VLT DSM, the control system of the largest deformable secondary mirror ever manufactured

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ABSTRACT

A proven technology for the shape control of large secondary deformable mirrors employs a magnetically levitated contact-less solution and relies on voice-coil actuators co-located to capacitive position sensors. The present work focuses on the description of the latest upgrade of this technology, as applied to the Very Large Telescope Deformable Secondary Mirror, the largest continuous facesheet adaptive mirror ever manufactured. The controller is based on a completely decentralized high frequency feedback coupled to a lower frequency improved feedforward. The system enhancements and performances are verified through electromechanical tests.

1. INTRODUCTION

The first operative AO system with voice-coil actuators co-located to capacitive position sensors was mounted on the Multiple Mirror Telescope (MMT)\textsuperscript{1} at the end of the 90s. Such a system is based on the first generation of the technology described here and features a 0.64 m shell diameter controlled at 336 points. A second generation realization, fully working since the beginning of 2012, is represented by the twin secondary adaptive mirrors mounted on the Large Binocular Telescope (LBT),\textsuperscript{2,3} which has 672 actuated points and a 0.91 m shell diameter. The LBT can be considered the state of the art for working adaptive optics applications, with performances never achieved before.\textsuperscript{4} A further second generation deformable mirror has been mounted on the Magellan Telescope (MAG).\textsuperscript{5} In the spring 2012 the final design and manufacturing of the European Southern Observatory (ESO) Very Large Telescope (VLT) secondary adaptive mirror was completed.\textsuperscript{6} The third generation VLT AO system features a secondary mirror with 1170 actuators and a 1.12 m diameter and is the largest adaptive mirror ever manufactured (see Fig. 1). New incoming realizations are envisaged for the Giant Magellan Telescope (GMT)\textsuperscript{7} and the European Extremely Large Telescope (E-ELT).\textsuperscript{8,9}

Magnetically levitated large adaptive mirrors are now a reality, fully proving their potentialities. However they represented and still embody an engineering challenge because of: their deformable structure with a dimension in the order of a meter, up to a thousand of actuated points, command rates up to 1 kHz, positioning accuracy within a few tens of nanometers. The contact-less solution provides several advantages: the reduction of assembly distortions, the possibility of mounting actuators with less stringent tolerances, the capability to operate the system with failed control points without any major performance degradation. Moreover, it allows to command relatively large average strokes, in the order of 100 $\mu$m, so that a single unit can carry out both chopping and high order corrections. Its main drawback is represented by ‘null-stiffness’ actuators, i.e. pure force actuators, so that all of the control points are coupled through the mirror stiffness. Moreover the shape control of large shells is affected by high modal densities, with many thousands resonances very close to each other, circa six per Hz, which are placed within both the controller and spillover bandwidth. A further critical point is the difficulty to accurately predict and describe the mirror dynamics because of the intrinsic multiphysics nature of the whole system. For example, the system performances are dependent on the controller gains, which in turn rely on the system passive damping to avoid instabilities associated to the control action. The damping, in turn, significantly depends on the thin air film entrapped between the mirror and its reference plane, whose properties are mainly determined by the air gap height, that changes because of the imposed shape control. An adequate modeling is required for both the fluid
dynamics and the structural damping but this is not an easy task since the modal damping is frequency dependent and control spillover affects resonances over a wide and modally dense frequency range. Another possible issue, which can affect system performances and effectiveness, is represented by the electromagnetic sensors-actuators cross-talk: the voice-coil electromagnetic fields can disturb sensor measurements, so deteriorating the positioning accuracy and possibly limiting system stability through the introduction of a feedback dependence between control forces and sensors noise/disturbance. This problem has been substantially eliminated by means of proper electromagnetic and electrostatic design of the actuator. Other practical problems, like the entrapping of dust between the shell and reference body and the system survival against extreme environmental conditions (wind, earthquake) have been solved by means of appropriate devices, applied and verified to the already deployed units.

The stringent system requirements, along with the above cited critical issues, make the controller design of large adaptive mirrors highly challenging. At the very beginning, the coupled nature of the problem suggested the use of centralized control systems. Nevertheless, the time needed to acquire, condition and process a large number of signals, together with the high computational power required to implement a centralized controller, hindered the adoption of such a solution. Moreover, even with centralized controllers, the system stability constraint prevents the possibility of achieving the commanded positions within the required bandwidth. The key difficulties of this kind of control problem are the high modal density which characterizes large shells and the impossibility of applying an adequate attenuation to a significant portion of the modes across and beyond the control bandwidth, because of the well-known relation between signal attenuation and phase lag and the lower effectiveness of the fluid dynamics damping on the highest modes.

After several attempts, a satisfactory solution has been implemented on the MMT secondary mirror. It is based on a two degrees of freedom controller based on a static feedforward (FF) combined with a completely decentralized proportional-derivative (PD) feedback (FB).\textsuperscript{1} Acknowledging the stiffness dominated nature of the control problem, the FF is designed to provide the static forces needed to deform the shell to a required steady shape. Such a term relies on an in the field identification of the shell stiffness matrix condensed at the actuation points, whose accuracy is a key element in granting a precise positioning. The computation of the FF contribution requires a complete centralization, but is carried out at a relatively low command frequency (250-1000 Hz), so becomes feasible even if it introduces a moderate phase lag in the optical control loop. On the contrary the FB contribution is digitally realized at a far higher frequency (40-100 kHz),
so to allow the addition of a sizable electronic damping of the modal resonances, within both the command bandwidth (up to 250-1000 Hz) and the spillover frequency region. The FB is also useful for improving the tracking capabilities and the rejection of external disturbances, e.g., wind and gravity effects, while the authority of its proportional gain is limited to the low stiffness/frequency modes. Such a control strategy demonstrated to be reliable and effective, so that it has been left substantially unchanged up to the VLT secondary deformable mirror. Meanwhile possible improvements for both the FF and FB paths have been suggested in the literature by several authors, e.g., the possibility of exploiting a distributed implementation of the static FF\textsuperscript{10,11} or the modification of the static FF term computation to obtain at the same time an efficient centralized integral action,\textsuperscript{12,13} along with a FF completion through the introduction of a system dynamics compensation.\textsuperscript{12,14,15} Within the area of distributed controls some applications proposed for large deformable mirrors can be found in.\textsuperscript{16–21} Nevertheless, despite the new technology potentialities available with respect to the first MMT implementation, a complete centralization of the FB remains quite cumbersome to apply, because of the intervened significant increase of controlled points.

The quite long experience acquired with thin shells deformable mirrors clearly demonstrated the leading role of the FF contribution in determining system performances. In practice the system is controlled mainly through FF, while the FB is required to condition the behavior of lightly damped modal shapes, which can be excited during the FF force application. Moreover secondary mirrors upgrades through different technology generations have always been undertaken by following the so called KISS principle,\textsuperscript{22} i.e. ‘Keep It Simple, Stupid’. This is because, once the design specs are satisfied, system reliability has always been favored with respect to pure performances, i.e., referencing the KISS’s author once more: “If it works do not fix it”. On the basis of the two just mentioned considerations it is easy to understand why, once the decision to upgrade the VLT mirror control strategy was taken, the attention has been focused on FF improvements, choosing their most easy and reliable implementation. In practice the new VLT controller exploits the same features of its predecessors with the addition of a dynamic FF term, designed on the basis of the proposal of,\textsuperscript{12} with a few further improvements. The result is a specialization to the problem at hand of the PD control with FF system dynamics compensation developed in the robotic field during the eighties.\textsuperscript{23,24}

### 2. VLT DSM SYSTEM DESCRIPTION

The VLT adaptive secondary has been designed to replace the current telescope secondary unit, so that the optical requirements and the mechanical interface remain unchanged. The adaptive mirror features a 1.12 m diameter and it is aspheric convex with a 4.55 m radius curvature. The shell is supported through a membrane placed at its central hole with a 96 mm diameter. It constraints the in-plane mirror displacements, while introducing a negligible out-plane stiffness. The 2 mm thick continuous facesheet Zerodur mirror is controlled through 1170 actuated points. The contact-less technology is based on voice-coil motors, with the coils placed at the top of 190 mm long actuator fingers applying their forces to the mirror through an electromagnetic interaction with permanent magnets glued on the rear mirror face. The actuators bottom side is rigidly connected to the cold plate, see Fig. 1, which acts both as the support and heat sink for the coils. Capacitive position sensors are exploited to measure the distance between the reference plane and the rear mirror face, so to well approximate a co-location with the actuators. The system controller is implemented through the use of an on-board unit, featuring 78 dedicated digital signal processors (DSP), which guarantee an overall computational power of 152 GMACs/s. Every DSP board handles 16 coils by means of high efficiency switching drives. A dedicated hexapod linked to the cold plate allows a fine support positioning and switching between different focal stations.

The VLT secondary implements a third technology generation, which ensures several improvements. The cold plate is connected directly to the hexapod, to obtain a higher mechanical stiffness, the Zerodur reference body has been highly light-weighted to reduce its mass and further improve the mechanical stability. The switching voice-coil drivers can halve the required power, so limiting energy consumption and heat dissipation needs at the same time. The computational power is increased and the all electronic hardware made more compact.

### 3. CONTROL STRATEGY

On the basis of the wavefront sensor measurements the optimal mirror shape generator sends a step command to the deformable mirror, with a maximum command frequency of 1 kHz. The \((n_a \times 1)\) step command vector, where \(n_a\) is the
number of actuation points, is then shaped in time through a polynomial function which guarantees a $C^1$ acceleration, $f_{sh}(t)$, so that the reference position signal to be tracked at the $(k+1)$ command step can be written as

$$d^*(t) = d'_{(k)} + (d'_{(k+1)} - d'_{(k)})f_{sh}(t).$$  \(1\)

In this way every step command is split into an initial time varying part, followed by a steady reference position.

The $(n_a \times 1)$ control forces vector is the sum of a FF and a FB contribution

$$f^* = f_{ff}^* + f_{fb}^*.$$  \(2\)

The $(n_a \times 1)$ FB force vector is a completely decentralized PD control realized through a digital implementation at a relatively high sampling frequency, of 61 kHz.

$$f_{fb}^* = [\text{\$G_{d,\$}}]p + [\text{\$G_{p,\$}}](d^* - p),$$  \(3\)

where $[\text{\$G_{d,\$}}]$ and $[\text{\$G_{p,\$}}]$ are $(n_a \times n_a)$ diagonal matrices representing respectively the derivative and the proportional gains, while $p$ and $p$ are $(n_a \times 1)$ vectors of the control points position and velocity. The position is acquired directly from the capacitive sensors, while the velocity is estimated through pseudo-derivative filters.

The FF term can be further split into two different contributions

$$f_{ff}^* = f_{ff_1}^* + f_{ff_2}^*,$$  \(4\)

where $f_{ff_1}^*$ is the $(n_a \times 1)$ static FF vector and $f_{ff_2}^*$ the $(n_a \times 1)$ dynamic FF vector. The static FF provides the static forces to deform the mirror and requires a completely centralized computation, which is performed at the command frequency of up to 1 kHz. The static FF force is shaped in time within each command step, in the same way as for the reference signal, usually through a slower function, $f_{sh}(t)$, to help limiting the excitation of high frequency dynamics. The force applied at the $(k+1)$ command step can be written as

$$f_{ff_1}^*(t) = f_{ff_1}^*(k) + K^*(d'_{(k+1)} - s'_{(k)})f_{sh}^*(t),$$  \(5\)

where $K^*$ is the $(n_a \times n_a)$ identified static FF matrix, \(^{1,12}\) i.e. a close relative of the true mirror stiffness matrix condensed at the actuation points, see\(^{25,26}\) for a discussion of the differences between the identified and the true mirror stiffness matrix.

The above presented terms are already implemented on all the working adaptive secondary mirrors based on the here considered technology. The last control upgrade on the VLT system introduces the $(n_a \times 1)$ dynamic FF vector

$$f_{ff_2}^* = M^*d^*(t) + C^*d'(t)$$

$$= M^*(d'_{(k+1)} - d'_{(k)})f_{sh}^*(t) + C^*(d'_{(k+1)} - d'_{(k)})f_{sh}^*(t),$$  \(6\)

whose contribution is based on the $(n_a \times n_a)$ matrices $M^*$ and $C^*$, which ideally represent the mirror mass and damping matrices condensed at the actuation points. These matrices could be identified as for $K^*$,\(^{14}\) but the work\(^{12}\) suggests the possibility of achieving satisfactory performances by approximating both of them as scalar diagonal matrices

$$f_{ff_2}^* = [\text{\$m^*\$}]d^* + [\text{\$c^*\$}]d',$$  \(7\)

where the scalar values $m^*$ and $c^*$ can be easily tuned through an in the field optimization of the system responses as, for example, proposed in\(^{27}\). A possible starting value for $m^*$ is represented by the mass of the mirror divided by the number of controlled points, i.e. a lumped approximation. With respect to\(^{12}\) the dynamic FF scheme implemented for the VLT has been improved by introducing a scheduling of the scalar value $c^*$ as a function of the air gap between the mirror controlled points and the reference plane. This improvement is required because the fluid dynamic damping is highly affected by the air gap height and an improved dynamics compensation requires a scheduled damping FF. So there is an optimal scalar value $c^*$, which is a function of the air gap, retrieved directly through the commanded position, $c^* = c^*)(d')$. In this way the matrix $[\text{\$c^*\$}]$ remains diagonal but no more scalar, because each of its diagonal elements is a function of the corresponding commanded position. The parameter scheduling as function of $d'$ has been tuned directly on the system by modifying experimentally the value at different gaps in order to maintain the same dynamic behavior. Then, the obtained values have been fitted as polynomial function of the gap, thus allowing an effective implementation of the adaptive scheduler in the real time controller.
4. EXPERIMENTAL SETUP AND ELECTROMECHANICAL TESTS

VLT experimental tests have been wholly carried out by exploiting the dedicated electronics developed for the control of the secondary adaptive mirror. The decentralized FB and the FF contributions are implemented through a hard-real time code, partly on the DSPs (see Sec. 2) and partly through the use of field-programmable gate array (FPGA). The D/A converters and current drivers for the voice-coil motors are mounted on the very same DSPs control boards.

The conditioning of the capacitive position sensors is handled by dedicated boards placed on the actuators fingers as well. In this way the distance from their pickup points is minimized, so reducing their signals noise while improving its integrity. The sensors bandwidth is about 26 kHz, with a noise of 3-4 nm RMS. The initial sensors calibration is based on piston motions of the shell and mechanical references, with the assumption of a uniform force distribution among all the actuators, so that any significant elastic deformation of the mirror can be avoided.\(^6\) The related calibration accuracy is in the order of 10 nm over the full motion range of 120 \(\mu\)m. This calibration process will be eventually improved by a similar procedure, but using optical metrology. A key point of the experimental setup is represented by the environmental stability of the controlled system, which is highly correlated to the stability of the capacitive position sensors. Typical sensor thermal drifts are in the range of 3 nm/\(\degree\)C, while the humidity drift achieves a value of about 15 nm/(g m\(^{-3}\)).

The actuators have a bandwidth of about 25 kHz. The actual force of each actuator is not acquired directly, but can be estimated accurately by knowing the driven coil current and the motor current/force constant.

The electronics itself represents a powerful tool to perform static and dynamic experimental tests. The synchronous excitation of all the actuators and acquisition from all co-located sensors is allowed by dedicated hardware and software features, so that it is possible to test the system capability to track predefined time histories, with a maximum command rate equal to the digital control frequency (61 kHz). In this way measurements and commands can be compared directly and typical performance estimators of the control system, like single actuator step responses, modal step responses or modal transfer functions can be obtained without the need of any external data acquisition system.

Here the modal tests are shortly summarized to highlight the improvements obtained through the introduction of the dynamic FF term. The commanded modal shapes are computed as the right singular vectors of the identified static FF matrix \(K^*\), ordered for increasing singular values, i.e. stiffness. The system requirements ask for a settling time to 90\% of modal step commands within 1 ms. The performance has to be checked in presence of a mirror tilt, up to \(\pm 12\) arcsec, as required for field stabilization. Fig. 2 shows the 100th mirror modal shape; the corresponding modal time step responses, superimposed to a mirror tilt of 12 arcsec, are reported in Fig. 3. Figure 3 compares two modal step responses obtained with different control schemes, the first one based only on the static FF (standard FF), the second one exploiting the dynamic FF.
contribution. Both the responses fulfill the system requirements in terms of settling time; however, the response obtained with the standard control requires a smoother shaped command to limit the overshoot within 10% of the modal step amplitude, so that the resulting settling time remains acceptable but clearly higher than that achievable with the dynamic FF contribution. In practice, the dynamic FF allows to limit response overshoots, so that the same overshoot obtained with standard FF is achieved despite a faster raising command, i.e. the dynamic FF improves the system settling time.

The aforementioned comment applies to all of the modal shapes. A summary of the system response behavior with respect to all the 1170 modal shapes obtained from the static FF matrix can be appreciated in Figs. 4 and 5. These figures clearly show that the dynamic FF allows to obtain the same responses overshoots with an improved settling time, within all the considered frequency range. Figs. 4 and 5 confirm that both the control strategies guarantee the system specification fulfillment, with an overshoot within 10% of modal amplitude and a settling time below 1 ms. Fig. 4 also suggests that low order modes are almost critically damped, mainly because of the fluid dynamics contribution, so that their overshoot is limited. The modes up to about the 100th show an increasingly overshoot because the damping is decreasing and they belongs to a frequency range which is still excited by the control action. Moving to higher order modes the system response approaches a static behavior and is dominated by static FF forces. The appreciable overshoot increment of the very high order modes is an artifact due to the small modal step amplitude, which becomes close to sensors noise.

5. FINAL REMARKS

The experimental tests confirm the importance of the dynamic FF in improving the deformable mirror transient behavior, which is expected to provide better overall optical performances eventually. The upgraded controller allows to obtain faster raising times to reference step commands, while limiting overshoots, so that an improved settling time is obtained. This result has been achieved with marginal modifications with respect to the previous controllers and with a negligible added computational cost. In this way a step toward improved performances has been accomplished without any risk to compromise the system reliability gained through a 15 years work.
Figure 4. Modal overshoot.

Figure 5. Modal settling time.
REFERENCES


