LBT Primary Mirror Cover
FEA of as built configuration

Technical note
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4 Scope of work
The aim of this note is to report on some FE verifications about the Mirror Cover performances from the structural stand point. In particular we will focus on the identification of weak points for a possible allocation of stiffening devices (locking devices). All the analyses cover the cases related to a retracted position of the covering blades. For this condition, the two extreme positions of fully deployed and retracted spider are considered. When possible, the results of the FEA will be cross checked with the experimental results obtained during the test campaign at LBTO, according to what described in LBT Doc.No. 727S156 issue A, dated 14 June 2006.

5 The model
The FE model has been developed in ANSYS by using the as built drawings. It has been entirely modeled by Shell93 bi-quadratic elements having 8 nodes per side. Some other details have been modeled by Solid95 bi-quadratic brick elements, MASS21 elements and LINK8 elements where appropriate.

The mass of the FE model resulted 2569 Kg, that seems reasonable with respect to the 2880 Kg measured value since all the mass of the weldings, the mechanical details of the deployment mechanism on the T structure and other minor features have been neglected.

The mass of the deployable blades parts has been split and lumped on the two sides of the T structure, 66% percent of this deployable mass has been put close to the big boxed beam (where the wheels normally rest in parking condition) while 34% of the mass is supposed resting on the edge of the central disk as per the following pictures.

This splitting has been estimated on the basis of the center of gravity of the sector covered by the blades in parking condition (+/- 22.5 deg sector). The mass density on this sector of blades has been assumed constant over the covered area (the effect of blades tapering is neglected in terms of this mass splitting).
Figure 2. The FE model, with the final mesh. The mass of the model matches the measured value with a good level of representativeness.

The following simplifications and/or assumptions have been set in the analyses:

1. The main bearing on the rotation shaft is modeled as a couple of steel rings having the same geometry of the actual bearing but rigidly connected for piston, tip, tilt and de-centers. In other words only the rigid rotational motion about the bearing axis IS ALLOWED, but the stiffness of the bearing contact itself, along the different directions is neglected. The FE modeling of this parts has been carried out by using suitable DOFs couplings in a local cylindrical coordinate system, at the relevant nodes (see the green points in the previous picture, close to the bearing area).

2. The basement of the pedestal is supposed to by clamped to a perfectly rigid telescope (at C-ring level) in 8 points only (all DOFs removed here). This 8 points are located at level of the eight stiffening ribs in the basement (see cyan and yellow arrows at the bottom of the model in the previous picture).

3. The mass of the deployable part (880 Kg measured) has been split in 530 Kg on the internal circular plate (six lumped masses in cyan) and 350 Kg on the internal disk (three lumped mass in cyan). See picture.

4. One locking device has been modeled with its actual stiffness (solid model) and depending on the particular load case it is supposed to be disengaged.
AND/OR perfectly engaged into its hole. No other intermediate situations are considered such as the ones deriving from a partial play between the pin and its hole.

5. The main actuation screw is modeled as a rod (LINK8) working in its direction only (depending on structure position). Only the axial stiffness related to the screw and its engaged length is considered, the effect of the trust bearings in the reducer, the screw-nut contact stiffness and other not mentioned features are neglected for lack of information. Considered the length of the screw, this assumption seems reasonable.

Figure 3. Detail of the FE model from the rear. The eight attachment points at the base of the pedestal are visible, as well as all the green couplings between the relevant DOFs at the bearing location. The actuation screw is the light green line on the center-left.
6 FE static analyses
The results of the different FEAs will be summarized in the subsections hereafter reported with a minimum level of description and a few pictures per case. Other pictures will be collected in the final appendix for more information concerning the deformed and modal shapes.

- Telescope Zenith pointing, parked, stow pin engaged / disengaged:
The maximum displacement along the “optical” axis is about 26.7 mm, this has been verified with both linear and non linear analysis in large deformations. The displacement comes from the combination of the tilt of the bearing basement, coupled to a deflection/torsion of the rear boxed beam plus a deflection of the main protruding beam. I would say the none of these effects is dominating on the other. In this particular case the effect of the anti-rotation locking device is nearly null. The stress level in the structure, just to mention it, is well under control and is anyway concentrated only at the eight interface points to the C ring structure.

<table>
<thead>
<tr>
<th>Case</th>
<th>Deflection in gravity vector direction [mm]</th>
<th>Measured value @ LBTO [mm]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engaged</td>
<td>26.7</td>
<td>32</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
<tr>
<td>Disengaged</td>
<td>26.7</td>
<td>32</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
</tbody>
</table>

Table 1.
Figure 4. Zenith pointing, parked, stow pin engaged: Z axis displacements in [m]. Max value is 26.7 mm (blue).

Figure 5. Horizon pointing, parked: detail of Z axis displacements on the rotating shaft [m]. In this particular case the stow pin isn’t engaged but the difference is negligible. Tilt on the basement under the bearing is not negligible a causes a tilt of the rear boxed beam.
Trying to anticipate some consideration of the effect of a stiffening to be done at the free end of the rear boxed beam, it seems that by locking (mainly) the Z displacements at such a location one could have a benefit (see next).

- **Telescope Zenith pointing, deployed position, stow pin engaged additional support wheel under T beam**

In this load condition a further support has been added in form a simple vertical reaction at the root of the protruding beam as indicated. The reaction has been placed just under one of the vertical reinforcements.

![Figure 6. The effect of the vertical reaction (wheel) at the base of the protruding beam.](image)

The deflection of the tip in this condition is about 16.7 mm Vs. 25.1 mm in the same condition (plot not reported) but without this extra support.

The value of the vertical reaction at such location is pretty high and is about 28000 N but without any issue of local stress if the junction is properly studied.

A similar reaction effect could be obtained, as suggested, by a simple wheel attached under the bottom of the beam and sliding on a suitable circular track fixed to the floor.
This type of constraint is not bilateral, but we will assume it for the purpose of the following modal analysis showing the first two modes.

Figure 7. 1\textsuperscript{st} mode at 2.9 Hz.

Figure 8. 2\textsuperscript{nd} mode at 4.6 Hz.
Telescope Horizon pointing, parked position, stow pin engaged / disengaged / total play on actuation screw:
In this condition the deflection is dominated by a rigid motion about the bearing axis that is not properly limited neither by the actuation screw nor by the stow pin.

<table>
<thead>
<tr>
<th>Case</th>
<th>Deflection in gravity vector direction [mm]</th>
<th>Measured value @ LBTO [mm]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stow Pin Engaged</td>
<td>29.1</td>
<td>60 mm</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
<tr>
<td>Stow pin Disengaged</td>
<td>53.8</td>
<td>60 mm</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
<tr>
<td>Only stow pin engaged, effect of activation jack neglected</td>
<td>42.5</td>
<td>60 mm</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 2.

The value above reported would suggest that, if the stow pin was engaged during the tests at LBT, it is not working properly in the sense that there are plays at level of the pin/hole coupling or maybe the guiding cylinder where the pin is supposed to be driven is over sized with respect to the pin itself. More likely it will be a mix of several factor, including effects on the activation screw.

Please note that we consider also a different situation in these analyses, that is the one without the contribution in stiffness provided by the activation jack (screw). In this case the deformation is 42 mm, suggesting that if working properly, the stow pin only provides more rigidity with respect to the condition of jack screw only.

In the idea of improving this particular load condition, the additional locking device should be located along the Y direction of the ANSYS coordinate system. This improvements should be very effective if made in a pre-loaded fashion (see next).
We computed the deflection in the "optical axis" direction as well and it resulted 1.9 mm max. Test value is not available.

See next pictures.
Figure 9. Horizon pointing, parked, stow pin engaged: X axis displacements in [m]. Max value is 29.1 mm (red).

Figure 10. Horizon pointing, parked, stow pin removed: X axis displacements in [m]. Max value is 53.8 mm (red).
Figure 11. Horizon pointing, parked, stow pin removed: Z axis displacements in [m] (optical axis direction). Max abs value is 1.9 mm (blue).
- **Telescope Horizon pointing, arm deployed position, covering blades retracted, stow pin engaged / disengaged / total play on actuation screw:**

In this case the FE model has been rebuilt in a rotated layout according the description above. The static deflection under gravity had been computed for three cases are hereafter resumed.

The maximum displacements along X (direction of gravity) is reported, it is located at the free end of the rear boxed beam.

<table>
<thead>
<tr>
<th>Case</th>
<th>Deflection in gravity vector direction [mm]</th>
<th>Measured value @ LBTO [mm]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stow Pin Engaged plus actuation jack</td>
<td>13</td>
<td>56 mm</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
<tr>
<td>Only stow pin engaged</td>
<td>17</td>
<td>56 mm</td>
<td>Not sure pin was engaged or not during tests</td>
</tr>
<tr>
<td>Only activation jack resisting</td>
<td>58</td>
<td>56 mm</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 3.

![Figure 12. X direction gravitational displacements [m].](image-url)
Figure 13.

Figure 14.
Figure 15.
7 FE modal analysis
The modal shapes and frequencies have been computed for the parked position only (with and without stow pin installed). The Block Lanczos algorithm has been selected and the first ten modes extracted.

Results are the following:

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Freq.[Hz] Pin Engaged</th>
<th>Freq.[Hz] Pin Removed</th>
<th>Test result @ LBTO [Hz]</th>
<th>Description of the FE mode shape possible / improved by....</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.21</td>
<td>2.27</td>
<td>N.A.</td>
<td>In plane rotation about the bearing axis. To limit in-plane motion of the free-side of the boxed beam</td>
<td>Mentioned as very low resonant frequency in LBTO doc.</td>
</tr>
<tr>
<td>2</td>
<td>3.87</td>
<td>3.83</td>
<td>3.3</td>
<td>Z piston mode + boxed beam lat. Bending. To limit vertical motion of the free-side of the boxed beam</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>10.38</td>
<td>10.01</td>
<td>N.A.</td>
<td>Description of the FE mode shape possible / improved by....</td>
<td>Notes</td>
</tr>
<tr>
<td>4</td>
<td>11.49</td>
<td>10.98</td>
<td>N.A.</td>
<td>Description of the FE mode shape possible / improved by....</td>
<td>Notes</td>
</tr>
<tr>
<td>5</td>
<td>20.10</td>
<td>20.10</td>
<td>N.A.</td>
<td>Description of the FE mode shape possible / improved by....</td>
<td>Notes</td>
</tr>
<tr>
<td>6</td>
<td>26.69</td>
<td>26.65</td>
<td>N.A.</td>
<td>Description of the FE mode shape possible / improved by....</td>
<td>Notes</td>
</tr>
<tr>
<td>7</td>
<td>35.42</td>
<td>35.28</td>
<td>N.A.</td>
<td>Complex, local + disk modes</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>51.14</td>
<td>50.90</td>
<td>N.A.</td>
<td>Complex, local + disk modes</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>58.71</td>
<td>58.76</td>
<td>N.A.</td>
<td>Complex, local + disk modes</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>71.62</td>
<td>71.52</td>
<td>N.A.</td>
<td>Complex, local + disk modes</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.
Figure 16. First Modal shape: stow pin engaged: In plane rotation about the bearing axis. 3.2 Hz.

Figure 17. Second Modal shape: stow pin engaged: In plane rotation about the bearing axis. 3.8 Hz.
Figure 18. Third Modal shape: stow pin engaged: In plane rotation about the bearing axis. 10 Hz.

Figure 19. 4th Modal shape: stow pin engaged: In plane rotation about the bearing axis. 11.5 Hz.
Figure 20. 5\textsuperscript{th} Modal shape: stow pin engaged: In plane rotation about the bearing axis. 35.4 Hz.

Figure 21. 6\textsuperscript{th} Modal shape: stow pin engaged: In plane rotation about the bearing axis. 26.6 Hz.
Figure 22. 8\textsuperscript{th} Modal shape: stow pin engaged: In plane rotation about the bearing axis. 51 Hz.
8 FE modal analysis with countermeasures

We tried to explore the effectiveness of possible countermeasures to limit the low resonant frequencies in the piston and lateral rotation modes in particular. This together with the large deflection of the structure at horizon pointing (when the arm is deployed) seem to be the most serious issues.

From the first point of view (natural frequencies in parking) we studied the suggested possibility to stiffen first the free-end of the boxed beam by a couple of jacks (locking devices) and then to add another stiffening device acting at the tip of the protruding beam:

**Rear Locking devices**

These locking devices are TBD but shall able to pull the structure against the upper C-ring as per picture hereafter.

Figure 23. The two locking rods located at the end of the boxed beam. Each jack is 40 mm in diameter and 500 mm long.

The two additional jacks are modeled as pure rod elements having the cross section of a 40 mm diameter circular area (steel). These jacks are assumed to be as long as 500 mm (maybe to much considering the available volume but the shorter the better).
We run again the modal analysis and compared the results to the baseline situation. In both the cases hereafter summarized we assumed the main stow-pin engaged as well.

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Baseline stow pin engaged [Hz]</th>
<th>Additional locking devices (two) [Hz]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.21</td>
<td>6.75</td>
<td>Rotational modes: NOTE that the order of two first modes in the enhanced model are reversed to compare directly the shape.</td>
</tr>
<tr>
<td>2</td>
<td>3.87</td>
<td>4.10</td>
<td>Piston modes: NOTE that the order of two first modes in the enhanced model are reversed to compare directly the shape.</td>
</tr>
<tr>
<td>3</td>
<td>10.38</td>
<td>19.88</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>11.49</td>
<td>25.46</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>20.10</td>
<td>31.28</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>26.69</td>
<td>35.53</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>35.42</td>
<td>48.60</td>
<td>-</td>
</tr>
<tr>
<td>8</td>
<td>51.14</td>
<td>61.06</td>
<td>-</td>
</tr>
<tr>
<td>9</td>
<td>58.71</td>
<td>74.94</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>71.62</td>
<td>77.01</td>
<td>-</td>
</tr>
</tbody>
</table>

Figure 24. Comparison of frequencies with and without the additional locking rods.

The conclusion is that the rotational mode, in the plane of the mirror cover, can be stiffen to twice its frequency in this ideal FE conditions.

The other locking device, working in a vertical direction (optical axis direction) provides a limited benefit, it helps keeping the boxed beam stable in terms of displacements but this still has a sort of torsion along its axis and its free tip still rotates since the locking devices work as rods.

Moreover, even by locking more efficiently the torsion at the tip of the boxed beam, there is still the issue of the deflection of the tapered beam protruding to the center. This effect doesn't seem to be very prone to be cured by any kind of "rear" stiffening.

Hereafter the new modes shapes are reported, for the first three modes only.
Figure 25. First resonant mode for the “stiffened” layout. 4.1 Hz.

Figure 26. Second resonant mode for the “stiffened” layout. 6.75 Hz.
Figure 27. Third resonant mode for the “stiffened” layout. 19.9 Hz.

Front Locking device
These locking device are still TBD but shall able to pull the keep the protruding beam of the swing arm stable in the vertical and horizontal direction thanks to the V-shaped layout. It could be fixed to the big boxed beam of the wind-bracing structure, just under the catwalk around the cell. The feasibility of such a solution is still to be assessed.

The layout is the one reported in the following picture.
Figure 28. The model of the Swing arm with the front locking device only. No rear lockings.

The results of the modal analysis are reported in the following table and the shape of the first modes in the next pages.

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Baseline stow pin engaged [HZ]</th>
<th>Additional frontal locking device (triangle) [Hz]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.21</td>
<td>9.10</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>3.87</td>
<td>10.40</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>10.38</td>
<td>11.50</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>11.49</td>
<td>20.33</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>20.10</td>
<td>22.65</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>26.69</td>
<td>31.45</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>35.42</td>
<td>51.13</td>
<td>-</td>
</tr>
<tr>
<td>8</td>
<td>51.14</td>
<td>55.23</td>
<td>-</td>
</tr>
<tr>
<td>9</td>
<td>58.71</td>
<td>66.44</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>71.62</td>
<td>72.94</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5. Comparison for the first ten modal shapes. The lowest resonant frequency move from 3.21 to 9.1 Hz.
Figure 29. First resonant mode for the “V locking stiffened” layout. 9.10 Hz.

Figure 30. Second resonant mode for the “V locking stiffened” layout. 10.40 Hz.
Figure 31. Third resonant mode for the “V locking stiffened” layout. 11.50 Hz.

Figure 32. Fourth resonant mode for the “V locking stiffened” layout. 20.33 Hz.
Boxed beam locking in deployed position

Another case that we studied in order to understand possible countermeasures to be implemented on the system is sketched in the following picture.

Figure 33. The model in the deployed position with extra locking devices at the tip of the boxed beam.

In principle this approach could help in increasing the natural frequencies of the system when is deployed.

By running a modal analysis we obtained the following results, compared to the baseline case, without any stiffening.

In the following we report also some pictures of the resonant modes.
<table>
<thead>
<tr>
<th>Mode #</th>
<th>Baseline stow pin engaged [Hz]</th>
<th>Lockings in deployed condition [Hz]</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3.21</td>
<td>5.88</td>
<td>Modal shapes are different and not directly comparable... see pictures</td>
</tr>
<tr>
<td>2</td>
<td>3.87</td>
<td>6.79</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>10.38</td>
<td>19.76</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>11.49</td>
<td>25.61</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>20.10</td>
<td>34.63</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>26.69</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>35.42</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>8</td>
<td>51.14</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>9</td>
<td>58.71</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>71.62</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 6. Modes with stiffeners in the deployed position

Figure 34. 1st mode with stiffeners in the deployed position
Figure 35. 1\textsuperscript{st} mode with stiffeners in the deployed position

Figure 36. 1\textsuperscript{st} mode with stiffeners in the deployed position
Figure 37. 1st mode with stiffeners in the deployed position

Figure 38. 1st mode with stiffeners in the deployed position
9 Conclusions

We have modeled and analyzed the M1 cover as-built configuration. Such model resulted being within 10% of the real system measured mass. We studied both static load cases and modal analysis of the cover in its parked and deployed configuration (umbrella not deployed). The results still reasonably match the measured performances depending on the assumptions made on the stow pin effectiveness and deployment mechanism play. We repeated also the modal analysis after adding two more restraints that could be implemented on the telescope, but the improvement on the lowest modes is only partially successful. In particular cover arm piston mode is not stiffened at all by such restraints.

As a further step we added a V shaped locking device on close to the tip of the main beam and this provided some benefit in terms of resonant frequencies but can be implemented only in parked position when the tip of the beam is accessible.

Finally, with the goal of improving the performance of the arm when it is in the deployed condition, we evaluated the effect of a locking to be applied at the end of the boxed beam once the swing arm is just in front of the cell. Apart from the feasibility of this solution that is not covered by this note, the improvement in terms of natural frequencies is about a factor 2X, with the first piston mode of the protruding beam at about 5.8 Hz.