**Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008**

Wind Evaluation Breadboard Control Architecture, Dynamic Model and Performance

Teodora Viera*, Pablo Zuluaga*, Marcos Reyes*, Miguel Núñez*, Javier Castro**

**Instituto de Astrofísica de Canarias, Canary Islands, Spain
(Tel: +34-922-605-200; e-mail: tvc@iac.es).
**Gran Telescopio de Canarias, Canary Islands, Spain
(Tel: +34-922-315-139; e-mail: javier.castro.lopez@gtc.iac.es).*

Abstract: The WEB (Wind Evaluation Breadboard) for the European ELT (Extremely Large Telescope) Design Study is a primary mirror and telescope simulator formed by seven segments, including position sensors, electromechanical support systems and support structures. The purpose of the WEB is to study the effects of wind on the control of the positions of the segments. This paper describes the control architecture, the dynamic model generated based on the Finite Element Model and the performances achieved in simulations.

1.INTRODUCTION

The Extremely Large Telescope (ELT) Design Study (DS) is a technology development programme undertaken under the European Commission (EC) Sixth Framework Programme (FP6) by institutes and companies in Europe, Israel and Australia. The DS covers the development of enabling technologies and concepts required for the design and construction of the European ELT in the optical and infrared range, with a diameter in the order of 40m.

The WEB (Wind Evaluation Breadboard) is one of the main ELT DS work packages, with the participation of IAC, MEDIA, JUPASA and ESO. The WEB is a simulator of the primary mirror of the large telescope. It contains 7 hexagonal panels, 3 position actuators per panel, 24 edge sensors, 18 accelerometers and all the hardware necessary to control and test responses to dynamic loads (wind) and pseudo-static loads (gravity and temperature variations). The high rigidity structures and mechanisms are designed to allow wind tests at different inclinations and orientations.

The purpose of the WEB is to study the effects of wind on the real time control of the positions of the segments. The segment simulators, from now on panels, will simulate two types of optical materials, the heavy type that corresponds to Zerodur (414 kg each one) and the light-weight type (131 kg each one), which corresponds to SiC. The WEB, of around 7 meter diameter, 2.6 meter high, and 20 ton weight will eventually be exposed to wind flow on a representative astronomical observatory site (IAC Teide Observatory, Canary Islands, Spain), in order to ascertain the performance of the panel supports and control systems in relation to wind excitation, with the aim of verifying that high spatial and temporal frequency wind disturbances can be controlled to acceptable accuracy.

1.1 The WEB Mechanical Design

The WEB is equipped with an azimuth axis to rotate the panels to different angles of incidence with respect to the wind, and with an elevation axis from 0º to 60º of inclination. The principal subsystems are (Fig. 1):

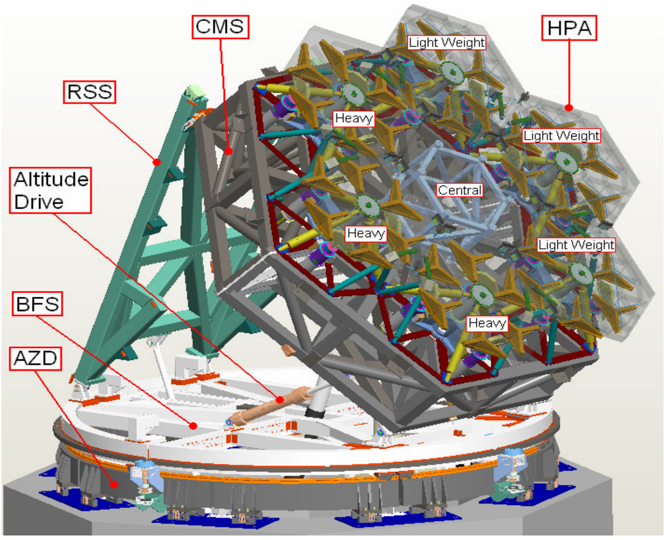

Fig.1. The WEB Mechanical Design

Azimuth Drive (AZD): This system is the base structure fixed to the ground. It consists of welded plates, and the rails separated into eight circular sectors.

Base Frame Structure (BFS): This structure is assembled on the AZD rails to allow the azimuth rotation. It is used to support the Altitude Drive, the Rear Support Structure (RSS) and the Cell Mechanical Structure (CMS).

Cell Mechanical Structure (CMS): This is a high stiffness structure that supports the hexagonal panel assembly (HPA).

978-1-1234-7890-2/08/$20.00 © 2008 IFAC

10.3182/20080706-5-KR-1001.0119

Altitude Drive and Rear Support Structure (RSS): Elevation of the CMS is carried out by a hydraulic system. The AZD, BFS, CMS and RSS form the telescope simulator, designed by MEDIA.

Hexagonal panel assembly (HPA): These are the segment simulators mounted onto the CMS, designed by IAC. They are composed of one central fixed panel, and six active panels (three heavy and three light-weights). The central panel is supported by a truss and is used as a reference. Each active panel is supported isostatically by three mechanical systems. The axial support is a whiffle tree. The X,Y displacements are restricted by a membrane (Lateral Support) and the Z rotations are restricted by the Torsional Constrainer (Fig. 2).

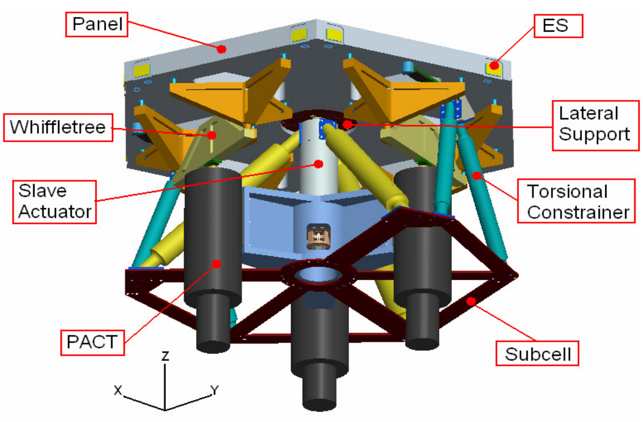

Fig.2. The Hexagonal Panel Assembly.

1.2 The WEB Electronic Hardware Design

The WEB Electronic Hardware is mainly composed of:

18 Position Actuators (PA). Manufactured by TNO (The Netherlands). These are installed supporting the whiffle trees, to control the panel active Degrees Of Freedoms (DOFs), which are the X,Y rotations (panel tip and tilt) and the Z displacements (panel piston). They are a two stage system. The coarse stage takes care of the required actuator stroke of 30mm with μm accuracy. The second stage is a voice coil actuator without friction and hysteresis that bridges the gap to nm accuracy with a translation capacity of about 10 μm. The PA has been designed as a “soft” force actuator with an axial stiffness that will give the (Zerodur) mirror segment a suspension mode of less than 10 Hz in the axial direction.

Two different types of sensors: 24 Edge Sensors and 18 Accelerometer Sensors. The Edge Sensors (ES), manufactured by FOGALE (France), are inductive sensors formed by two plates located on the side of the hexagonal panels facing each other. These sensors measure the difference in position between two adjacent panels in the Z axis (perpendicular to the segment surface). The Accelerometers Sensors (AS), Si-Flex SF1500S, manufactured by Colibrys, are located on the rear face of the panels (in the projection of the PA over the panel), to measure its acceleration.

Two independent processing units. The WEB Real Time processor and The PA Control Platform computer. The WEB Real Time processor is a PXI-8187RT, a manufactured by National Instruments. The PA Control Platform computer is based on the Innovative Integration SBC6713e Digital Signal Processor. Two DSPs are used, controlling nine PAs each. Each DSP is connected to the WEB Ethernet network in order to communicate with the WEB Real Time processor.

1.3 Global Coordinate System of HPA.

A natural way to parameterize the orientation of the mirror is by defining a set of body coordinate axes located at the centre of gravity of the mirror. The in-plane translations along the x-axis and y-axis and rotation about the z-axis are restrained by the mirror segment parallel guidance. The out-of-plane degrees of freedom will be controlled by the local segment control system. These degrees of freedom are called piston, tip and tilt, for translation along the z-axis, rotation around the y-axis and rotation around the x-axis respectively.

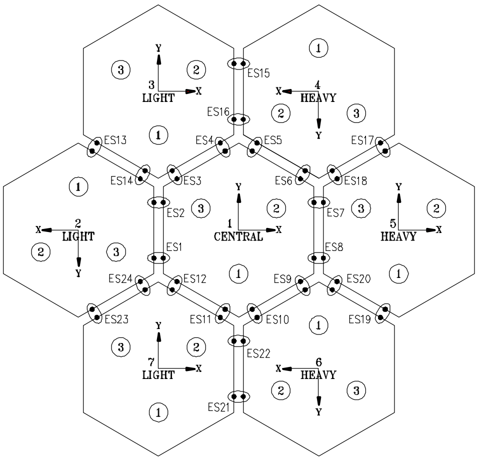

Fig.3. Global Coordinate System of HPA.

The election of this global coordinate system was imposed by the transformation matrix between the piston,tip,tilt panel movements and the forces applied on the PAs of each panel. This way the transformation matrix is equal for all the panels.

Fig. 3 shows the distribution of the panels (heavy & light-weight panels), the positions and numbers of the ES, the projection of the PAs over the panels (circles 1,2,3) and the positions of the accelerometer sensors (circles 1,2,3).

2.DYNAMIC MODEL

The control system and the control algorithms used, depend heavily on the dynamic behaviour of the mechanical system. For that reason this dynamic behaviour, obtained at this stage by modal analysis using a (FEM), will be discussed first. The FEM was generated as a free vibration analysis in Ansys 10.0. The model has 82677 elements and 220028 nodes. Lineal beams were used for CMS, lateral HPA and torsion supports. Springs were used for joints between AZD and BFS and for PAs. Masses were used for PA voice coils and subcells. Shells were used for BFS, AZD, Panels, PAs and Whiffletrees.

2.1 *General Concept*

The matrix of the transfer functions between inputs and outputs of the system can be obtained by modal superposition [1][2], which can be expressed as:

$$\mathbf{H}(s) = \sum_{i=1}^{n} \mathbf{v}_i \mathbf{u}_i^T \frac{1}{s^2 + 2\xi_i \omega_i s + \omega_i^2} \qquad (1)$$

where

H is the matrix of the transfer functions between inputs and outputs of the system.
$\mathbf{u}_i$ is the column vector of displacements in the inputs into the system in mode '*i*'.
$\mathbf{v}_i$ is the column vector of displacements in the outputs from the system in mode '*i*'.
ω_i is the angular frequency of mode '*i*'.
ξ_i is the damping ratio of mode '*i*'.
n is the number of modes considered in the model.

In order for equation (1) to be valid, the modal displacements must be normalized to unit modal mass.

As an alternative to the matrix of the transfer functions, the behaviour of the system can also be expressed via a space state model, as follows:

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{s} \quad \mathbf{y} = \mathbf{C}\mathbf{x}$$

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\Omega^2 & -2\Xi\Omega \end{bmatrix} \quad \mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{U} \end{bmatrix} \quad \mathbf{C} = \begin{bmatrix} \mathbf{V} & \mathbf{0} \end{bmatrix} \qquad (2)$$

$$\mathbf{U} = \{\mathbf{u}_i\} \quad \mathbf{V} = \{\mathbf{v}_i\} \qquad \text{where}$$

Ω is a diagonal matrix which includes the angular frequencies of the modes.
Ξ is a diagonal matrix which includes the damping factors of the modes.
U is a matrix which brings together the modal displacements of the inputs for the different modes.
V is a matrix which brings together the modal displacements of the outputs for the different modes.

The matrix of the transfer functions (**H**) has been used to design the controllers quickly and the state space model has been used primarily to verify the stability of the closed system looking at the closed loops poles.

The inputs which have been selected for the system are the forces applied on each panel by the PAs and by the wind. The action of each PA is applied as two opposing forces acting on two nodes of the model, which correspond to the interfaces of the PA with the whiffletree and with the central support structure (Fig. 2). For this reason, in equation (1) and (2) the difference between the modal displacements of those two points is used.

The force of the wind is modelled as punctual forces applied to the surface of the panel at the three points of intersection with the axes of the PAs.

The outputs considered are the displacement of the Edge Sensors, obtained as the difference between the points of the panels where they are attached, and the displacements of the panels' surface where the accelerometers are fixed. The accelerometers have not been explicitly modelled, having used the position instead of acceleration at those points.

2.2 *WEB Case*

All modes of the model up to 500 Hz have been analysed, amounting to 906 modes. Fig. 4 shows the transfer function between a PA and the corresponding position of its accelerometer for one heavy panel (green) and one light panel (blue). Panels of the same type have very similar behaviour.

The fundamental modes of the system are the 18 modes associated with the piston, tip and tilt vibration of the panels on the 'soft' stiffness of the PAs. These modes have frequencies of between 8 and 11 Hz for the heavy panels and between 13 and 18 Hz for the light weight panels. There are additional significant modes, local to the panels, starting at 70Hz. Global modes coming from the telescope simulator are at 22, 31 and 46Hz, however their effects are very small thanks to the decoupling of the PAs.

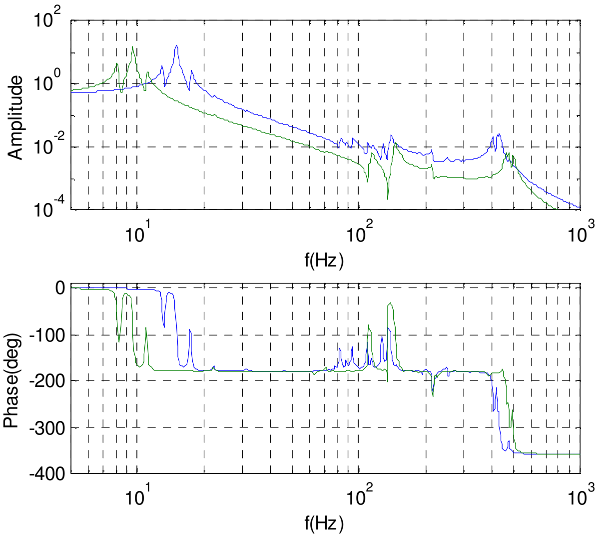

Fig.4. Bode plot of the transfer function between a PA and the corresponding accelerometer position for one light panel (blue) and one heavy panel (green).

The concept of the 'soft' actuators comes from concept of having the panels floating in the space with no mechanical link with the cell structure and controlled by the forces introduced with the force actuators. In this condition, the panels are independent. The purpose of the cell structure is to absorb the reactions of the force actuators. In practice it is necessary to link the panels to the cell with a 'soft' stiffness. For frequencies above the natural vibration frequency of the mass of the panels acting on the 'soft' stiffness the concept of decoupled panels is maintained, being coupled for frequency below these modes.

2.3 *Model reduction*

In order to simplify the dynamic model, the number of modes has been reduced. The criteria selected for reduction was to eliminate those modes with a contribution to the global transfer function, acting in the direction of the mode and lower than a certain threshold.

$$\text{Retain mode if } h_j(s) > \text{threshold} \quad H_j(s)$$

$$h_j(s) = \mathbf{v}_j^T \mathbf{v}_j \mathbf{u}_j^T \mathbf{u}_j \frac{1}{s^2 + 2\xi_i \omega_i s + \omega_i^2}$$

$$H_j(s) = \sum_{i=1}^{n} \mathbf{v}_j^T \mathbf{v}_i \mathbf{u}_i^T \mathbf{u}_j \frac{1}{s^2 + 2\xi_i \omega_i s + \omega_i^2}$$

Using a threshold of 0.5 the model has been reduced to 387 of the 906 initial modes without affecting the dynamical behaviour.

3. CONTROL ARCHITECTURE

Fig. 5 shows the control architecture. There are two function blocks: the Panels Global Controller and the Panel Local Controller. This architecture uses “soft” position actuators which decouple the motion of the panels from the cell and other panels. This means that the control of each panel can be treated independently in a local panel controller, if the panel positions are also measured independently. The position of the panels can be estimated from the readings of the ESs by a matrix multiplication. This function is not local to the panel controller since it is necessary to know all the ES readings to estimate the position of each panel. The measurement of the accelerometers is by nature local to the panel and they provide us high bandwidth measurements without computation cost.

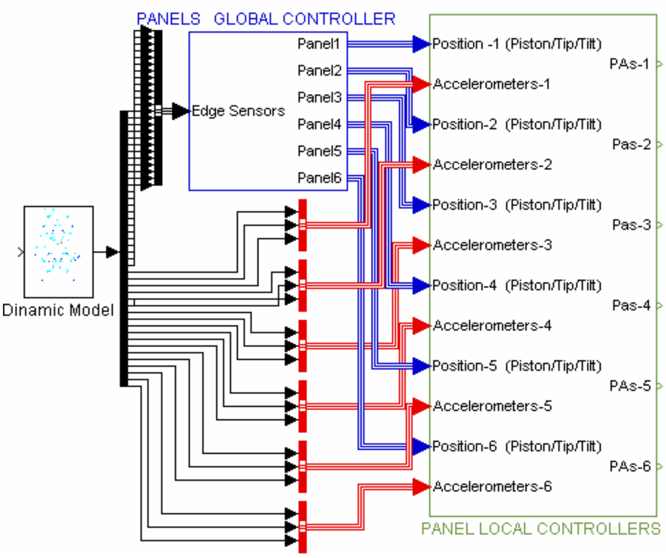

Fig. 5. The Control Architecture.

These two functions will be distributed in the hardware architecture in a natural way: The Panels Global Controller is calculated by The WEB Real Time processor, which transmits the panel position to each local controller. The Panel Local Controllers are calculated by the PA Control Platform.

3.1 *The Panels Global Controller*

The Panels Global Controller is formed by a matrix $\mathbf{B}_{18x24}$ (which relates the reading of the ESs to the movements of the panels), 18 PI controllers (each movement of each panel has its own PI controller) and 18 low pass filters (see Fig.6). ES readings have a delay of 1026 µseconds and they are filtered by Bessel low pass filters (cut-off frequency of 200 Hz), these facts are considered in simulations.

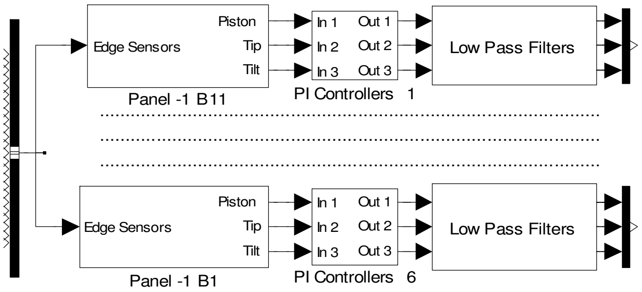

Fig. 6. The Panels Global Controller Architecture.

3.2 *Panel Local Controller*

Each Panel has its own Panel Local Controller, which is explain in the next section. The bandwidth of the PA Amplifier is 2 kHz and it is considered in simulations.

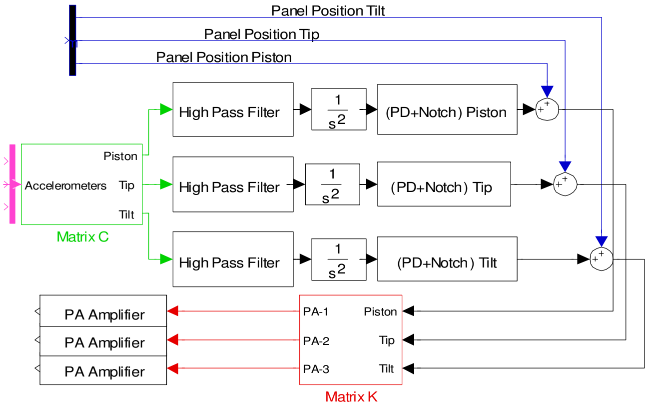

Fig. 7. The Panel Local Controller

4.RESULTS

4.1 *Tuning of Panel Local Controller*

The Panel Local Controller consists of three PD+Notchs controllers working in the piston, tip and tilt modes of the panel (Fig.7). The reading from the three accelerometers are projected to these modes by means of a matrix **C**, and the control forces produced by the three controllers are projected on the actuator forces by means of a matrix **K**. Figures 8 and 9 shows the outputs of the Matrix C, based of our dynamic model, when the PAs are excited to provide each mode to the light and heavy panels. Figure 10 shows the crosstalk between modes for the light panel. Piston and tip are coupled at low frequencies due to the asymmetry of the panel support (due to the design); however coupling disappears at frequencies where panel inertia is dominant.

For each mode, a PD controller enhanced with notch filters to avoid the destabilizing effect of several modes has been independently tuned. One notch was introduced for the tip-tilt modes, and two for the piston modes. Once all 18 controllers are working together the system maintains stability, but the slight coupling requires a gain reduction of 2.24db to get the same gain margin as in the original controllers. Figure 11 shows the attenuation transfer function to perturbations in each mode as measured by the accelerometers. For theses figures the high-pass filter used in this controller was fixed to a low frequency to see the behaviour of the controller.

The attenuation of the Local Panel Controllers at very low frequencies is not possible in practice. These controllers use feedback from accelerometers and the panel position has to be estimated by double integration. The first problem is that the acceleration noise integrated to low frequencies gives an inadmissible position noise. The second problem is that any bias error of the acceleration signal will also be integrated to infinite. These are the reasons why the Panel Local Controllers include a high-pass filter.

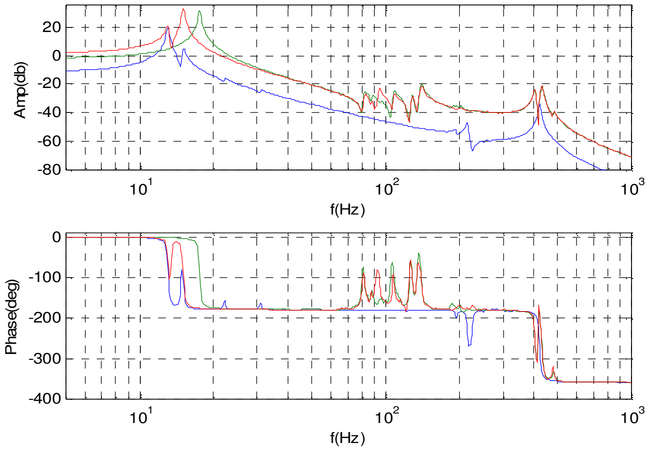

Fig. 8.Plotting of the light panel response for piston excitation (blue), tip excitation (red) and tilt excitation (green).

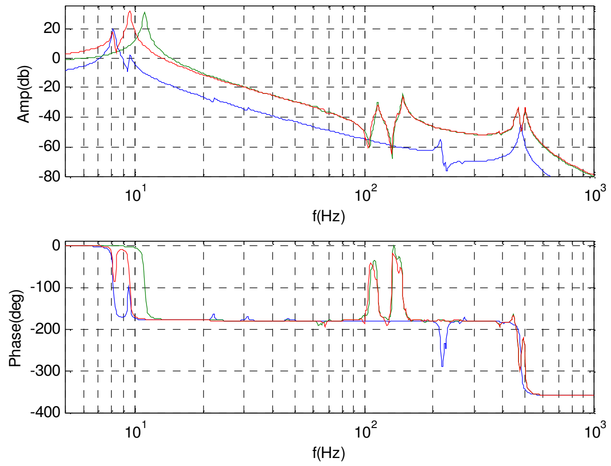

Fig. 9 Plotting of the heavy panel response for piston excitation (blue), tip excitation (red) and tilt excitation (green).

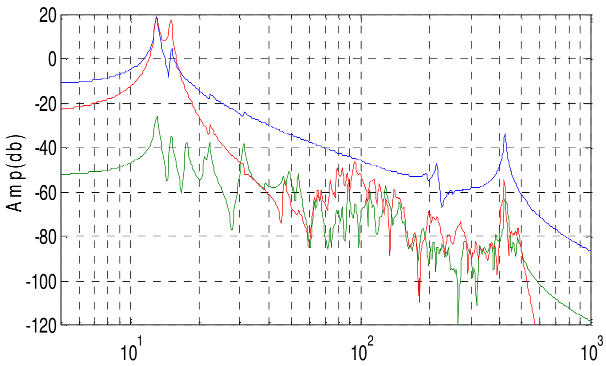

Fig. 10. Amplitude of the transfer functions of piston (blue), tip (red) and tilt (green) to piston excitation.

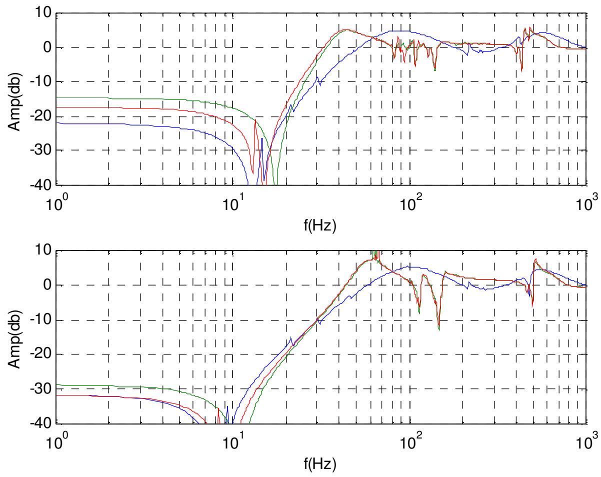

Fig. 11. Attenuation of each mode using Panel Local Controller alone. Piston (blue), tip(red) and tilt (green). Light panel above, heavy panel below.

4.2 *Tuning of Panels Global Controller*

Estimation of the 18 panel positions (piston, tip and tilt for 6 panels) is performed by multiplying the vector formed by the 24 measurement by an estimation matrix $\mathbf{B}_{18x24}$. This matrix can be obtained from the matrix $\mathbf{A}_{24x18}$ which relates ES signals to the position of the panels, by least squared method. However, the matrix $\mathbf{A}_{24x18}$ is singular, due to the geometric disposition of the ES on the panels. When the six external panels rotate simultaneously around the edge common to the central panel, the ESs does not have any relative motion and the measurement is zero. For this reason matrix **B** is obtained from $\mathbf{A}_{24x18}$ using a singular value decomposition (SVD). Other telescope such as Keck [1] [2] and GTC [3] [4] uses a different disposition of sensors to avoid this singular mode.

The attenuation of perturbation at low frequencies is the responsibility of the control loop closed using the ESs. The first attempt was to introduce a proportional gain with the edge sensors compensating a reduction of the proportional gain in the Panel Local Controllers; however this approach destabilized the system. The reason is the presence of local modes of the panels which affect the edge sensors, located at the panel edges, but which have no significant influence on the accelerometers, located in the middle of the panel.

The solution was to introduce a low pass filter in the control action of the ESs. This filter is complementary to the high-pass filter used in the panel controller. Using this approach it has been possible to introduce integral and proportional gain to the position measurement performed by the ES. Figure 12 shows the attenuation of perturbations as measured by the accelerometers, having used a cut-off frequency for the low and high pass filters of 5Hz.

It can be seen that piston and tilt have a limited attenuation at low frequencies, and they do not show the 20dB/decade slope expected due to the integral gain. The wind perturbation deform the panels in addition to move them, given that the Global Controller correct the motion of the panels as measured by the Edge Sensors, the panel motion as measured by the accelerometers will have the effect of the panel deformation. Figure 13 shows the attenuation of some of the edge sensors. In this case a low frequency attenuation is obtained.

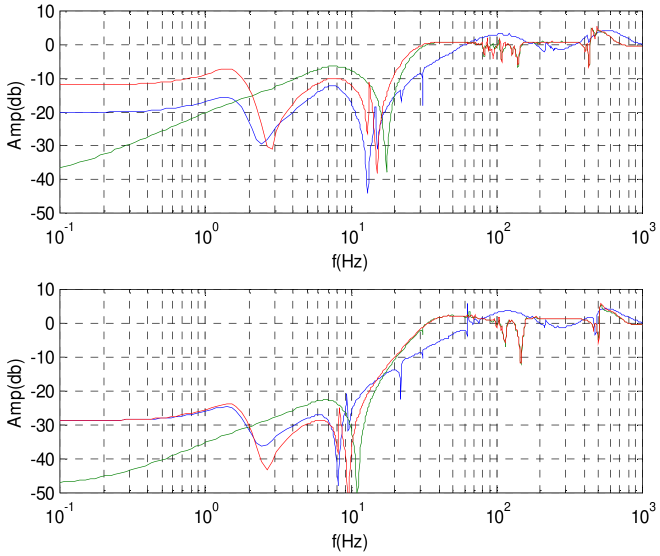

Fig. 12. Attenuation of perturbations using the complete control scheme. Piston (blue), tip(red) and tilt (green). Light panel above, heavy panel below.

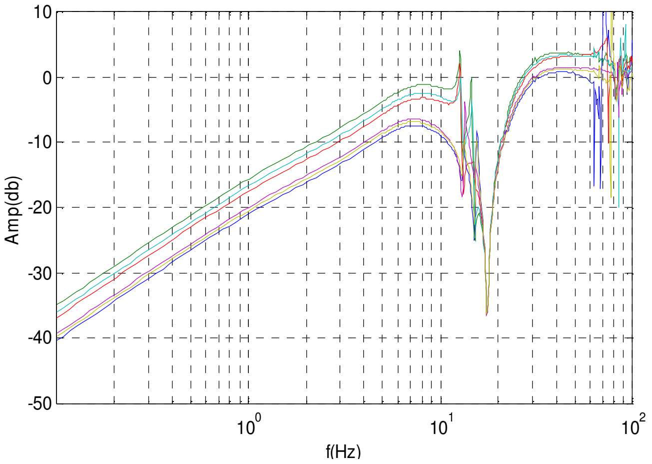

Fig. 13. Attenuation function of the six sensors connected to panel 1 (light) for a tilt excitation on the panel.

5.CONCLUSION

The main conclusion of the work performed to date is that the approach of using ‘soft’ position actuators to decouple the behaviour of the panels permits to tune the controller of each panel independently without having to use a MIMO system with a large number of inputs and outputs. The use of accelerometers to sense panel position, independently of the motion of the cell structure, is also part of this decoupling strategy.

Global controller using ESs, which needs to process simultaneously all sensors to estimate the panel positions, can be performed at low frequency (tens of hertz). This is not critical for the WEB, using only 24 ESs to control 18 PAs. However, in the case of an ELT using thousands of PAs and ESs this supposes a significant reduction of computational power required, which we consider an advantage even taking into account that currently there are techniques to broach this global controller in the range of kilohertz.

The decoupling concept is valid having cell modes with natural frequencies larger than the natural modes of the panels on the ‘soft’ stiffness, which is the case of WEB. In the case of an ELT the structure will have very low frequency modes. These modes will coupled the segments, but in a frequency region well within the closed loop bandwidth of the local controllers. It is necessary to check if this coupled destabilise the system. In that case, the coupling can be treated at the level of the global controller, where all the segments are treated simultaneously.

6.ACKNOWLEDGEMENTS

This activity is supported by the European Community (Framework Programme 6, ELT Design Study, contract No 011863).We would like to acknowledge all MEDIA staff involved in the WEB design, as well as ESO ELT DS Project Office.

7.REFERENCES

[1] Jared, et al, “The W.M. Keck Telescope segmented mirror active control system”, Advanced Technology Optical Telescopes IV, Lawrence D. Barr, Editor, Proc. SPIE 1236, pp 996-1008, 1990.

[2] R.W. Cohen, T.S. Mast, J.E. Nelson, “Performance of the W.M. Keck Telescope active mirror control system”, Advanced Technology Optical Telescopes V, Larry M. Stepp, Editor, Proc. SPIE 2199, pp 105-116, 1994.

[3] J.M. Casalta, J. Ariño, M. Canchado, E. M. Geijo, A. Molins, A. Tomàs,L. Zago. “The Performances of GTC Secondary Mirror Drive Unit”. Proc.of SPIE’s 2004 Astronomical Telescopes and Instrumentation, Vol. 5495, pp 507-517

[4] E. García, L. Zago, D. Gallieni “Special and innovative aspects of the GTC M2 Drive mechanism”, Proc.of SPIE’s 2002 Astronomical Telescopes and Instrumentation, Vol. 4837