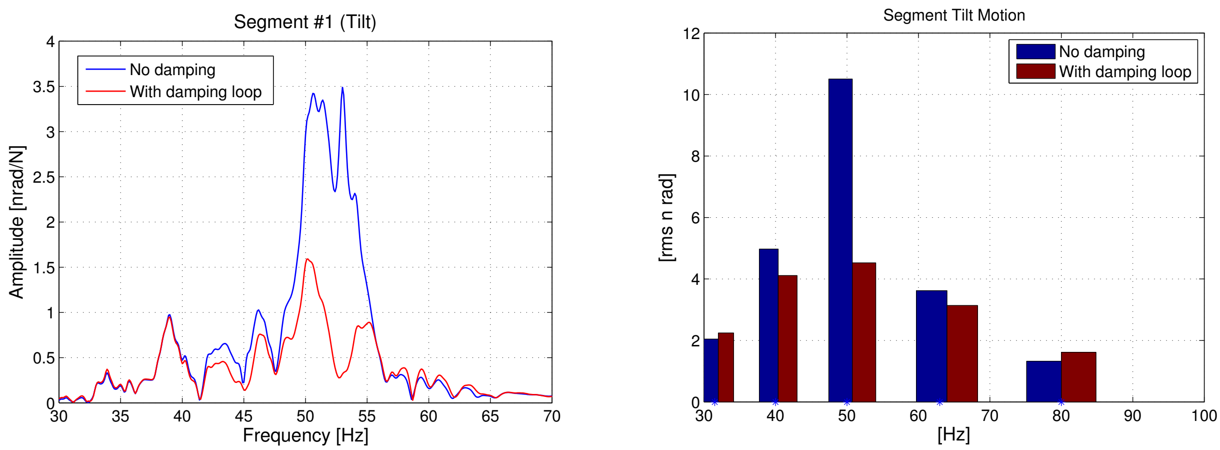

Figure 13. Damping loop: simulated transmission response to a tilt motion of a segment (left), estimated segment rms tilt motion (right) for different frequency intervals due to a controlled excitation generated from a source near the segment subunit (white noise)

The frequency responses from each vibration source input to the piston/tip and tilt (PTT) of each segment in closed loop (position actuator internal servo-loop/damping loop) are calculated. Figure 13 shows the sensitivity or transmission response from an excitation source (force in [N] in z-direction) near the segment subunits to the motion of a segment (tilt in [nrad]). In the same figure an estimate of tilt rms motion for different frequency intervals is depicted. The excitations are assumed to be white noise with spectral density of 1 $N/\sqrt{Hz}$, [0-100] Hz). The resulted values are compared for two cases of no damping and with the damping loop active, where using the active damping loop a reduction of the segment motion by a factor of two in the interested frequency range of 40-50 Hz is obtained.

5. CONCLUSIONS

Using positive position feedback for active damping of the hard PACT provides significant improvements at very low implementation cost. Design of a common robust position controller for all segments for all telescope elevations is much simpler due to better damped resonances and increase frequency response roll-off. No additional notch filters are needed anymore. Vibration transmission at resonance frequencies, which is seen as a drawback of hard PACT is significantly reduced. The damping controller design has only few parameters which can efficiently be tuned by a straight forward measurement-based tuning method. No additional sensor is needed for the damping loop.

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