EUROPEAN SOUTHERN OBSERVATORY

Organisation Européenne pour des Recherches Astronomiques dans l'Hémisphère Austral
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ESO - EUROPEAN SOUTHERN OBSERVATORY
Karl-Schwarzschild-Straße 2 - D-85748 Garching b. München

Technical Note

DATE: 14 March 2001

FROM: F. Koch

TO: F. Rigaud

COPY: R. Arsenau, N. Hubin

SUBJECT: **MACAO-VLTI Tip-Tilt-Mirror FEM Analysis**

Introduction

The scope of this Analysis is to investigate the resonance frequencies of the Gimbal Mount design of the MACAO Tip-Tilt-Mirror (TTM). In case of low frequencies (< 300 Hz), possible design improvements will be analysed.

FE Model

Based on the IGES files provided by the contractor, a 3D FE Model has been built with ANSYS 5.6 consisting of about 25000 Elements and 41000 Nodes. Half of the FE Model is shown in Figure 1.

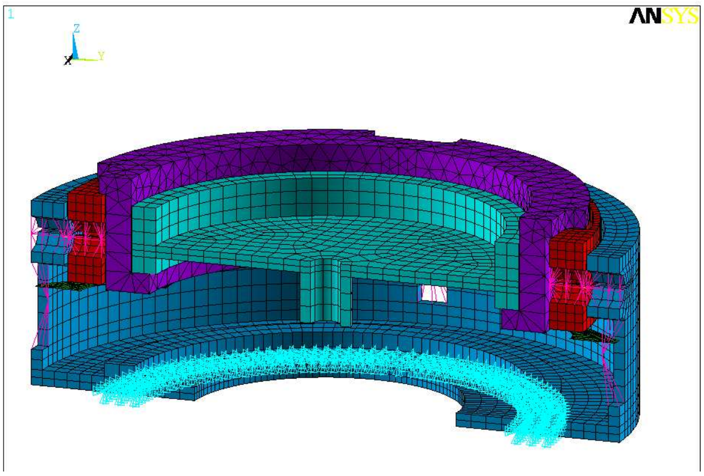

Figure 1: FE Model of TTM Gimbal Mount (half model).

The three ring structures (inner, intermediate and outer) as well as the Dummy Mirror are modelled with 3D Solid elements. The four Sensor target Supports are represented by 3D Shell elements. Each of the four flexible pivots is simulated by a set of stiff 3D Beam elements and an appropriate Beam element that simulates the stiffness of the flex pivot (5020-400) (provided by the contractor):

- Torsional stiffness: 12 Nm/rad
- Radial stiffness: 5.7 10^6 N/m
- Axial stiffness: 11 10^6 N/m

The following material properties for FORTAL Aluminium have been used:

- Youngs modulus: 70000 MPa
- Poisson's ratio: 0.34
- Mass density: 2770 kg/m^3

The nodes interfacing to the Gimbal Mount support structure are constrained in translational x, y and z-direction.

A modal analysis has been carried out to analyse the natural frequencies of the system. Furthermore, a harmonic response spectrum analysis has been performed to simulate the influence of the linear actuator forces to the Transfer Functions seen by the position sensors.

Results of Reference Analysis

The table below summarises some analyses options, the total mass, the centre of gravity, the mass moments of inertia, the first 30 eigenfrequencies of the Gimbal Mount system as well as their corresponding effective masses.

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  ANALYSIS TYPE . . . . . . . . . . . . . . . . . .MODAL
     EXTRACTION METHOD. . . . . . . . . . . . . . .BLOCK LANCZOS
  NUMBER OF MODES TO EXTRACT. . . . . . . . . .    30
  TOTAL MASS =  0.50367E-02 ton
                        MOM. OF INERTIA        MOM. OF INERTIA
    CENTROID             ABOUT ORIGIN          ABOUT CENTROID
                              [ton*mm^2]             [ton*mm^2]
 XC = -0.17877E-09      IXX =    35.55         IXX =    34.32
 YC = -0.17877E-09      IYY =    35.98         IYY =    34.74
 ZC =  -15.669          IZZ =    60.42         IZZ =    60.42

   24436  ELEMENTS       40638  NODES          131952   DOFs
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Mode	Frequency [Hz]	MX	MY	MZ	IXX	IYY	IZZ
		Effective Mass and Inertia in [%] of Total					
1	0.007	0.1	0.0	0.0	0.0	33.4	0.0
2	0.009	0.0	0.1	0.0	20.5	0.0	0.0
3	135.3	0.0	0.0	46.9	0.0	0.0	0.0
4	167.2	0.0	0.0	0.0	0.0	0.0	0.0
5	167.2	0.0	0.0	0.0	0.0	0.0	0.0
6	167.2	0.0	0.0	0.0	0.0	0.0	0.0
7	167.2	0.0	0.0	0.0	0.0	0.0	0.0
8	260.9	1.4	28.7	0.0	0.0	0.0	12.3
9	263.7	5.3	18.8	0.0	0.0	0.0	16.7
10	268.6	42.9	0.3	0.0	0.0	0.0	4.3
11	732.6	0.0	0.0	1.7	0.0	0.0	0.0
12	804.6	0.0	0.5	0.0	17.6	0.0	0.4
13	806.5	0.0	0.0	0.0	2.9	0.0	2.5
14	957.9	0.0	4.3	0.0	0.1	0.0	0.0
15	960.9	0.6	0.0	0.0	0.0	0.1	0.0
16	1003.6	0.0	0.0	0.0	0.0	0.0	0.0
17	1013.7	0.0	0.0	0.4	0.0	0.0	0.0
18	1020.2	2.2	0.0	0.0	0.0	0.3	0.0
19	1074.7	0.0	0.0	0.0	0.0	0.0	0.3
20	1094.9	0.0	0.0	0.0	0.0	0.0	2.2
21	1202.3	0.0	0.0	4.6	0.0	0.0	0.0
22	1345.7	0.0	0.0	2.4	0.0	0.0	0.0
23	1353.7	0.0	0.0	0.0	0.0	0.0	0.0
24	1481.4	8.2	0.0	0.0	0.0	23.3	0.0
25	1561.9	0.0	8.5	0.0	15.3	0.0	0.0
26	1586.6	0.0	0.0	0.0	0.0	0.0	0.0
27	1592.4	0.0	0.0	0.0	0.0	0.0	0.0
28	1602.4	0.0	0.0	0.0	0.0	0.0	0.0
29	1670.1	0.0	0.0	0.4	0.0	0.0	0.0
30	1712.5	0.0	0.0	0.2	0.0	0.0	0.0
Cumulative [%]:		60.7	61.3	56.7	56.5	57.1	38.7

The table shows, that the first 10 eigenfrequencies are below 300 Hz. Mode 1 and 2 at 0.007 and 0.009 Hz are, as expected, the rigid body tilt modes about the y- and x- axis respectively. Their frequencies depend on the inertia of the movable parts and on the torsional spring rate of the flex pivots.

The mode shape of the 3rd mode at 135 Hz is shown in Figure 2. This mode is a flexible mode of the intermediate ring and has an astigmatic-like shape. Unfortunately, this mode shows also a piston mode of the mirror and will be monitored by the sensors. 47 % of the total mass in vertical z-direction is involved in this mode.

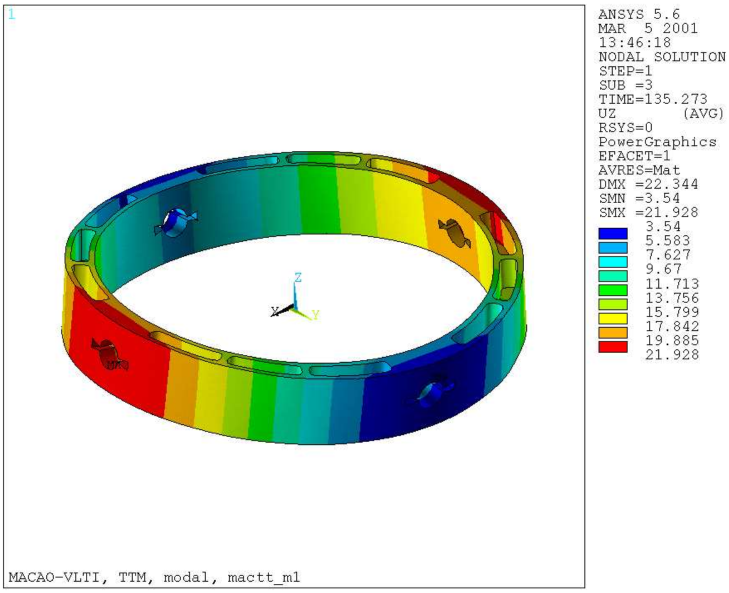

Figure 2: Mode 3 at 135 Hz.

Modes No. 4, 5, 6 and 7 are local modes of the four Sensor Target supports. The corresponding mode shape is shown in Figure 3 and forms local torsion effects. The participating effective masses of these modes are very small.

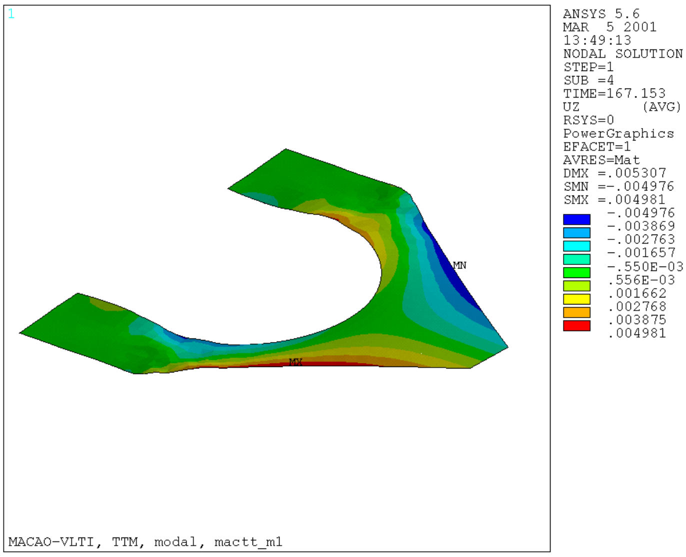

Figure 3: Mode 4 at 167 Hz.

The next group of mode shapes is presented in Figure 4, Figure 5 and Figure 6. All of these modes (between 261 Hz and 269 Hz) correspond to pure horizontal rigid body motions of the movable parts. They directly depend on the radial and axial stiffness of the flex pivots, and the mass and inertia of the movable parts. Although, the effective mass of these modes in z-direction is zero, they might have an influence on the control bandwidth and might cause instability effects.

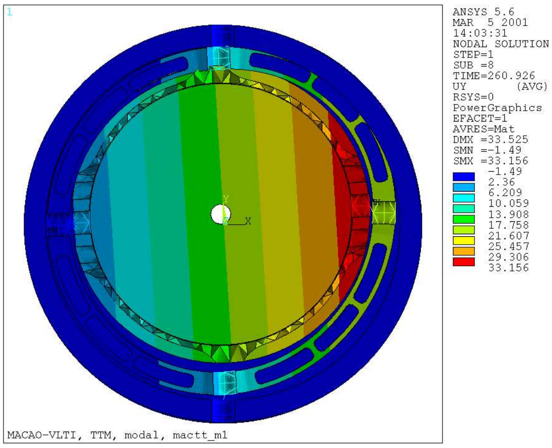

Figure 4: Mode 8 at 261 Hz.

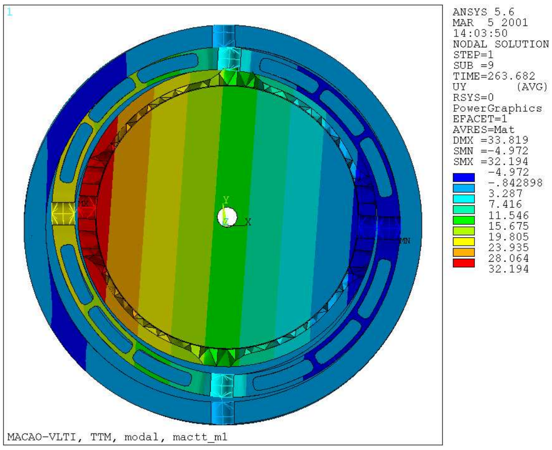

Figure 5: Mode 9 at 264 Hz.

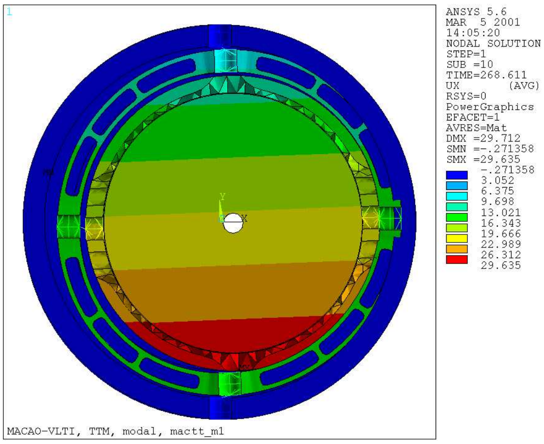

Figure 6: Mode 10 at 269 Hz.

The first flexible mode of the system outside the desired bandwidth occurs at 733 Hz (mode 11). As can be seen in Figure 7, its mode shape is an astigmatic deformation of the dummy mirror and the inner ring.

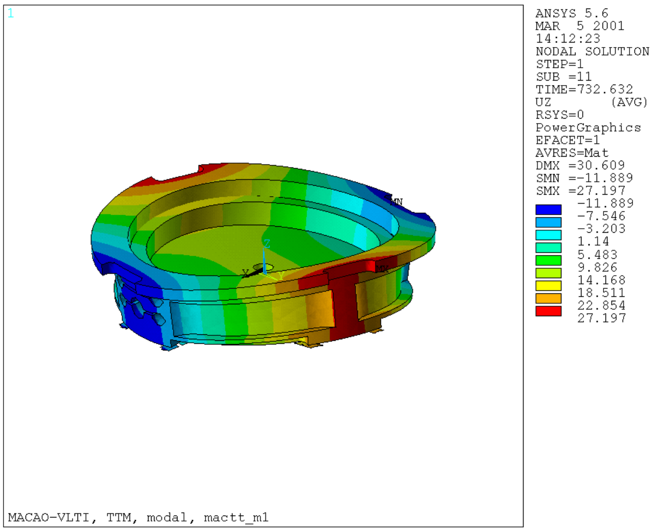

Figure 7: Mode 11 at 733 Hz.

In order to estimate the effect the actuator forces to the sensor transfer functions and the control bandwidth, some typical open loop transfer functions have been computed. The harmonic forces (61 N and the Sensor locations are shown in Figure 8. The indices mean:

- uz = target displacement in z-direction,
- Fz = actuator Force in z-direction
- xp = location positive x-axis
- xm = location negative x-axis
- yp = location positive y-axis
- ym = location negative y-axis

The analysis simulates a pure tilt excitation of the movable parts about the y-axis.

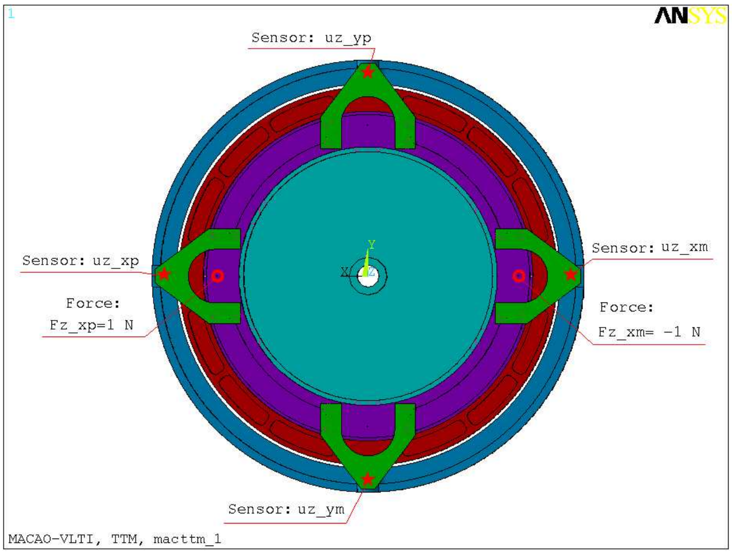

Figure 8: Location of Harmonic Forces and Sensors.

The resulting transfer functions of the four sensor targets are shown in Figure 9 in terms of displacement amplitude in z-direction and phase angle. It is obvious, that mode 3 at 135 Hz and modes 8, 9 and 10 at about 265 Hz will be seen by the sensors located opposite to the excited actuators, i.e. at the positive and negative y-axis.

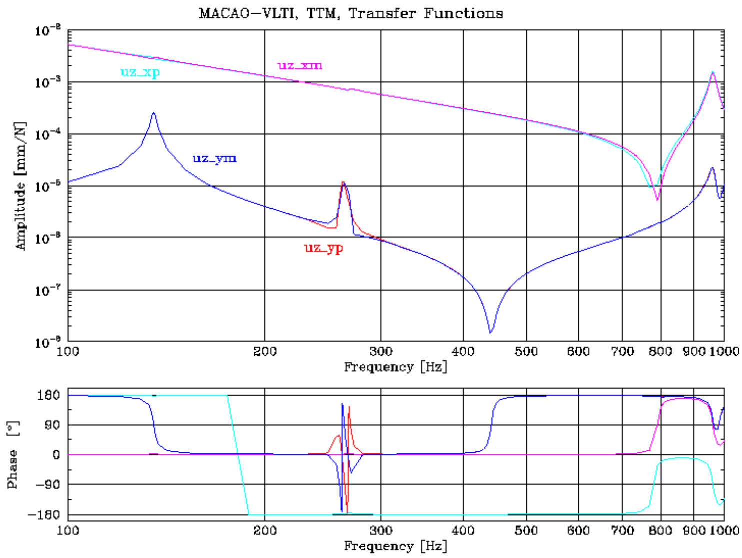

Figure 9: Transfer Functions of Reference Analysis.

Results of Modified Analysis

In order to improve the design, the frequencies of mode 3 and modes 8, 9 and 10 have to be increased. The following design measures are proposed:

1. Close the cavities of the intermediate ring structure on the top and bottom surfaces to improve the torsional stiffness of the ring.
2. Increase the radial and axial stiffness of the flex pivots.
3. Add vertical ribs at the outer rim of the Sensor Target Supports.

Design modifications No. 1 and 2 have been implemented into the FE Model by adding 3 mm thick plates on the top and bottom surface of the intermediate ring, and by increasing the pivot stiffness values by a factor of four. The 3rd design measure has not been taken into account in the modified FE model.

Due to these modifications the torsional mode of the intermediate ring could be increased from 135 Hz to **242 Hz**, and the horizontal rigid body modes of the movable parts could be increased from about 265 Hz to **380 Hz** and **431 Hz** respectively.

The relative improvement is demonstrated in Figure 10 and Figure 11 that compare the Transfer Functions of the "uz_yp"- Sensor between the reference (red) and the modified analyses (blue). It is obvious that the proposed design modifications will improve the dynamic performance of the system significantly, i.e. bandwidth increased to 242 Hz.

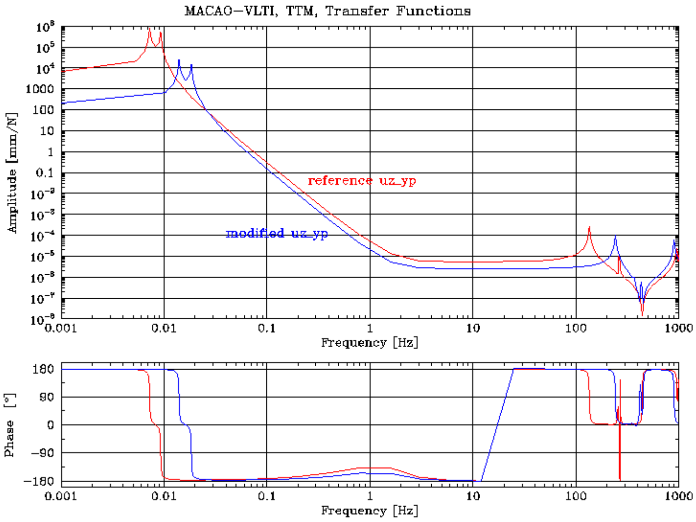

Figure 10: Transfer Functions of Reference and modified Analyses (full frequency range).

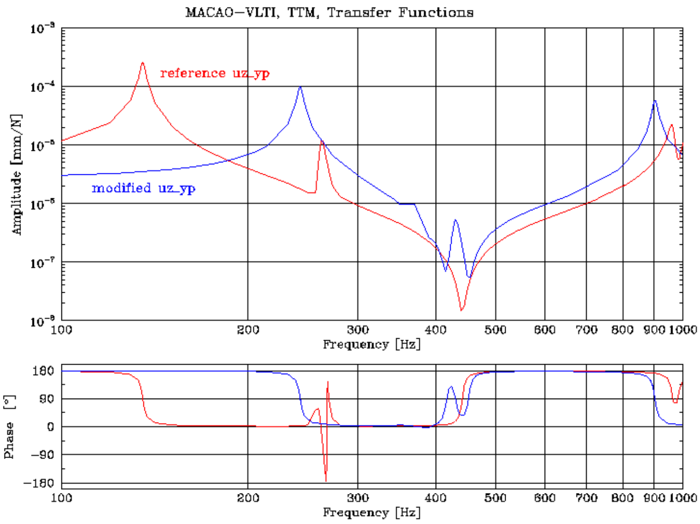

Figure 11: Transfer Functions of Reference and modified Analyses (100 Hz41000 Hz).

Conclusions

Based on the analyses the following conclusions can be drawn:

1. The lowest natural eigenfrequency of actual TTM Gimbal Mount design at **135 Hz** is far below the desired frequency of 300 Hz. This mode has an impact on the control bandwidth.
2. Three rigid body modes in horizontal directions (between 260 and 270 Hz) are below 300 Hz and might influence the control bandwidth too.
3. Local modes at 167 Hz occur at the Sensor Target supports. However, these modes are not seen by the sensors Transfer Functions.

The following design modifications are recommended to increase the bandwidth:

1. Close the top and bottom surface of the open cavities of the intermediate ring structure to increase its torsional stiffness,
2. select Flex pivots with higher radial and axial stiffness,
3. improve the level of lightweight design to reduce the mass and inertia of the movable parts, and increase their stiffness, and
4. increase the out-of-plane stiffness of the Sensor Target support plates, e.g. add vertical ribs at the outer rim.