MODAL TEST ON THE PININFARINA CONCEPT CAR BODY "ETHOS 1"

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ABSTRACT

This paper describes the modal test and analysis carried out on the body of a 2 passenger cabriolet concept car designed by Pininfarina. It discusses the specific problems in a modal test on such a car body of extruded aluminium profiles. Next, the experimental modal model is correlated with an MSC/NASTRAN analytical model, and a diagnosis is made to improve the analytical model.

INTRODUCTION

The "Ethos" project is a complete industrial project, achieved with the aid of the latest technologies. With this project, Pininfarina aimed to make a major contribution to the protection of our environment by building a concept car with fully recyclable materials, and with extremely low emission pollution.

One of the primary means to achieve this is to design the car body based on the space frame technique of extruded aluminium profiles. Although the primary design objective was a reduced weight, the dynamic behaviour of such a car is of particular importance to meet comfort and safety requirements.

The paper describes and discusses the two-shaker modal test on the car body, and the comparison of these results with the modal behaviour predicted with an MSC/NASTRAN model..

THE ETHOS CAR BODY

Figure 1 shows a picture of the tested car body. This body only weighs 85 kg. The total weight of the assembled car is approximately 700 kg.

Making use of recyclable extruded aluminium profiles nevertheless results in unusual rigidity properties. This is of particular importance for a cabriolet "barchetta" car, because of obvious comfort and safety reasons.

The Ethos car is a small car with a length of 3.63 m, a width of 1.66 m, and a height of 1.05 m. It is powered by a 1200 cc 2-stroke engine with 3 cylinders, with a maximum torque of 125 Nm at 3500 rpm.

More details on this concept car can be found in [1].

EXPERIMENTAL TEST SET-UP

The tested car body, with a total mass of only 85 kg, was suspended with elastic cords to simulate free-free conditions as close as possible. Two electro-magnetic shakers were used to excite the structure in the vertical direction. Figure 2 shows the geometry model, and the two shaker positions.

The acquisition front-end counted 14 channels. Next to the 2 force signals, 12 acceleration signals from PCB Flexcels were acquired simultaneously (4 nodes in 3 directions at a time). FRFs were measured between 0 and 300 Hz, using a frequency resolution of 0.58 Hz.

The structure was excited by two uncorrelated burst random force signals. These were generated by the workstation, and sent out to the shakers. The H1 estimator was used to compute FRFs from auto- and cross power spectra. 50 averages reduced the measurement noise.

94 nodes were measured in 3 directions. In total, 564 FRFs were measured with the LMS CADA-X Test system.

MODAL MODEL EXTRACTION

From the dual input FRF set, a modal model was extracted using the frequency domain direct parameter identification ([2], FDPI) technique to identify frequencies, damping values, and mode shapes. Modal participation factors were identified with a least squares frequency domain curve-fitter.

During the modal parameter extraction, a lot of modes appeared to be clustered around resonance peaks. Figure 3 shows the stabilisation diagram for a narrow frequency band around 125 Hz. The function on this diagram is an average of all available FRFs. Although there are only four resonances apparent from the averaged FRF, a lot more modes seem to stabilise reasonably well. A similar problem occurred in other frequency bands too.

Investigation of the individual FRFs around 125 Hz reveals a significant difference for adjacent points. The 125 Hz mode is a bending mode of the beam profile to attach the dashboard on. Figure 4 shows 2 functions. The solid line is the averaged FRF for the 3 DOFs of the centre node on this profile. The dashed line is the averaged FRF for the DOFs at both adjacent nodes, each only 40 cm away from the centre. The clear resonance frequency shift of 1.7 Hz is due to the mass loading effect. Adding the combined mass of 36 grams of the triaxial accelerometer on a dynamically flexible point such as the centre of this profile drops its resonance frequency significantly enough to make it show as another mode in the stabilisation diagram.

The frequency range between 20 and 160 Hz was analysed in several sub-bands. In total, 22 modes were extracted. After elimination of the repeated modes due to the mass loading effect described above, the final experimental modal model contains 16 modes, listed in Table 1.

Mode nr	Frequency . (Hz)	Damping (%)	Description
1	39.3	2.10	First torsion
2	45.0	1.21	First vertical bending
3	50.4	2.04	Skew vertical bending
4	77.1	0.52	First horizontal bending
5	81.0	1.01	Second torsion
6	89.0	0.63	Second vertical bending
7	101.9	0.35	Front torsion/rear bend
8	122.0	0.29	Third torsion
9	123.7	0.29	Local front panel mode
10	125.7	0.23	Third vertical bending
11	131.4	0.29	Floor membrane mode
12	132.8	0.24	Second horiz bending
13	145.4	0.31	Rear roll bar bending
14	152.7	0.20	Top/bottom shear
15	158.6	0.31	Front roll bar bending
16	160.4	0.19	First floor bending

TABLE 1: Experimental modes

From these results, it is immediately clear that the use of the space frame technology offers an important advantage for the dynamic behaviour. The first flexible mode is located near 40 Hz. This is exceptionally high for a cabriolet-type car, which usually has the first torsion of its car body-in-white at or even below 20 Hz.

CORRELATION WITH THE FE MODEL

An MSC/NASTRAN FE model was built for the dynamic behaviour of the car body. This model consisted of 26000 degrees of freedom. 4374 quadrilateral (CQUAD4) elements were used, as well as 1638 triangular (CTRIA3). A correlation of the experimental and analytical geometrical and modal models was carried out with the LMS Link software.

Figure 5 shows the MSC/NASTRAN model as described above. This geometrical model with more than 4000 nodes was correlated with the experimental modal test wire frame model with only 94 nodes.

Table 2 lists the FE modes obtained with a SOL 103 MSC/NASTRAN run and correlates them to the experimental modes.

Mode nr	Exp.. Frequency . (Hz)	FE Frequency (Hz)	Difference . (Hz)	Difference . (%)	MAC
1	39.3	55.2	9.8	18	0.95
2	45.0	54.8	15.5	28	0.94
3	50.4	-	-	-	-
4	77.1	88.0	10.9	12	0.54
5	81.0	98.9	18.6	19	0.64
6	89.0	102.5	13.8	14	0.84
7	101.9	117.2	15.5	13	0.69
8(*)	-	121.9	-	-	-
9(*)	-	122.4	-	-	-
10(*)	-	128.2	-	-	-
11(*)	-	130.1	-	-	-
12(*)	-	131.0	-	-	-
13	122.0	142.8	20.8	15	0.68

TABLE 2: Experimental and analytical mode correlation

On average, the FE frequencies are about 15 % too high. Nevertheless the mode shape correlation is particularly high for the first 2 modes (first forsion and bending modes) with a MAC value of 94%. Figure 6 shows a correlated mode pair.

Mode 3 is not predicted in the FE model. This mode is an asymmetric bending mode while the FE model was set up symmetrically.

The modes indicated by (*) do not correlate at all. Mode 8 is a membrane mode of the floor panel, and is not very well excited in the experimental test. Modes 9, 10 and 12 have their deformations in a purely horizontal plane and were not at all excited by the 2 vertically positioned shakers during the experiment. A test set up configuration with only vertical excitation does normally not pose any practical controllability problem for a car body-in-white. In this particular case though, some modes exhibit only defection in the horizontal plane (as shown in Figure 7). This can be explained by the fact that the rectangular shapes of the aluminium profiles have 2 explicit directions of preferred deflection (purely vertical and purely horizontal), while ordinary car body frames have most of their modes in a combined direction. This is certainly something to bear in mind for future modal tests on a similar structure.

Finally, mode 11 is a nice bending mode (Figure 8), but happens to have nodal points with hardly any deformation at the 2 exciter positions. Again a pre-test analysis based on the FE model results would have avoided the unlucky choice of the exciter positions and directions. [1]

Table 2 further shows low MAC values for other modes too. Visual comparison of these mode shapes shows that a low MAC value does not necessarily correspond to a poor correlation. As an illustration the low MAC value of 0.54 between the experimental mode 4 at 77 Hz and the FE mode at 88.0 Hz (Figure 9) is studied in more detail. A MAC variation technique (discussed in Ref [3]) revealed that an improved MAC value of 0.70 can be obtained by excluding only 5 procent of the motion (Figure 10) in the correlation analysis. This confirms the fact that local differences in dynamic behaviour can strongly influence correlation measures such as MAC.

ERROR LOCALIZATION

The correlation analysis revealed significant differences between measured and predicted modal parameters. This mismatch is partially due to the fact that the design model on which the FE model was based was altered before the prototype car body was built. The design modifications resulted in an increased total weight.

In such a situation it is questionable whether error localization methods can provide any indication on these non-conform model "errors" as they can only make changes to physical parameters in the FE model.

The used error localization method is based on a sensitivity formulation and described in Ref [3]) :

$$\{f_t\} = \{f_s\} + \left[S(\{p_s\})\right]\{dp\} \qquad (1)$$

where

$\{f_t\}$ represents the desired response state : the experimental frequencies

$\{f_s\}$ represents the current response state : the FE frequencies

$\left[S(\{p_s\})\right]$ is the matrix of sensitivity coefficients of the FE frequencies w.r.t. changes in thickness values

$\{dp\}$ represents the current state of the model parameters : thickness values of the shell elements

The frequency sensitivity matrix was evaluated by MSC/NASTRAN using the SOL 200 option (Figure 11). Thickness values on 51 PSHELL cards were selected as model parameters.

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[1] At the time of the experiment, the FE results were not yet available to show this particular behaviour.

Equation (1) is solved for parameter updates $\{dp\}$ using a Bayesian parameter estimation scheme that allows the analyst to express his confidence in each of the thickness values and in the target (experimental) frequencies.

The frequency sensitivity equation was complemented with a weight sensitivity equation to reduce the difference between the actual weight of the prototype car body (85.0 kg) and the weight computed from the FE model (80.9 kg).

The changes that were made to the original design are additional extruded aluminium parts that were welded to the frame. Assuming that these design changes will only have a mass effect, the error localization method was fed with following constraints :

- high confidence factors on model parameters with positive frequency sensitivity coefficients (stiffness effect). Actually the amount of thickness variation was limited to 10 procent.

- low confidence factors on model parameters with negative frequency sensitivity coefficients (mass effect). No limitations were put on thickness changes.

- locate the five most dominant "errors".

- assign high importance to the weight equation. (This is realised by having a weighting coefficient on the frequency sensitivity equations that is lower than the one on the weight equation).

Results of the error localization method are summarized in Table 3.

PSHELL nr	Weight change due to thickness variation (kg)
1170	2.95
1007	1.18
15	0.31
2000	-0.18
301	-0.18

TABLE 3: Located mass changes

The located error regions are visualized in Figure 12.

When comparing these results with the mass changes due to the actual design modification, visualized in Figure 13, it is clear that not all mass changes are predicted in an accurate way but that the most dominant errors are identified. The fact that not all mass modifications were predicted is due to the fact that possibly in these regions no mass effect could be simulated by an increased thickness value and / or the fact that the amount of not predicted mass change is small compared to the overall mass change.

Estimates for the new FE frequencies can be obtained by substituting the model parameter updates in Equation (1). In order to verify the quality of these new estimates a new MSC/NASTRAN run was performed with the new model parameters. Table 4 and Figure 14 summarize the results :

Mode nr	**Exp Frequency (Hz)**	**FE Frequency based on Eq. (1) (Hz)**	**FE Frequency MSC/NASTRAN re-analysis (Hz)**	**Difference (%)**
1	39.3	52.9	52.6	25
2	45.0	45.8	47.2	5
4	77.1	83.1	80.4	4
5	81.0	88.3	85.7	5
6	89.0	86.4	84.3	-6
7	101.9	98.6	100.0	-2
13	122.0	125.9	126.0	3
	Measured Weight (kg)	**Weight based on Eq. (1) (kg)**	**Weight MSC/NASTRAN re-analysis (kg)**	**Difference (kg)**
-	85.0	85.0	85.0	0

TABLE 4: Experimental and FE frequency comparison

It should be stressed that the re-analysis run with MSC/NASTRAN was only intended to verify whether the located mass errors would indeed improve the resonance frequencies and weight of the modelled structure. The modified model can not be considered as an improved model since the applied changes are physically not meaningful.

As can be observed from Table 4 all frequencies are aligned except for the first mode (torsion mode) where no significant improvement is found. Before any updating technique can improve this situation the FE model should incorporate the modifications that were made to the design. This topic however goes beyond the scope of this paper.

CONCLUSIONS

This paper described an experimental and analytical modal analysis of the body of the Ethos concept car as designed by Pininfarina.

Specific difficulties such as mass loading effects due to accelerometer attachment to such a light-weighted structure were emphasised for the experimental modelling.

The correlation of the analytical model with the experimental one gave an indication of the importance of a pre-test analysis to define proper excitation locations for the experimental modal test. An error localizaton method pointed out where and how the FE model should be modified to get closer in agreement with the test results. Further improvement is certainly possible, but goes beyond the scope of this paper.

ACKNOWLEDGEMENT

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[2] Lembregts F., "Frequency Domain Identification Techniques for Experimental Multiple Input Modal Analysis", PhD Dissertation, K.U. Leuven, Belgium, 1988.

[3].Brughmans M., Leuridan J. and Blauwkamp K, "The Application of FEM-EMA Correlation and Validation Techniques on a Body-in-White", Proceedings 1993 MSC World Users' Conference, 1993..

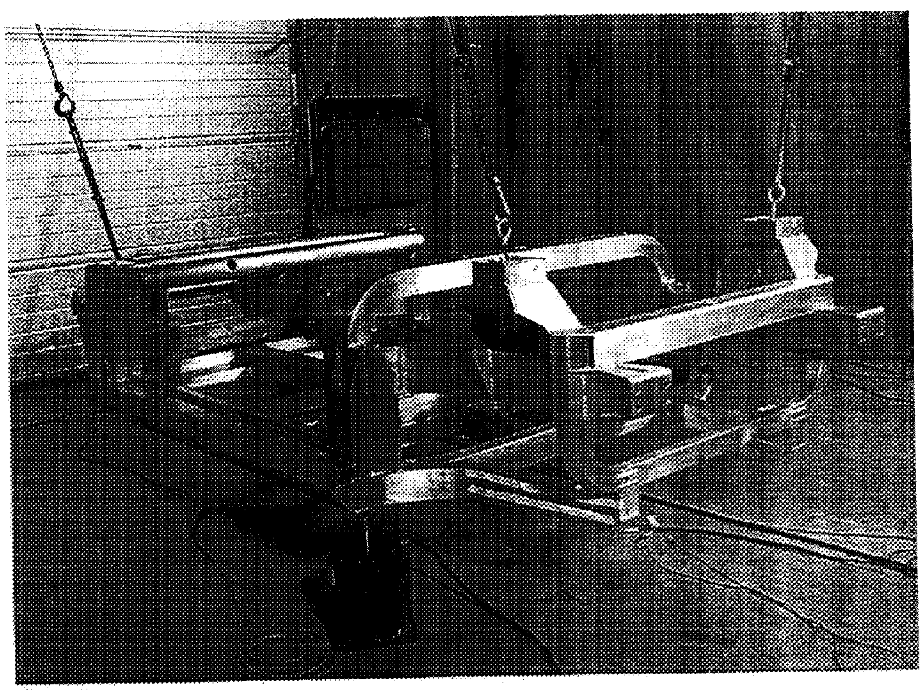

Figure 1 : The ETHOS car body

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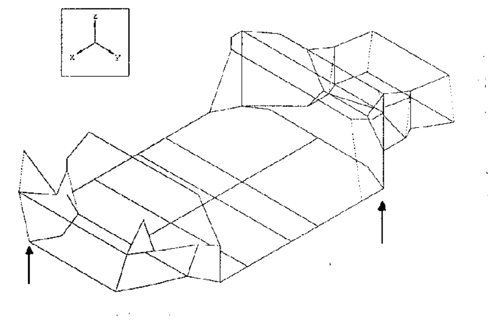

Figure 2 : Wire frame model with exciter locations

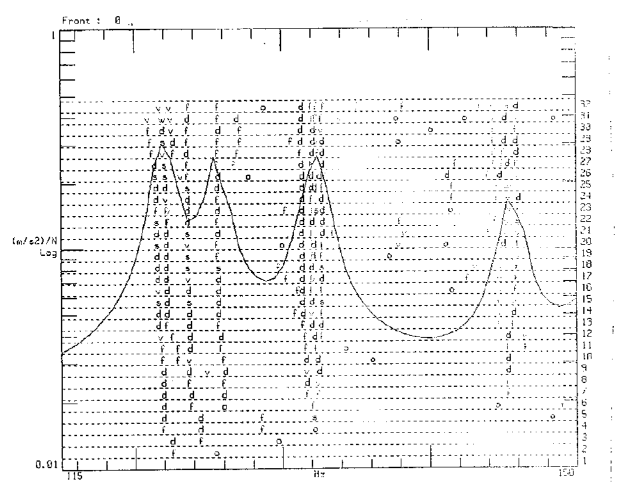

Figure 3 : Stabilisation diagram near 125 Hz

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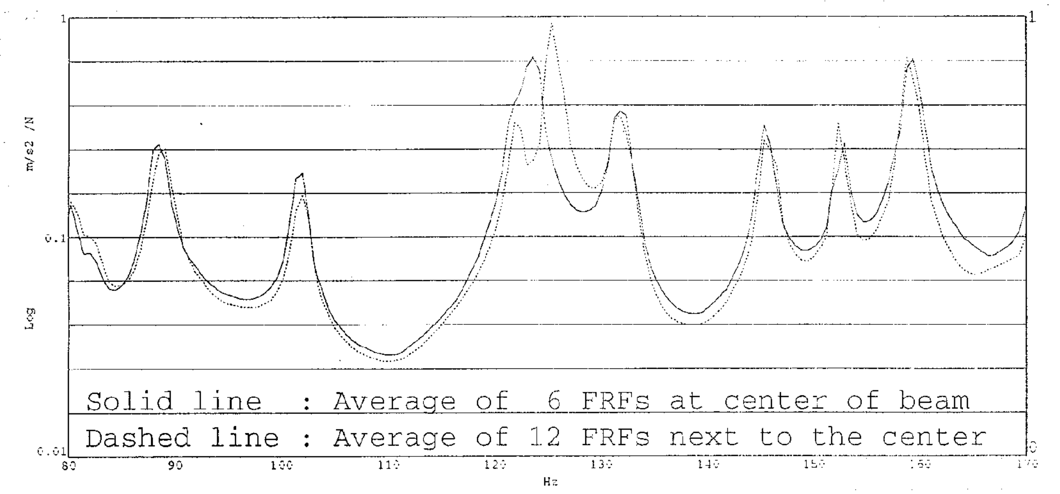

Figure 4 : Mass loading by accelerometer

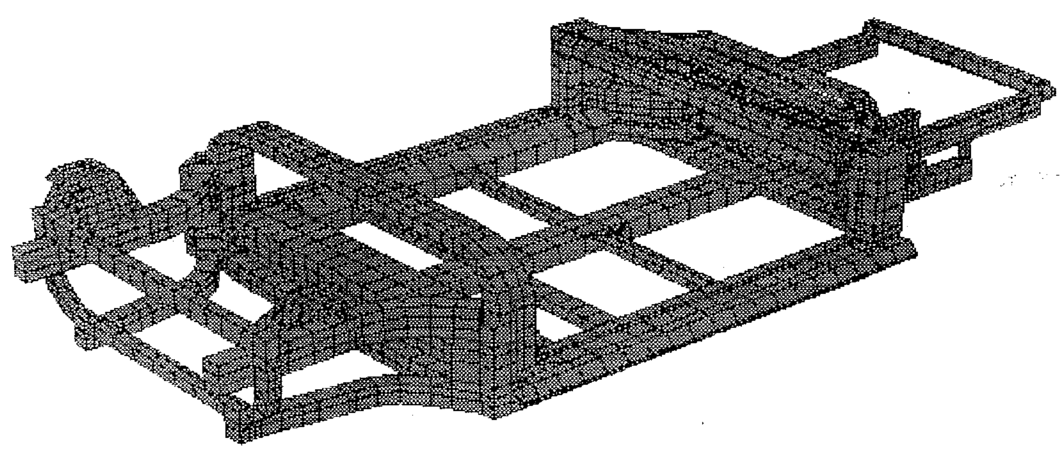

Figure 5 : Finiite Element Model MSC/NASTRAN

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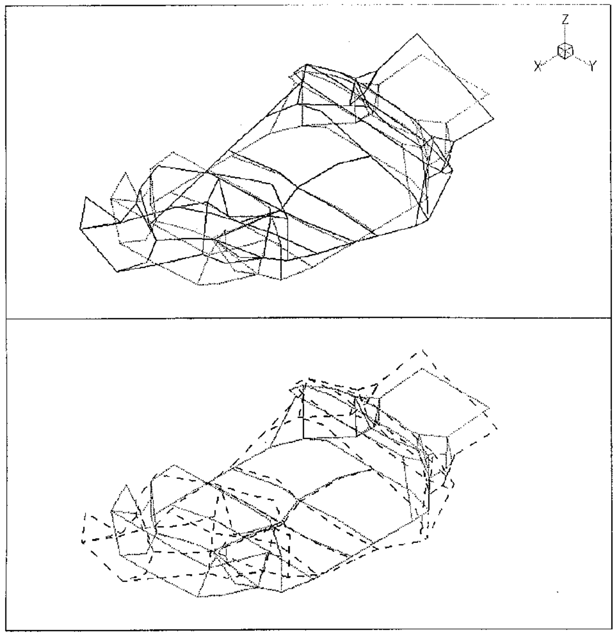

Figure 6 : Correlated Mode Pair for first torsion mode
(undeformed model in dots, FE mode in solid line, test mode in dashed line)

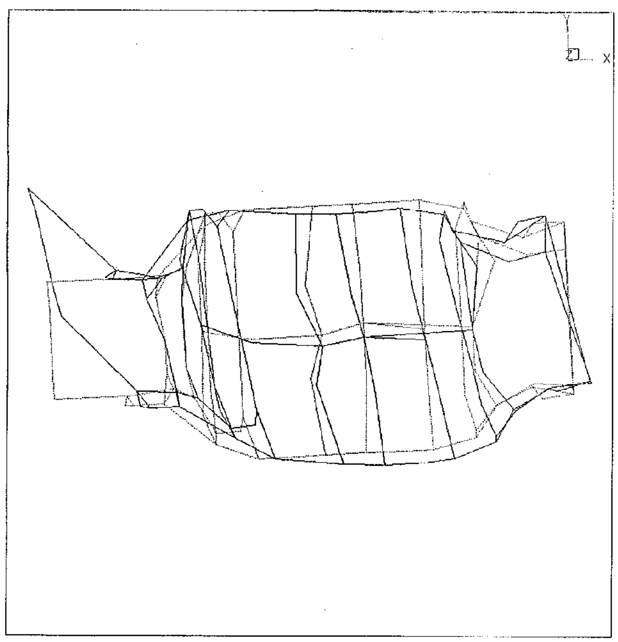

Figure 7 : FE mode in the horizontzal plane

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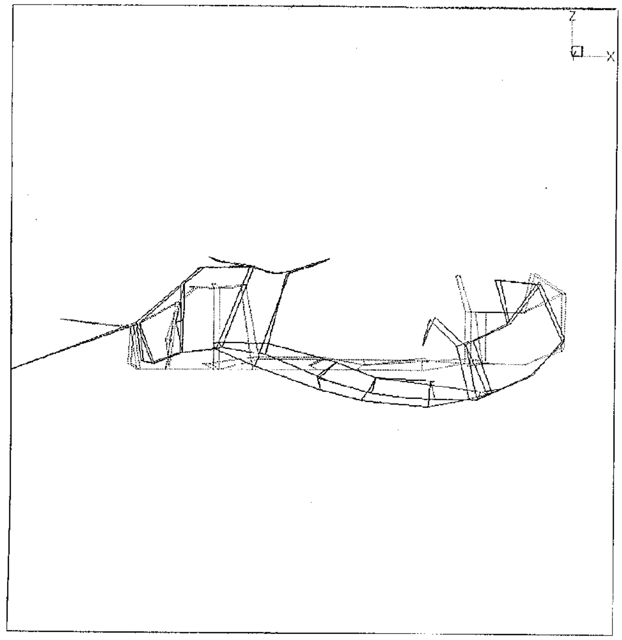

Figure 8 : FE mode at 130.1 Hz, poorly excited

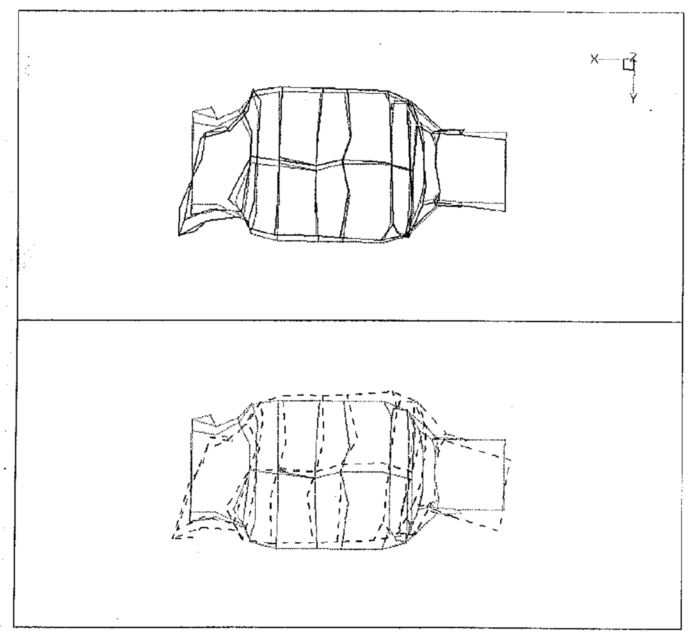

Figure 9 : Low MAC value 0.54 for FE mode (top) at 88 Hz.

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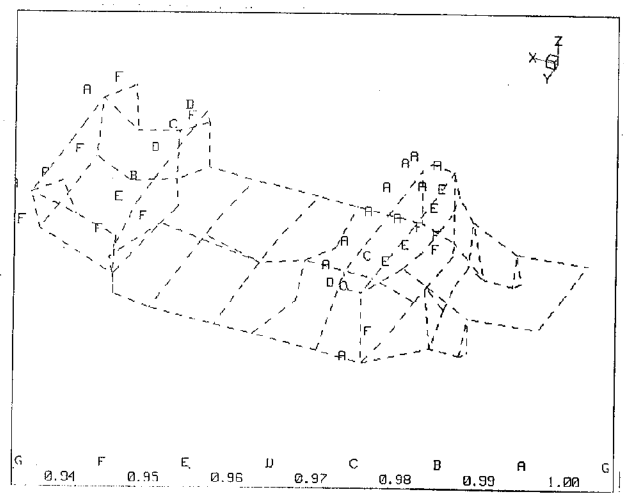

Figure 10 : FE dofs excluded from MAC analysis

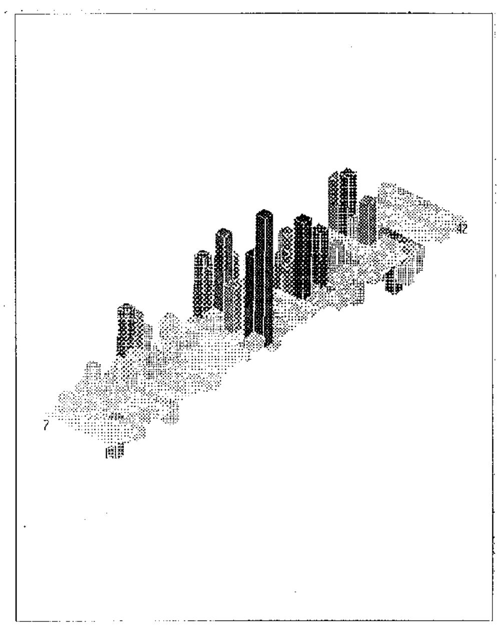

Figure 11 : MSC/NASTRAN sensitivity matrix (SOL200)

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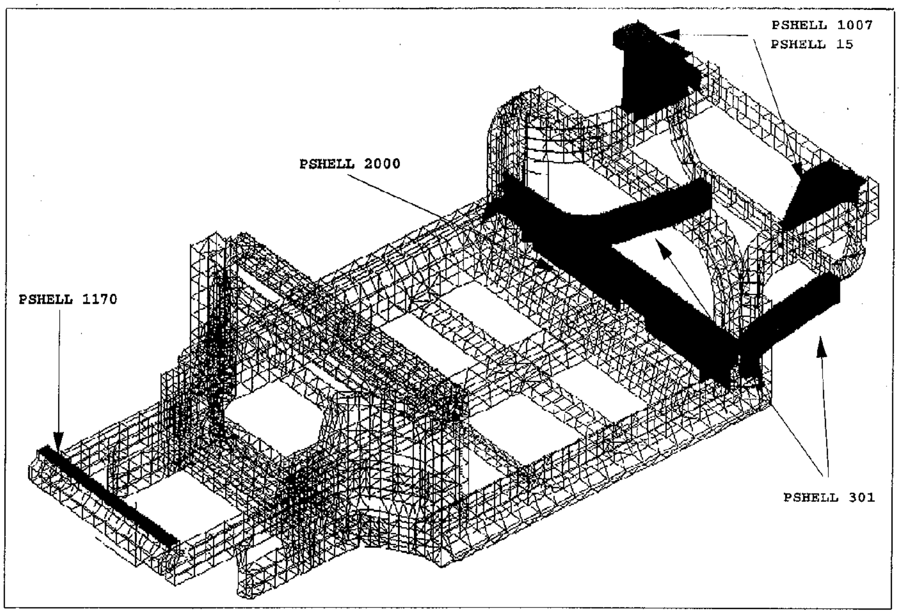

Figure 12 : Located mass changes

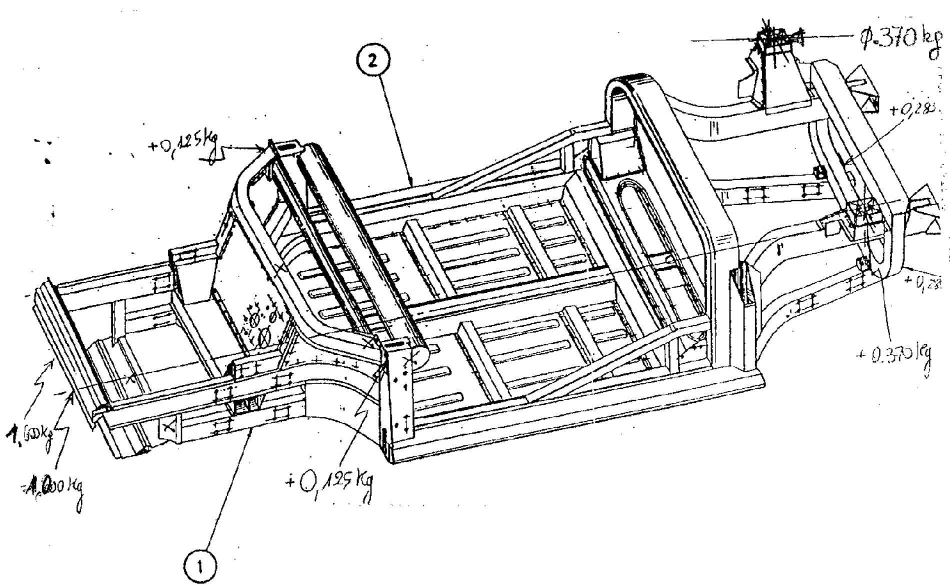

Figure 13 : Mass design modifications

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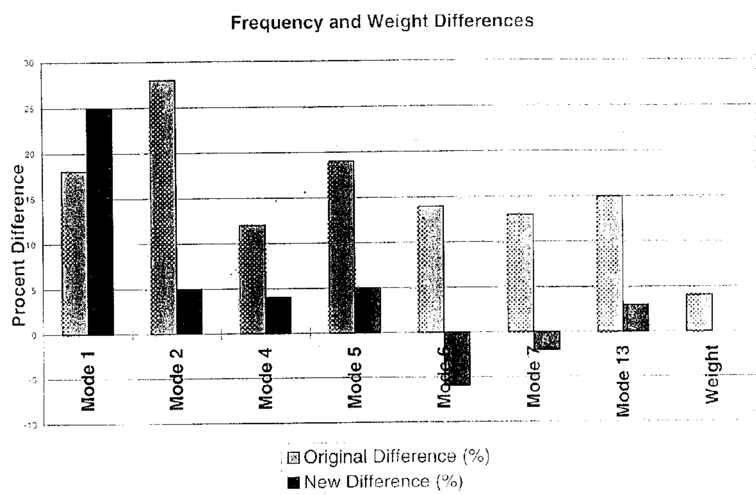

Figure 15 : Frequency and weight differences

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