The 3rd International Conference on
″Computational Mechanics
and Virtual Engineering″
COMEC 2009
29 – 30 OCTOBER 2009, Brasov, Romania

FINITE ELEMENT MODEL UPDATING OF AN ENGINE IN THE AUTOMOTIVE INDUSTRY

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Abstract:*Nowadays, due to the market concurrence in the automotive field and the high customers standards, the acoustic comfort have become the one of the most important engineering require.*
This paper deals with the NVH correlation test-calculation and the finite element (FE) model updating of an engine for automotive. The main objective for this approach is to obtain a more predictive FE model (modal frequencies, modal shapes), in order to improve the vibration behavior for this engine. Experimental and theoretical analysis used for this work, have allowed us to understand the real vibratory behavior and to obtain a new FE model more closed by reality. The final updating results are presented and compared within paper.
Keywords: finite element, automotive engine, NVH, modal analysis, updating.

1. INTRODUCTION

Acoustic comfort standards have become harder these recent years in particular for the economic vehicles. Controlling the vibratory behavior of our powertrains under the economical constraints, led us to use efficient numerical methods during all the development process of the powertrain.

The vibrations transmitted to the body car by powertrain remain the main source for low and middle frequencies. NVH numerical calculations are perform during the first steps of the development process of the powertrain. Then a first loop of prototypes allows comparing and readjusting the FE models, which will be more predictive to recommend modification for the final structure design of the power train.

From measurements of instantaneous pressure in cylinders engine, a first calculation is perform to determine efforts injected by the crankshaft on bearing and of the barrels of the cylinder block as well as efforts injected on the combustion chamber. Then a modal frequency response is calculated with these excitations on the FE model to estimate the accelerations at the engine mounting points.

When the first prototype engine components are available, a correlation between the calculated and experimental modal bases and as well an updating of the models have to be perform to ensure that the FE models will be predictive for the final recommendations to the design office. This step of updating is even more important when is relate to components like the cylinder block, the cylinder head or the oil pan, which will be identical for a whole generation of engine on various vehicles applications

In this case, we will describe methodology used to carry out a correlation between modal bases resulting from measurement and calculation and to update the FE model of the base engine structure of the 1,6L gasoline engine that is mounted on the all Dacia vehicles.

2. EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis is the process of determining the modal parameters (frequencies, damping factors, modal vectors and modal scaling) of a linear, time invariant system by way of an experimental approach. The modal parameters may be determined by analytical means, such as finite element analysis, and one of the common reasons for experimental modal analysis is the verification/correction of the results of the analytical approach (model updating).

The steps in performing Experimental Modal Analysis using Impact Testing method and Roving Tri-axial accelerometer technique are presented in Figure 1.

The following equipment is required to perform an impact test:

1. An ***impact hammer*** with a load cell attached to its head to measure the input force.
2. An ***accelerometer*** to measure the acceleration response at a fixed point and direction.
3. ***FFT analyzer*** to compute FRFs.
4. ***Post-processing modal software*** for identifying modal parameters and displaying the mode shapes in animation.

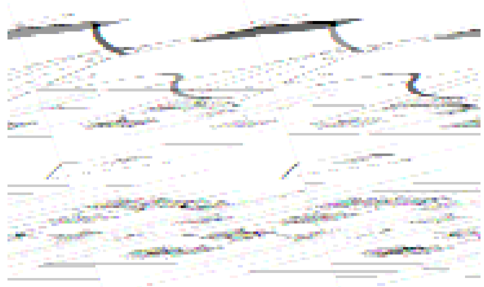

Figure 1: Impact Testing

In order to have high quality measurements and confident results from Experimental Modal Analysis (EMA) we follow the steps presented in the Figure 2.

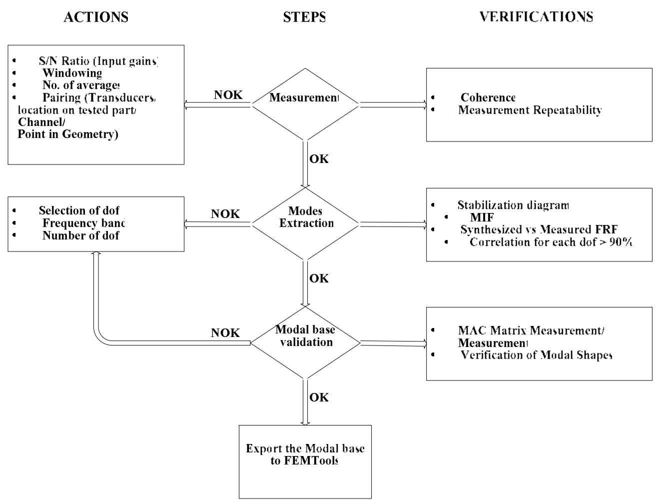

Figure 2: Steps in performing EMA

To check quality of modes extraction we perform a statistical analysis in order to verify the correlation between the synthesis and all the measured transfer functions (see Figure 3). The correlation is the normalized complex product of the synthesized and measured values.

$$correlation = \frac{\left|\sum_i (S_i \times M_i^*)\right|^2}{\left(\sum_i (S_i \times S_i^*)\right)\left(\sum_i (M_i \times M_i^*)\right)} \tag{1}$$

where,

S_i = the complex value of the synthesized FRF at spectral line i

M_i = the complex value of the measured FRF at spectral line i.

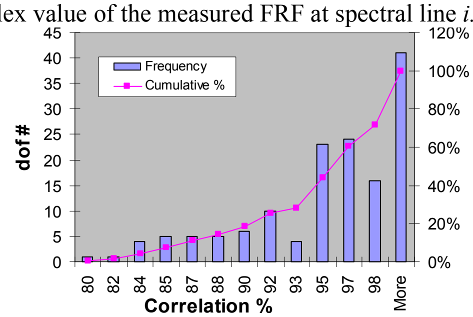

Figure 3: Correlation Synthesized/Measured

The high values of the correlation between synthesized and measured (Figure 3) indicate that the Modal Base is complete.

For the modal base validation, a Mode Assurance Criterion (MAC) (2) matrix measurement/ measurement is performed which compare the **extracted modal base** with itself in order to verify if the extracted modes are well-differentiated one from each other (Figure 4).

The MAC Matrix must denote a satisfactory orthogonally of the modal parameters, which permit to identify and distinguish the modes. The values of non – diagonal terms indicate a good modal extraction.

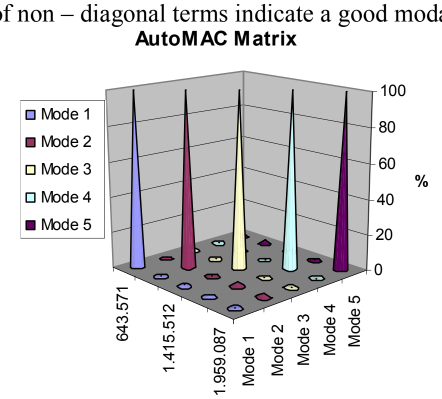

Figure 4: MAC Matrix Measurement/Measurement

Another method of modal model validation that we use too is to evaluate the modal vectors visually. While this can be accomplished from plotted modal vectors superimposed upon the undeformed geometry, the modal vectors are normally animated (superimposed upon the undeformed geometry) in order to quickly assess the modal vector.

Particularly, the modal vectors are evaluated for physically realizable characteristics such as discontinuous motion or out-of-phase problems. Often, rigid body modes of vibration are evaluated to determine scaling (calibration) errors or invalid measurement degree-of-freedom assignment or orientation. Naturally, if the system under test is believed to be proportionally damped, the modal vectors should be normal modes and this characteristic can be quickly observed by viewing an animation of the modal vector .

3. MODAL ANALYSIS AND CORRELATION TEST- CALCULATION

3.1 Description of the updating process

A finite – element (FE) model was created for an automotive engine in order to be studied from NVH (Noise Vibration Harshness) point of view. The objective is to provide a predictive FE model more closed by the real model. Due to this fact, in the forward calculations it is considered for the updating the base structure of the engine that contains: the engine block, the oil pan, the cylinders head and the cylinders head cover.

From the calculation point of view, in order to update the FE model is necessary to follow the next steps: modeling of the the FE model, correlation test-calculation models, sensitivity analysis and the updating analysis when it is necessary. This process is presented in the figure 1.

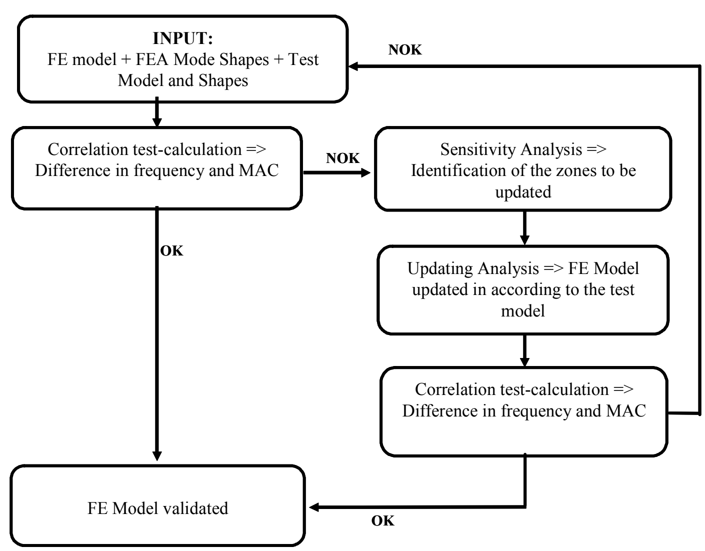

Figure 6: Model Updating Flow Chart

3.2 Description of the FE model

The finite element model of the base structure is composed by 4 parts assembled with 230000 elements. The types used for meshing are volume 2nd order and shell.

For the cylinder head, the engine block and the oil pan the mesh was performed by volume element.

The cover of the cylinder head is performed by mixed mesh (volume and shell). The links between the parts of the assembly are performed by the rigid element [2].

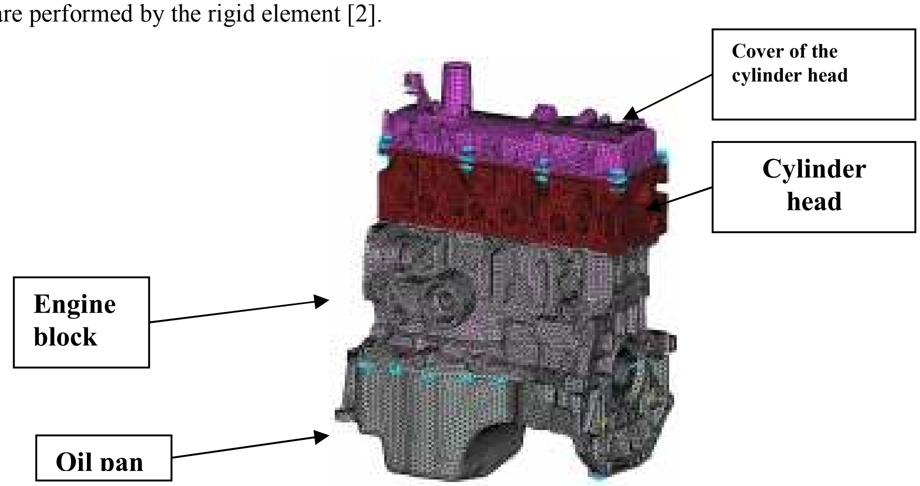

Figure 7: The finite element model of the base structure

3.3 Correlation Analysis

Firstly, it was analyzed each part free-free of the assembly in order to calculate the Diff. (the difference in frequency between the test and calculation modes) and the MAC (Modal Assurance Criterion). The advantage of using the MAC is given by the fact, that it is not required the stiffness and the mass matrices. The value 1 (or 100%) for MAC indicate two identical modes [3].

$$MAC(\Psi_a, \Psi_e) = \frac{\left|\left(\{\Psi_a\}^t \{\Psi_e\}\right)\right|^2}{\left(\{\Psi_a\}^t \{\Psi_a\}\right)\left(\{\Psi_e\}^t \{\Psi_e\}\right)} \tag{2}$$

where:

Ψ_a is the analytical eigenvector;

Ψe is the experimental eigenvector.

The results for a first comparison test – calculation are presented bellow in the next tables.

Table 1 – Correlation test-calculation of the part free-free, before updating

	FEA	Hz	EMA	Hz	Diff. [%]	MAC [%]
Cylinder head	7	1555	1	1484	4.75	93.2
	8	1863	2	1770	5.26	95.3
	9	2451	3	2324	5.48	91.7
Engine block	7	623	1	643	-3.22	94.9
	8	1170	2	1216	-3.79	95.8
	9	1383	3	1415	-2.27	90.8
	10	1686	4	1740	-3.10	89.9
	11	1871	5	1958	-4.43	95.1
Oil pan	7	166	1	158	5.14	84.7
	8	451	2	437	3.28	79.3
	9	550	3	537	2.43	90.1
	10	679	4	661	2.78	80.3

Then, we perform a first comparison test – calculation for the assembly of the base structure.

Table 2 – Correlation test-calculation of the assembly of the base structure, after updating

	FEA	Hz	EMA	Hz	Diff. [%]	MAC [%]
1	7	896	1	967	-7.39	35.8
2	8	916	2	814	12.61	61.6
3	10	1052	3	1012	3.93	81.8
4	14	1208	4	1029	17.45	24.4
5	15	1226	5	1102	11.21	73.3

As we can notice in the Table 4, we have obtained some unacceptables values for the 3 modes (EMA: 967Hz/814Hz/1029Hz), concerning the difference in frequency and the correlation (MAC). These modes are determined by a local behavior of the superior part of the assembly (the bracket of the cover of the cylinders head).

In order to improve the Diff and the MAC we propose four modifications of the finite element model: the updating in frequency of the cylinders head, the engine block and the oil pan, increasing of the rigidity on the contact zone of the bracket of the cylinders head cover, the adding of the mass of the measurement sensor on the bracket and the changing of the links between the oil pan and the engine block.

3.4 Updating Analysis

In order to perform the model updating, we use a dedicated software having an iterative model updating method based on a sensitivity formulation [3].

$$\{\Delta R\} = [S]\{\Delta P\} \tag{3}$$

where: $\{\Delta R\} = \{R_e\} - \{R_a\}$; $\{R_e\}$ is a vector of the experimental data considered like reference in the updating analysis and $\{R_a\}$is a vector of the predicted system responses;

$\{\Delta P\} = \{P_u\} - \{P_o\}$; $\{P_u\}$is a vector of the updated parameter values and $\{P_o\}$is a vector containing the initial parameter values;

[S] is the sensitivity matrix given by formula:

$$[S] = S_{ij} = \frac{\delta R_i}{\delta P_j} \tag{4}$$

The updated values of parameters P are obtained from (2) and (3):

$$\{P\} = \{P_o\} + [G](\{R_e\} - \{R_a\}) \tag{5}$$

where [G] is the gain matrix computed using a Bayesian estimation theory.

In according to the theoretical aspects presented above we intend to decrease the frequency difference between the two models, experimental and FE, for each part of the assembly. For this, we use the CAE specialize software, setting the responses in frequency and the parameters to be changed within the updating analysis.

The results as follow of the update analysis, for each single part of the assembly, are given bellow in the table 3 and figures 8, 9 and 10.

Table 3 – Correlation test-calculation of the part free-free, after updating

	FEA	Hz	EMA	Hz	Diff. [%]	MAC [%]
Cylinder head	7	1480	1	1484	-0.26	92.2
	8	1774	2	1770	0.23	95.3
	9	2334	3	2324	0.43	85.6
Engine block	7	643	1	643	-0.07	94.9
	8	1208	2	1216	-0.66	95.8
	9	1428	3	1415	0.91	90.8
	10	1741	4	1740	0.05	89.9
	11	1932	5	1958	-1.33	95.1
Oil pan	7	161	1	158	2.37	84.7
	8	439	2	437	0.54	79.4
	9	537	3	537	0.04	90.1
	10	662	4	661	0.14	80.1

Also, like a general view over the sensible zones of some parts resulted from the updating analysis it is usefully to read the color maps. Understanding in detail the vibratory behavior, we can recommend to the suppliers and design engineers, the modifications necessary to improve the technical definition.

For the cylinder head, in the figure 8 the blue zones represent the most sensible local elements of the FE model compared with the EMA model. Also, we can conclude that the middle of the part could be ameliorated from the technical definition point of view.

For the oil pan, in the figure 9 the blue zones represent the most sensible local elements of the FE model compared with the EMA model. Also, we can conclude that the middle of the part could be ameliorated from the technical definition point of view.

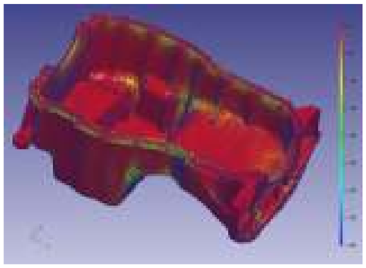

Figure 9: Parameter Change - E

Figure 8: Parameter Change - E

Finally, after the updating of the each single part of the assembly, we built a new base structure like FE model and we animate the results with successive iterations and correlations. The final results for the base structure are presented below in the Table 4.

Table 4 – Correlation test-calculation of the base structure, at the end of the updating analysis

	FEA	Hz	EMA	Hz	Diff. [%]	MAC [%]
1	8	839	1	814	3.15	94.1
2	9	1015	2	1013	0.20	88.9
3	10	1075	3	1102	-2.50	82.7

4. CONCLUSIONS

Due to the updating of the FE model, by successive iterations we have succeeded to obtain a good correlation test-calculation in frequency and modal shape for the base structure of the studied engine in this paper. In the table 5 we synthesize these results.

Table 5 – Comparison of the base structures, before and after the updating

Modal Shape	Diff. [%]		MAC [%]	
	before	after	before	after
Vertical Torsion	12.61	3.15	61.6	94.1
Horizontal Bending	3.93	0.20	81.8	88.9
Horizontal Torsion	11.21	-2.50	73.3	82.7

The updating method presented in this paper give us a more confidence in the results as follow of our modal and updating analysis and finally it helps us to increase the quality of our NVH calculations.

4.REFERENCES

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