Experimental modal analysis of the CMS MSGC Tracker prototype

P. Cupial, K. Artoos

Abstract

The modal parameters of the CMS MSGC Tracker prototype were measured experimentally. The measured quantities included the natural frequencies of the system, modal damping coefficients and mode shapes. Impulse excitation using an impact hammer as well as the excitation with an electrodynamic shaker driven by a white noise signal were used to measure the frequency response functions of the system. By applying to the shaker a harmonic signal at one of the natural frequencies of the system, the corresponding mode shape was excited. The mode shape was found by measuring the amplitude and the phase shift at different points on the structure.

The experimental results are used to verify the numerical results obtained in a FE analysis [1]. There is a good correspondence between the measured and calculated natural frequencies and the mode shapes, even though the measured frequencies are lower than the predicted values.

Keywords: CMS MSGC Tracker, vibration, experimental modal analysis

May 2001

1 CERN / EST-ME

Distribution: See list

Secretariat: EST-DV/Liliane Olivier
Table of Contents

1. INTRODUCTION .......................................................................................................................... 1
2. EQUIPMENT AND INSTRUMENTATION ..................................................................................... 2
   2.1 Accelerometers ......................................................................................................................... 2
   2.2 Impact hammer ......................................................................................................................... 2
   2.3 Electrodynam shaker, impedance head and stinger ................................................................. 2
   2.4 Analyser ................................................................................................................................ 2
3. MEASUREMENT METHODS ....................................................................................................... 2
4. RESULTS ..................................................................................................................................... 4
   4.1 Natural frequencies .................................................................................................................. 5
   4.2 Modal damping ....................................................................................................................... 6
   4.3 Mode shapes ........................................................................................................................... 8
5. CONCLUSIONS AND COMPARISON ...................................................................................... 11
6. ACKNOWLEDGEMENTS ......................................................................................................... 11
7. REFERENCES ............................................................................................................................. 11

List of figures

Figure 1 - General view of the CMS MSGC prototype and its support .............................................. 1
Figure 2 - The suspension of the electrodynam shaker ................................................................. 3
Figure 3 - The driving pin connecting the shaker through an impedance head to the tested structure ................. 4
Figure 4 - Frequency and impulse response functions ..................................................................... 5
Figure 5 - The panel between the two discs of the tracker prototype ............................................ 6
Figure 6 - Separation of the second mode of the structure with weights and the decay curve for this mode .......... 7
Figure 7 - The decay rate shown against the logarithmic scale ..................................................... 8
Figure 8 - Mode shape variation in the circumferential direction .................................................... 9
Figure 9 - Three-dimensional plots of the lowest four mode shapes ............................................ 10
Figure 10 - Nodal lines for the four lowest modes ...................................................................... 10

List of tables

Table 1 - Comparison of the calculated and measured natural frequencies ...................................... 11
1. INTRODUCTION

The experimental modal analysis of the CMS MSGC Tracker prototype was performed with the purpose of verifying an earlier Finite Element calculation and obtaining additional information about the system’s modal damping. The results of the FE analysis of the tracker prototype are discussed in reference [1], which includes the calculated natural frequencies and the mode shapes. The calculation and the experimental verification of the system’s modal parameters are important steps in building the dynamic model of a vibrating structure. With the known natural frequencies, mode shapes and modal damping coefficients, the response of any linear vibrating structure to a specified loading can be found using the modal superposition method. Possible dynamic loading in the final structure could be cooling circuits, ventilation and vibrations transmitted through the support structure.

The prototype of the tracker is shown in Figure 1. Two discs with the outer diameter equal to 2.4 m and the inner diameter of 1.4 m are connected at the inner radius by a cylinder made of composite material. The two discs are connected at the outer radius by several connecting panels. The inner radius cylinder is bolted on each side to a support frame, at two locations which lie in the horizontal mid-plane of the cylinder. These support conditions correspond to the boundary conditions of the restrained inner cylinder, which were used in the numerical calculations [1]. Even though the frame can be regarded as rigid from the static point of view, low frequency rigid-body modes of vibration of the prototype and the frame were seen during the tests. On the other hand, the proper vibration modes of the prototype did not couple with the vibration of the frame and it is expected that for these modes the condition of the fully restrained inner cylinder was valid.

Figure 1 - General view of the CMS MSGC prototype and its support
In experimental modal analysis, the most reliable way to support a structure is to suspend it freely from elastic strings. These support conditions have the advantage that they are easy to model in the numerical analysis as free boundary conditions. In the present dynamic measurements it was decided not to use this special type of support, and to use the same support conditions as for other measurements being done on the prototype.

2. EQUIPMENT AND INSTRUMENTATION

2.1 Accelerometers

Two accelerometers (PCB-ICP 352A10) were used in the analysis, with a sensitivity of about 1 mV/(m·s⁻²) and a frequency range between 1 Hz and 25 kHz. The mass of each transducer is only 0.7×10⁻³ kg. Mass loading effects on the structure were this way avoided. The accelerometers were waxed onto the structure with Petro wax (PCB 080 A109). The cables were securely fastened to the structure to minimise cable whip and connector strain.

2.2 Impact hammer

An impact hammer Bruel & Kjaer type 8202 was used in the measurement of the frequency response curves of the prototype. The hammer was equipped with a force transducer, which allowed the measurement of the applied impulse.

2.3 Electrodynamic shaker, impedance head and stinger

Harmonic and random excitation was applied to the structure with a modal exciter or “shaker” (MB Dynamics 50A modal exciter). This shaker can produce up to 225 N peak force (when cooled with forced air) between DC and 4000 Hz. The shaker together with the base had a mass of 25 kg and it was suspended with four turnbuckle hinges to reduce the mass load effects on the structure. A solid state power amplifier (MB Dynamics SL500VCF) with a frequency range between DC and 10 kHz drove the exciter. The exciter force was transmitted by a stinger (Modal shop 2150G12) via an impedance head (PCB-ICP 288D01) which was waxed to the structure. The impedance head can measure simultaneously the applied driving point force and the acceleration of the test structure with the sensitivity of 10 mV/(m·s⁻²) and 22.48 mV/N, respectively for acceleration and force.

2.4 Analyser

A dual channel FFT analyser (Bruel & Kjaer type 2034) was used to measure and analyse the signals from the accelerometers, the impedance head and the impact hammer. Moreover, the built-in signal generator of the analyser provided harmonic and random excitation signals applied to the electrodynamic shaker.

3. MEASUREMENT METHODS

The complete information about the dynamic behaviour of a linear vibrating structure can be obtained by measuring its natural frequencies of vibration, modal damping coefficients and the mode shapes. Given this information and the excitation source, the response of the structure can be found using the modal superposition approach. In order to verify the theoretical or numerical model of a vibrating system the standard approach is to compare the calculated natural frequencies and the mode shapes with those measured experimentally.
The natural frequencies and the modal damping coefficients are obtained from the so-called frequency response curves. These plots are defined as a response of a vibrating system to a harmonic excitation. Even though the frequency response curves can be obtained by using harmonic excitation with a variable frequency, it is more convenient to use signals with a continuous Fourier spectrum, which can excite the structure in a broad frequency range. Impulse and random noise signals are examples of such excitations.

Both impulse and random excitations were used on the MSGC Tracker prototype. For the impulse excitation, the impact hammer Bruel & Kjaer type 8202 was used. The impact hammer is equipped with a force transducer that allows the measurement of the magnitude of the applied pulse. The response of the system was measured with a lightweight piezoelectric accelerometer waxed at a selected point on the prototype. The signal from the impact hammer was connected to channel A of the FFT dual channel analyser and the signal from the accelerometer was input to channel B. By calculating the Fast Fourier Transform of the two signals, and dividing the two spectra, the frequency response curve is found. To obtain high quality frequency response curves, averaging of the results has been done over several impacts.

Higher quality frequency response curves can be obtained using random excitation. A white noise signal from the built-in generator of the analyser was applied through a power amplifier to the electrodynamic shaker. The shaker was suspended with four turnbuckle hinges from a support frame as shown in Figure 2.

![Figure 2 - The suspension of the electrodynamic shaker](image-url)
The shaker coil was attached to the structure via a “stinger” or driving pin and a load impedance head as shown in Figure 3. The driving pin has a high stiffness in the axial direction of the shaker coil and is flexible in the transverse direction. This minimises the influence of side load effects on the structure (such as forces other than those along the shaker axis and moments). The impedance head integrates a load cell and an accelerometer.

Similarly to the case of impact excitation, the response was measured with an accelerometer waxed at a selected point on the structure. The signals from the impedance head load cell and the accelerometer were conditioned and input to channels A and B of the analyser. Because of the random nature of the applied signal and the measured response, the averaging process was applied in calculating the frequency response curves.

In order to measure the mode shapes, the so-called sine dwell technique has been used. This method consists in exciting the structure at one of its natural frequencies. Two accelerometers have been used to measure the mode shape. The reference accelerometer was kept fixed at an arbitrary point on the structure. The other accelerometer was moved over the prototype. More than twenty points were measured along the circumference and four points along each of three selected radial lines. The amplitude of vibration and the phase shift relative to the reference were measured for each point.

4. RESULTS

The measurements were performed with and without the metal rods inserted into the structure to simulate the weight of the silicon modules. The measurements with the weights included frequency response curves obtained with impact and white noise excitation, modal damping and mode shapes. The measurements without the weights included the frequency response curves measured with a white noise excitation and the mode shapes.

The results from the analyser are presented with images of the screen, as no data exchange in a windows compatible format was available.
4.1 Natural frequencies

The natural frequencies of the prototype were measured using both impact hammer and shaker excitation. Practically the same values were found using the two approaches.

An example of the measured curves is shown in Figure 4, for the CMS MSGC tracker prototype loaded with the rods, which account for the additional weights of the silicon modules. The upper plot shows the frequency response curve and the corresponding impulse response is shown in the lower plot. The curves shown have been obtained using the white noise excitation applied to the shaker. The impulse response has been calculated as the inverse Fourier transform of the frequency response function.

![Image of frequency and impulse response functions](image)

**Figure 4 - Frequency and impulse response functions of the prototype loaded with the weights and excited with white noise.**

The four lowest natural frequencies read from the frequency response plot, for the structure with weights, are equal to:

6.06, 8.31, 12.94 and 19.50 Hz

The respective four lowest natural frequencies measured on the prototype without weights are:

16.63, 18.25, 25.38 and 37.94 Hz

The measured natural frequencies do not depend on the location of the excitation point or the location of the response measurement point. However, a given natural frequency can appear higher or lower in the frequency response curve depending on these details. The above values of frequencies were also measured when exciting the structure on one of the discs and measuring the response on the opposite disc. This means, that the above frequencies correspond
to modes, in which the two disks move together, rather than to separate movements of individual discs. This is in agreement with the theoretical predictions in reference [1].

As is normal for vibrating structures, the additional mass has an effect of lowering the system’s natural frequencies.

The lower plot in Figure 4 shows the impulse response curve. The impulse response shown is a combination of the decay curves of several modes.

Apart from the natural frequencies of the bending modes of vibration of the prototype discs, the natural frequencies of a panel connecting the two discs were also measured. The panel is shown in Figure 5. Impulse excitation with the impact hammer was used. To obtain all the natural frequencies of the panel, the location of the accelerometer and the excitation point had to be different from the symmetry point of the panel (symmetrical position of the accelerometer is shown in Figure 5).

The five lowest natural frequencies of the panel were found to be:

95, 177, 219, 290 and 346 Hz

4.2 Modal damping

To determine the modal damping of each individual mode, one approach is to separate the mode in question in the frequency response plot and find the decay rate for the selected mode. Even though many other methods of measuring damping exist (e.g. the half-bandwidth method), the decay rate approach has the advantage of giving accurate results in the present case of a lightly damped structure. The approach is illustrated in Figure 6, for the second mode. A region is isolated around the frequency of interest as is shown in the upper plot. The impulse response curve for the selected mode is obtained by the inverse Fourier transform (lower plot).
The logarithmic decrement of any mode is defined as:

$$\delta = \frac{1}{N} \ln \frac{X_i}{X_{i+N}}$$

where $X_i$ represents the peak amplitude after $i$ cycles and $X_{i+N}$ is the peak amplitude after $i+N$ cycles.

Other damping measures are often used, depending on the research field and individual preferences. For instance, in the finite element code ANSYS, the quantity input directly to the mode superposition method is the so-called modal damping ratio $\xi$. For small damping this is related to the logarithmic decrement by $\xi = \frac{\delta}{2\pi}$.

Even though the modal damping can be read from the decay rate of the bottom plot in Figure 6, it is better to show the decay envelope against the logarithmic scale, as is shown in Figure 7. For exponential decay one obtains a straight line and the modal damping ratio coefficient can be found as $\xi = \frac{1}{2 \cdot \pi \cdot f \cdot \Delta t_{8.7\,dB}}$. Here $f$ is the natural frequency in Hz of the mode in question and $\Delta t_{8.7\,dB}$ is the time during which the signal decays by 8.7 dB.
The following damping ratios have been found for the four lower modes of the prototype loaded with the bars:

\[ \xi_1 = 0.015, \xi_2 = 0.007, \xi_3 = 0.006, \xi_4 = 0.005 \]

Modes two, three and four are lightly damped. A higher damping was measured for mode one. For this mode dry friction damping from the possible movement of the loading rods can contribute to the increased damping of this mode. Such a movement was observed and was heard during the measurement.

### 4.3 Mode shapes

The four first mode shapes were measured by harmonically exciting the prototype at one of the system’s natural frequencies. The force was applied by the shaker at the right-hand side of the prototype, just below the horizontal axis (see Figure 1). A reference accelerometer was kept fixed at the location on the horizontal axis, on the left of the prototype, close to the prototype’s outer diameter. The angular position of the reference accelerometer is taken as 0°. The excitation point thus corresponds to an angle of about 180°.

A second accelerometer was moved over the surface of the front disk of the prototype to measure the mode shape. The amplitude of the acceleration and the phase shift at a given point relative to the reference point were measured. For a lightly damped structure the mode shapes are practically real-valued functions. This is equivalent to the fact that when the structure vibrates in one of its modal shapes, different points can move either in phase with the reference point (the phase shift equal to 0°) or out of phase (the phase shift equal to 180°).
The results of the measurements at points along the prototype circumference are shown in Figure 8. The dots represent the values measured experimentally. The continuous lines were obtained by the least square fit to the measured values. For the first and second mode a good fit is obtained using a constant term and the sine and cosine terms with one wave along the circumference. The analytical formulae resulting from the least square fit are shown under the plots in Figure 8. For mode three, sine and cosine terms with two waves give a good fit, for mode 4 a shape with three waves gives a good approximation to the measured results. For most points, the measured values do not have a big scatter about the least square fit curves, however individual points exist which are further away from the curves. These points reflect some local variations in the mode shapes, which can originate from the fact that the discs are not continuous elements but are structures with many cuts.

To obtain the spatial plots of the mode shapes, the amplitudes and phase shifts were measured along radial directions, at three angular positions corresponding to 12°, 108° and 232°. All the radial plots had a similar shape with practically zero displacements at the inner radius of the disc (which is attached to the cylinder) and the amplitude increasing towards the outer radius. Using the measured three radial lines, interpolation was used to obtain values at radial lines which were not measured directly. The spatial mode shapes were then obtained by assuming the mode shapes to be the product of the least square fit functions in the circumferencial direction and the interpolated radial lines. The result is shown in Figure 9, for the lowest four modes. Figure 10 shows the nodal line patterns (in these plots 0°, which is the location of the reference sensor, is the vertical position on the left of the plots).

The first mode shape, which corresponds to the frequency 16.63 Hz, is a drum-like mode with an additional one wave. The second mode is principally one wave in the circumferencial direction plus a small constant term. In our view this is caused by a coupling between the closely spaced modes one and two (with the respective frequencies of 16.63 Hz and 18.25 Hz). By applying a harmonic excitation at the lowest natural frequency the first mode will be at resonance and will dominate the response. However, other modes are also excited, especially those with close frequencies. For modes three and four, the mode shapes are very well approximated by the surfaces with two and three waves in the circumferencial direction.
Figure 9 - Three-dimensional plots of the lowest four mode shapes

Figure 10 - Nodal lines for the four lowest modes

The above measured patterns correspond to the theoretical mode shape predictions for structures with circular symmetry. The mode shapes of such structures have the respective
number of waves along the circumferential direction. However, due to the circular symmetry
the angular node locations are not determined uniquely and the nodal pattern can be rotated by
an arbitrary angle. If the structure is excited at a given location at one of the system’s natural
frequencies, the response is at the corresponding mode shape. The location of the nodal lines
will depend in this case on the force application point. The results of the measurements on the
prototype show that for all the measured modes an anti-node point with a maximum
displacement corresponded to a position close to 0°, 180° from the shaker attachment point.

5. **CONCLUSIONS AND COMPARISON**

Table 1 shows a summary of the calculated and measured lowest natural frequencies. The
calculated values correspond to the case without the weights. Quite good agreement exists
between the calculated and the measured values, but all measured values are lower than those
calculated using the finite element method.

<table>
<thead>
<tr>
<th></th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3 (Hz)</th>
<th>Mode 4 (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Calculation [1]</strong> without weights</td>
<td>23.8 (Hz)</td>
<td>24.8 (Hz)</td>
<td>31.5 (Hz)</td>
<td>____</td>
</tr>
<tr>
<td><strong>Measurement without weights</strong></td>
<td>16.63 (Hz)</td>
<td>18.25 (Hz)</td>
<td>25.38 (Hz)</td>
<td>37.94 (Hz)</td>
</tr>
<tr>
<td><strong>Measurement with weights</strong></td>
<td>6.06 (Hz)</td>
<td>8.31 (Hz)</td>
<td>12.94 (Hz)</td>
<td>19.50 (Hz)</td>
</tr>
</tbody>
</table>

Table 1 - Comparison of the calculated and measured natural frequencies

The measured mode shapes one and two are as those calculated numerically [1]. The measured
mode three has two nodal lines and is different from mode three given in [1]. The measured
mode three is most likely the next theoretical mode not discussed in [1]. Both theoretical modes
two and three discussed in reference [1] have one nodal line – vertical for mode two and
horizontal for mode three, and have slightly different frequencies. For a perfectly symmetrical
structure these two modes would have the same frequency and differ only in the rotation of the
nodal lines.

In experimental modal analysis, it is always possible that some modes are not excited if the
force is applied close to the nodal point. However, using impact excitation at different points on
the prototype it was verified that no other frequencies from those shown in Table 1 were excited
in the analysed frequency range.

6. **ACKNOWLEDGEMENTS**

We would like to thank Harri Katajisto, Antti Onnela and Kari Tammi for preparing the
prototype and their interest in the experiments reported in this note.

We would also like to thank the Institute of Mechanics and Machine Design of the Cracow
University of Technology in Poland for the loan of the dual channel analyser and the impact
hammer used during the tests.

7. **REFERENCES**

[1] Harri Katajisto, Big wheel modal analysis using FEM, TOB-Note 00.4, July 25,2000