

ENGINEERING STUDIES ON A CFRP STRUCTURE FOR A HIGH-ENERGY PHYSICS EXPERIMENT

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SUMMARY: The performance of the future high-energy physics experiments depends on the stability and design of the supporting structures of the particle's detection elements. These structures, typically with several meter long, must be light, highly stable, stiff and radiation tolerant in an environment where external vibrations, high radiation levels, material ageing, temperature and humidity gradients are not negligible [1]. These requirements generally orient the materials choice towards Carbon Fibre Reinforced Plastics (CFRP) with highly performing fibres and resins qualified for aerospace applications. The engineering design of these structures comprises a careful development of finite element models and further validation by prototype testing.

This paper presents the methodology used in the certification process of one of these structures: a 2.5 meter CFRP-honeycomb disc known as alignment wheel. A comparison of simulation results with experimental measurements in a full-scale prototype, concerning the dynamic characterisation of the structure, is presented. The analysis and the full-scale test provide an optimised approach of the construction with respect to deformations, natural frequencies and material used.

KEYWORDS : Modal Testing, CFRP, Lightweight Structures, High-Energy Physics Experiment

INTRODUCTION

The European Laboratory for Particle Physics foresees in 2005 the construction of two large purpose high-energy physics experiments - CMS and ATLAS - at the Large Hadron Collider (LHC) presently in preparation at CERN. The Compact Muon Solenoid experiment (CMS) aims studying the collisions of very high energy hadron beams. The main scope of this experiment is the investigation of the most fundamental properties of matter and the identification and precise measurement of muons, electrons and photons. The inner detector, Tracker, plays a major role in all physics searches and its performance depends upon the intrinsic detector characteristics, on the stability of the supporting structures and on the overall position monitoring and alignment system. On the other hand, the performance of the Position Monitoring System (PMS) for the Tracker is strongly related with the stability and performance of two composite structures known as alignment wheels [2]. Each alignment wheel, with 2.5m diameter, is a sandwich structure with 2mm thick skins of standard carbon fibre composite and 16mm core of aramid honeycomb. It is a fundamental piece of the alignment system and presents stability requirements in the order of some tens of micron. The structure provides 24 connections for the various elements of the PMS plus 3 for the fixations to the detector. It is horizontally divided in two halves due to operational purposes. The full scale prototype mechanical drawing is shown in Fig. 1.

In situ, the structure sits in the centre of several sub-detectors, which will selectively filter and possibly amplify vibrations at specific frequencies. While seismic vibrations can be quantified, other sources like vibrations from ventilation systems, pumps for cryogenics, vacuum and cooling, can only be realistic quantified by performing measurements on the final structures. Thus, in the engineering design the preliminary goal is to increase the first natural

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frequency as high as possible without leading to bulky or heavy structures, which would be in conflict with physics requirements. Then the vibration amplitude should be estimated and compared with design criteria and experimental measurements, and the need of passive or active control should be determined [3]. The results concerning the dynamic testing of the alignment wheel prototype, presented here, are an essential part in the study of the propagation of vibrations in the structure and in the quantification of the vibration risk.

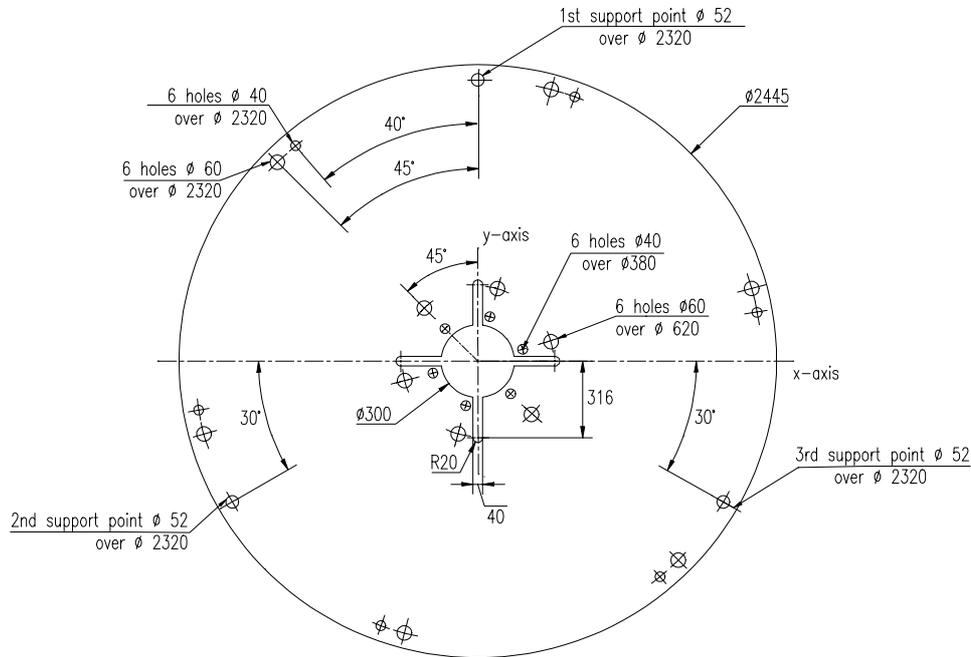


Fig. 1 Mechanical drawing of the 20mm thick full scale prototype of the alignment wheel (x-axis and y-axis are in-plane directions; z-axis is out-of-plane)

EXPERIMENTAL SET-UP

The verification test program of the alignment wheel involves the determination of frequency response functions, natural frequencies, damping factors and mode shapes; together with the validation and update of the finite element model.

In order to approximate the free boundary condition the structure is suspended from 2 fixation points by steel flexible cables (Fig. 2). By choosing a free support instead of a grounded support, the structure is isolated from parasite excitations and the modelisation in itself is facilitated [4]. On the other hand, with this kind of support, the structure is constrained in in-plane motion but free in the normal direction with rigid mode behaviour. In order to ensure a very low frequency for the suspension cables, a lumped mass of 4 Kg is mounted on each cable lowering their frequency to 1.87 Hz.



Fig. 2 Lumped mass to lower the steel cable suspension's natural frequency

A single-point excitation is adopted. The data taking schematics of the experimental set-up is shown in Fig. 3.

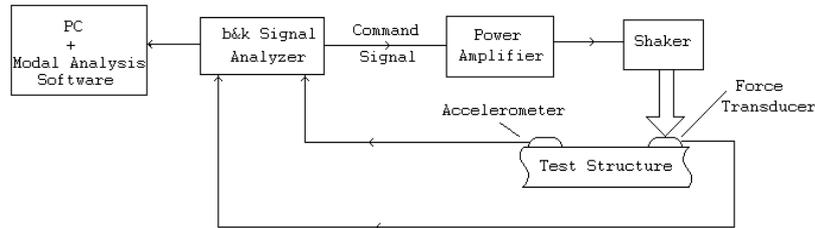


Fig. 3 Schematic layout of the data taking set-up used in the modal testing

An electromagnetic shaker provides the excitation signal to the system through a piezoelectric force transducer and the response is measured by an accelerometer attached to it by a thin layer of bees wax. A spectrum analyser is used to measure the force and the response signals needed to build the frequency response functions (FRFs) of the system. The excitation force is broadband and is generated by a signal generator that sends the signal to a power amplifier driving the electrodynamic vibration exciter. The waveform generator is built-in the analyser. The excitation is random and continuous in time [5]. The FRFs obtained during the analysis are stored in a computer running the modal analysis identification package CADA-PC from LMS. The experimental set-up is shown in Fig. 4.



Fig. 4 Experimental set-up. Alignment wheel full scale prototype suspended

The measurements lead to the determination of the system's frequency response inertance (g/N) from which modal parameters are identified. The use of a shaker guarantees a good signal-to-noise ratio and an accurate control of the applied force.

The connection between the shaker and the structure is done by means of a steel stinger. This attachment has high axial stiffness but low transverse stiffness giving good directional control of the excitation (Fig. 5). The shaker itself is supported by 4 steel cables attached to a frame (Fig. 4). This procedure ensures that the reaction force imposed on the shaker (opposite to the one applied to the structure) is not transmitted to the test structure.

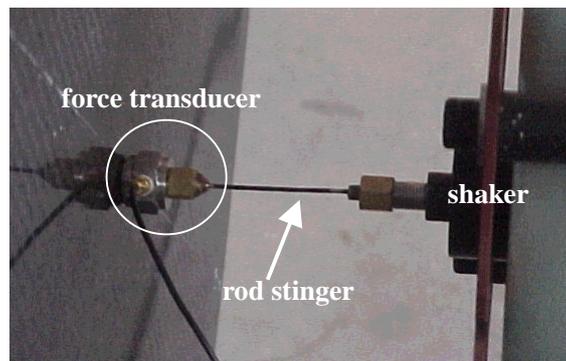


Fig. 5 Detailed system excitation

The measurement of the structure's response to the excitation spectrum is performed on a uniformly distributed measuring mesh of 128 points, as is shown in Fig. 6.

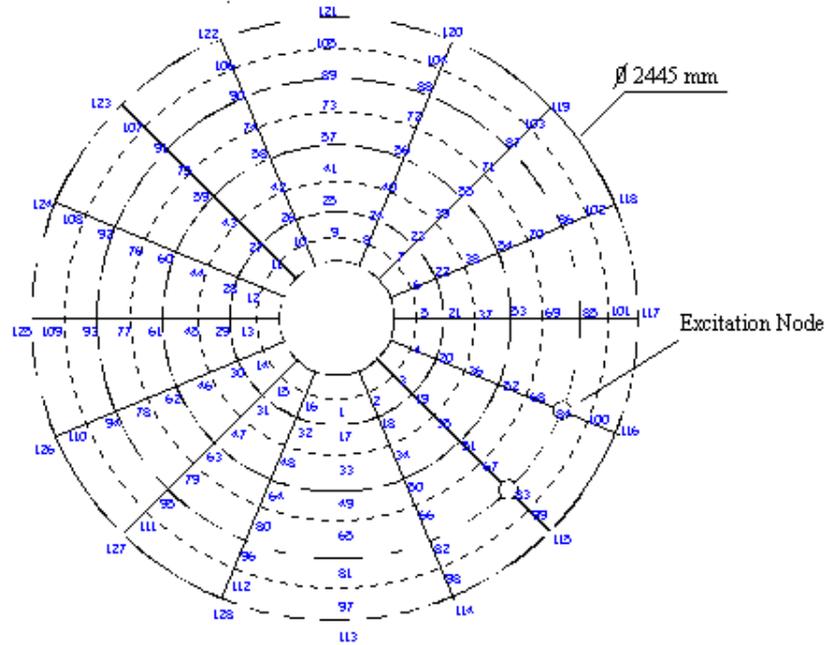


Fig. 6 Measurement mesh. Excitation applied at node 84

RESULTS

The experimental measurements are performed in the frequency bandwidth 0 to 100 Hz with a frequency resolution of 0.125 Hz. Each FRF results from a frequency domain averaging of 25 samples, used to reduce the influence of random errors. To avoid the leakage errors an Hanning window is applied to weight the excitation and the response data. For modal parameter estimation, a multi degree technique is used. It uses a time domain algorithm to identify poles (frequency and damping values) by the so-called *Least Squares Complex Exponential* (LSCE) method. In a second phase, residues are identified with the *Least Squares Frequency Domain* (LSFD) technique. Modal parameters are extracted from the 128 measured FRFs by a modal analysis software: CADA-PC from LMS. Ranging from 0 to 100 Hz, Fig. 7 and Fig. 8 show 3 measured FRFs: the driving point FRF $H_{84,84}$ (response node 84, excitation node 84), the FRF $H_{11,84}$ (response node 11, excitation node 84) and the FRF $H_{119,84}$ (response node 119, excitation node 84) overlapped with the Modal Indicator Function (MIF) and the Sum function of the measured FRFs, respectively [5] [6].

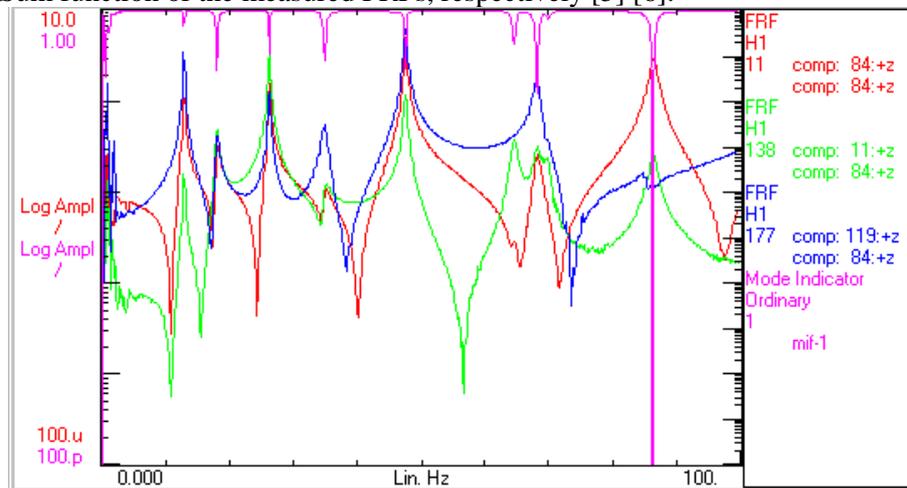


Fig. 7 Measured frequency response functions and MIF function

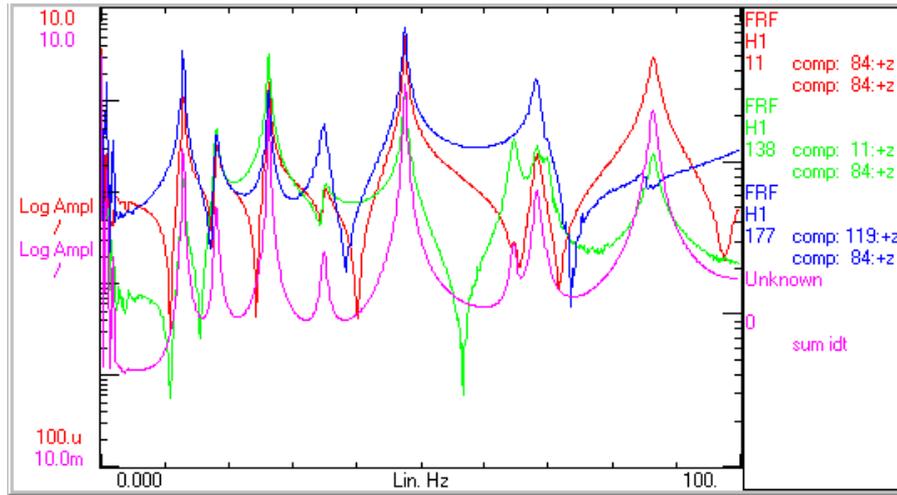


Fig. 8 Measured frequency response functions and SUM function

The Modal Indicator Function, built upon the set of the measured FRFs, is used to show the location of each mode.

The presence of 5 modes in the analysis bandwidth of 6-69Hz were identified. Table 1 shows the experimental modal analysis parameters.

Mode Nr.	Frequency [Hz]	Damping [%]
1	12.80	0.69
2	18.04	1.29
3	26.25	0.39
4	34.87	1.22
5	47.59	0.44

Table 1 Experimental modal parameters obtained with the LMS CADA-PC software

In order to validate the modal model identified, some FRFs are synthesised and compared with the measured ones. Fig. 9 shows the measured and synthesised FRF $H_{11,84}$. A good correlation is achieved.

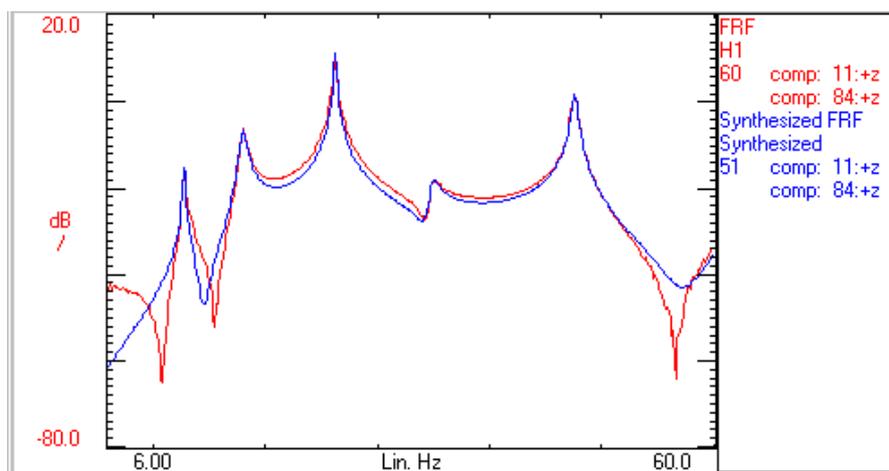


Fig. 9 Measured and synthesised FRF $H_{11,84}$

The validation of the model is completed using the Modal Assurance Criterion (MAC). This criterion correlates a mode shape for one pole against the mode shape of all the others poles. If the model is valid, little or no correlation between modes should be found as confirmed in Table 2, showing the MAC matrix.

Freq. [Hz]	MAC Matrix					
	Mode Nr.	1	2	3	4	5
12.80	1	100.0	0.0	0.0	1.1	0.0
18.04	2	0.0	100.0	3.3	0.0	0.1
26.25	3	0.0	3.3	100.0	0.0	0.0
34.87	4	1.1	0.0	0.0	100.0	0.0
47.58	5	0.0	0.1	0.0	0.0	100.0

Table 2 Modal assurance matrix obtained with the LMS CADA-PC software

The biggest off-diagonal term of the MAC Matrix is 3.3%, thus, there is no correlation between mode shapes and a good analysis is assured. The first five experimental mode shapes are presented in Fig. 10 to Fig. 14. The left hand side figures show the nodal lines together with the wire frame animation of the mode while the right hand side figures show clearly the displacement amplitudes at each node of the measurement mesh.

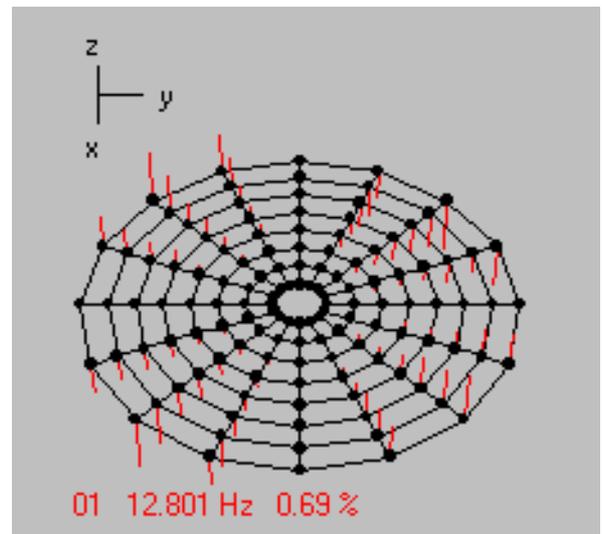
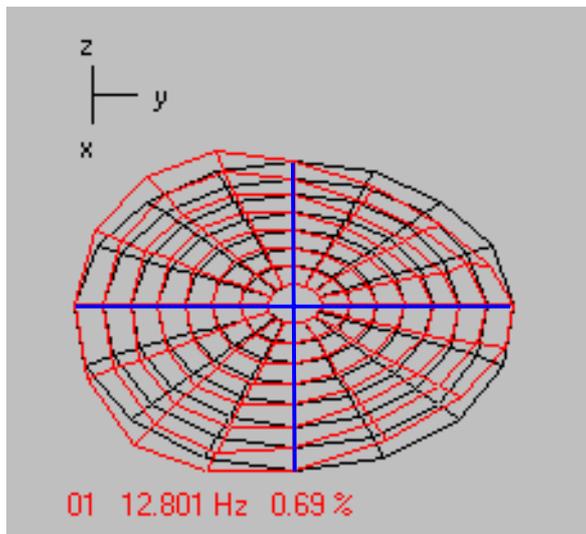


Fig. 10 First mode shape (12.80 Hz)

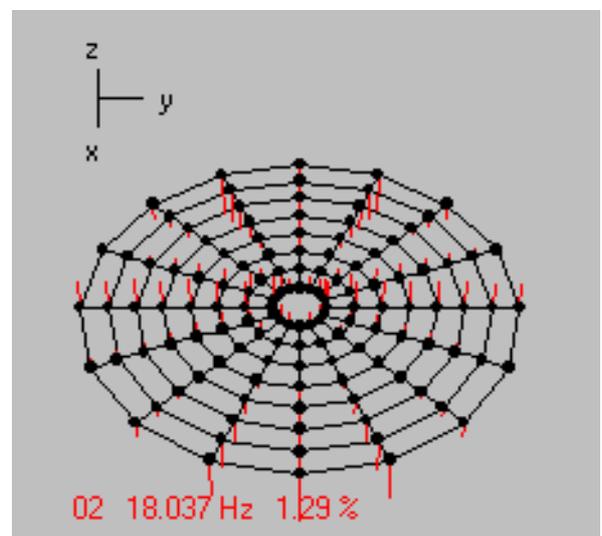
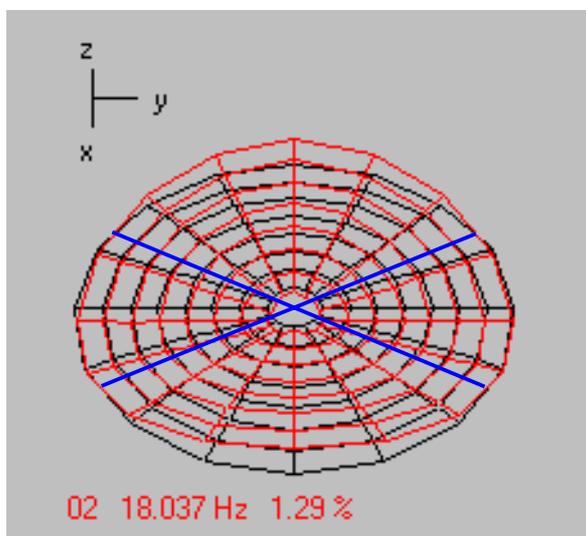


Fig. 11 Second mode shape (18.04 Hz)

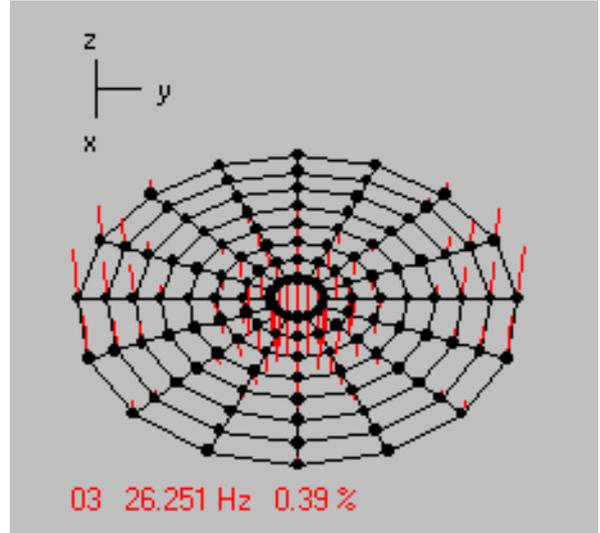
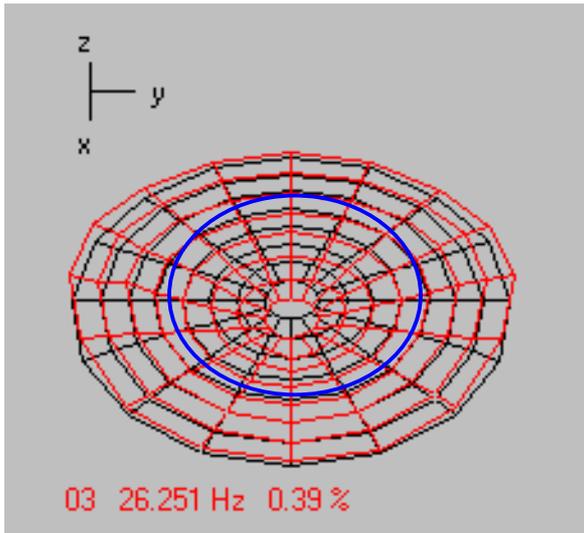


Fig. 12 Third mode shape (26.25 Hz)

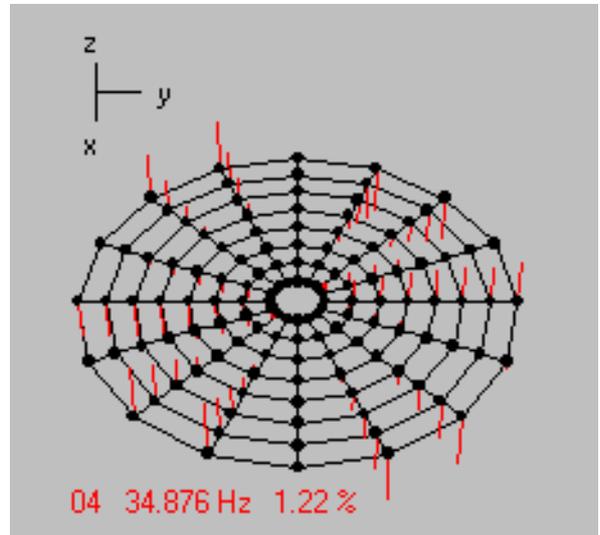
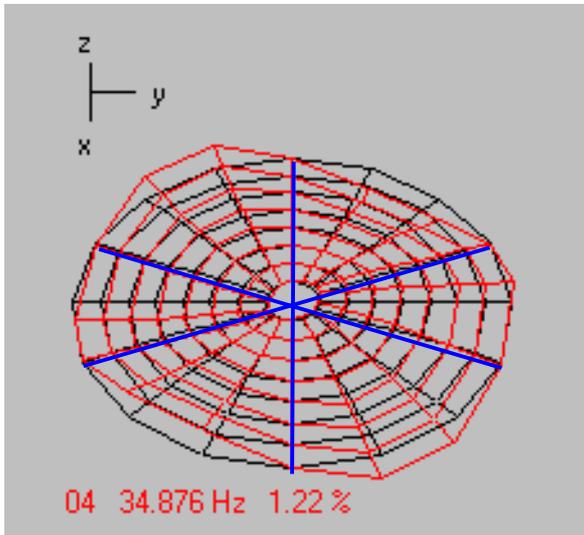


Fig. 13 Fourth mode shape (34.87 Hz)

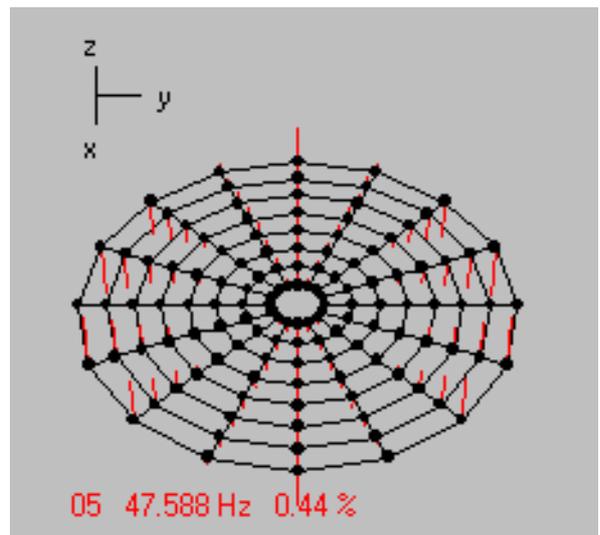
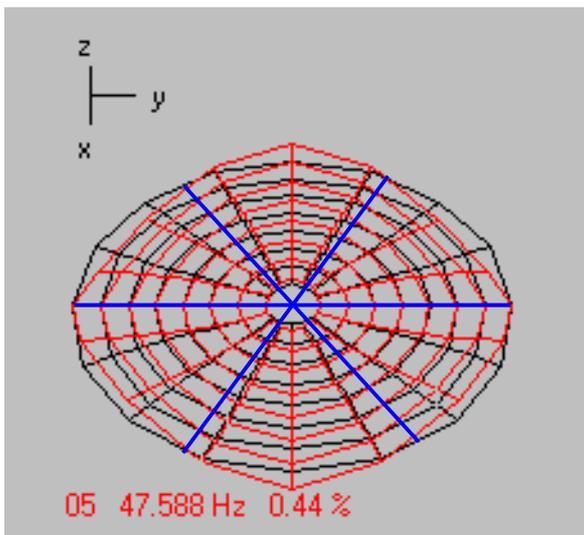


Fig. 14 Fifth mode shape (47.59 Hz)

The first mode shows the existence of two perpendicular nodal diameters. The second mode shape, very similar to the first one, presents again two nodal diameters rotated with respect to the ones of the previous mode. The third mode clearly presents one nodal circle, while, the fourth mode, together with the fifth, have three nodal diameters.

FINITE ELEMENT ANALYSIS

The aim of this study is to adjust the finite element (FE) model with the dynamic characteristics obtained experimentally and to reduce the errors as low as possible. The update of the model, performed using a commercially available finite element software, ANSYS 5.6.2., is done in two different stages. First the determination of the equivalent mechanical properties of the sandwich is done using a small circular disk of 800mm diameter with no holes and made with the same material. The experimental determination of the natural modes of this disk, using the same procedure described above, and the comparison with the theoretical ones enables the calculation of the equivalent Young Modulus of the sandwich structure. This result is then introduced in the alignment wheel finite element model and the cross check with the experimental results is performed.

The 800mm diameter disk was tested in a free boundary condition. The first four natural frequencies were: 108.6Hz, 164.6Hz, 268.1Hz and 307.6Hz. According to [7] the natural frequencies of circular plates with free edges can be determined by the expression (1).

$$f = \frac{\lambda}{2\pi} \frac{1}{R^2} \sqrt{\frac{D}{\rho h}} \quad , \quad D = \frac{Eh^3}{[12(1-\nu^2)]} \quad (1)$$

where, f , is the natural frequency in Hertz, D , the flexural rigidity, E , the Young modulus, ν , the Poisson ratio, h , the plate thickness, ρ , the mass density, R , the plate radius and, λ , the frequency parameter. For the first four natural frequencies, λ , is respectively equal to 5.358, 9.003, 12.439 and 20.475.

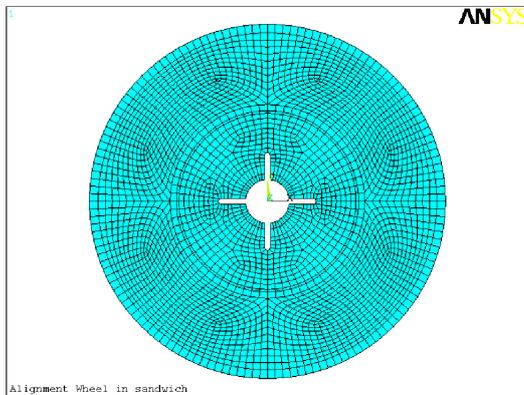


Fig. 15 Finite element model of the alignment wheel (ANSYS 5.6.2.)

Based in the experimental values of the first four natural frequencies of the small disk and on the above expression, the equivalent Young Modulus was considered to be equal to 3.50GPa. This value together with the equivalent mass density of the sandwich structure (330 Kg.m^{-3}), determined by weighting a sample of material, were introduced in the finite element model of the alignment wheel (Fig. 15). A 4-node shell element with 6 degrees of freedom per node and with both membrane and bending capabilities was chosen for the model (SHELL63).

Table 3 shows the first five natural frequencies obtained experimentally and by finite element calculation, together with the error percentage.

Mode Nr.	Exp. Freq. [Hz]	FE Freq. [Hz]	Error [%]
1	12.80	11.23	-12.3
2	18.04	16.33	-9.5
3	26.25	27.01	2.9
4	34.87	39.23	12.8
5	47.59	47.26	-0.7

Table 3 First five natural frequencies of the alignment wheel structure obtained experimentally and by finite element analysis

One can observe that in spite of having determined the mechanical properties of the sandwich material in an indirect way the difference between experimental and simulation natural frequencies is lower than 13%. These results validate the FE model of the alignment wheel, as well as, the material homogenisation considered in the calculations. The obtained errors are undoubtedly related with the lack of experimental data in the determination of the mechanical properties. The first five corresponding mode shapes obtained with ANSYS 5.6.2. are presented in Fig. 16 to Fig. 20. The nodal lines lie in the green zones.

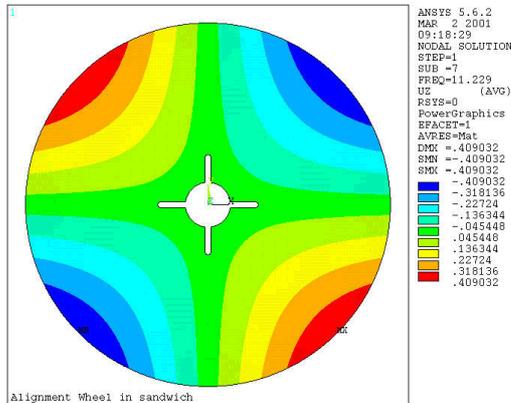


Fig. 16 First mode shape

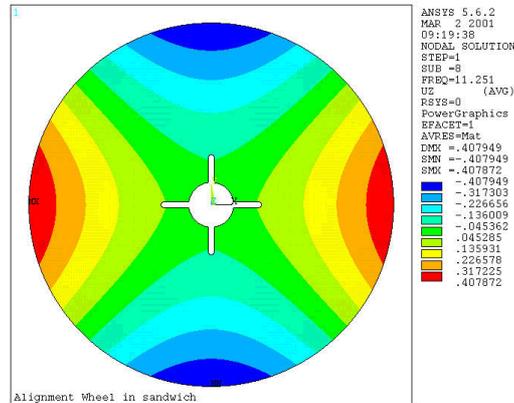


Fig. 17 Second mode shape

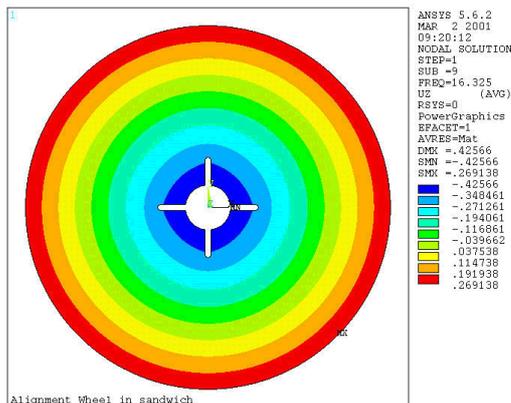


Fig. 18 Third mode shape

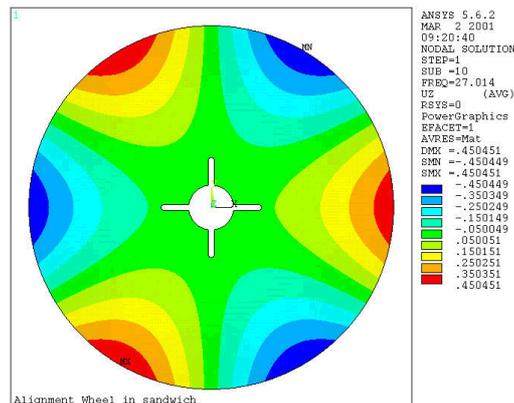


Fig. 19 Fourth mode shape

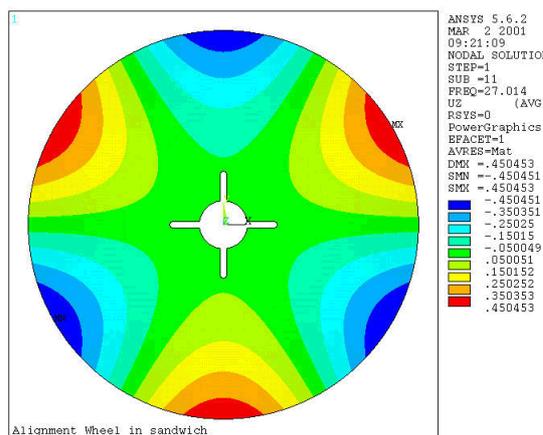


Fig. 20 Fifth mode shape

Table 4 shows the nodal circles (s) and nodal diameters (n) obtained experimentally and by finite element analysis.

	1 st Frequency		2 nd Frequency		3 rd Frequency		4 th Frequency		5 th Frequency	
	n1	s1	n2	s2	n3	s3	n4	s4	n5	s5
Exp.	2	0	2	0	0	1	3	0	3	0
FE	2	0	2	0	0	1	3	0	3	0

Table 4 Nodal diameters (n) and nodal circles (s) obtained experimentally and by finite element analysis

The FE model assumes the material as perfectly homogeneous and does not take into account the joint between the two halves of the structure, the embedded aluminium inserts and the edge sealing material of the sandwich structure - adhesive in the exterior diameter and aluminium parts in the interior diameter. Due to these facts, there is a non-direct correspondence between the mode shapes and the frequencies obtained experimentally and by simulation. This means, for example, that in the case of the first and of the second mode shapes, both with two nodal diameters rotated with respect to each other, the finite element model associates two similar frequencies (11.23Hz and 11.25Hz) and not distinct ones as observed experimentally. This fact is again observed for the fourth and fifth modes (27.01Hz and 27.01Hz), both with 3 nodal diameters. Once again, the FE model cannot distinguish between the two cases.

However the relevant fact is that the same natural frequencies and the same mode shapes found experimentally are again found by simulation. Therefore, in spite of the assumptions made in the FE model, these results validate the FE model in itself, as well as, the mechanism used to fix and to keep in place the two halves of the alignment wheel.

CONCLUSION

The experimental results concerning the dynamic characterisation of the full scale prototype of the alignment wheel were obtained and compared with the simulation ones. Concerning the first five natural frequencies, errors less than 13% were achieved. The identified mode shapes were in good agreement with the ones calculated using the finite element analysis software ANSYS 5.6.2.

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