Applications of Finite Element Model Updating Using Experimental Modal Data*

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A general procedure for Finite Element model updating, using experimental modal data, is briefly described and applied to some real life structural dynamics problems. The method avoids many of the problems of incompatibility and inconsistency between the experimental and analytical modal data sets, allowing flexible but automated model updating. A software program has been developed integrating the key tasks like pretest planning, correlation analysis, sensitivity analysis and model updating. Exchanges of analytical and experimental data with external data bases are accomplished by interface programs and a neutral file system.

The objective of integrating test and analysis results is to develop better product designs by bringing together the best of both worlds. Analytical models allow one to predict the behavior of a structure and to conduct parametric studies. Experimental techniques provide the necessary backup and verification data. This approach is illustrated by Figure 1, which shows the product development cycle with emphasis on structural dynamics.

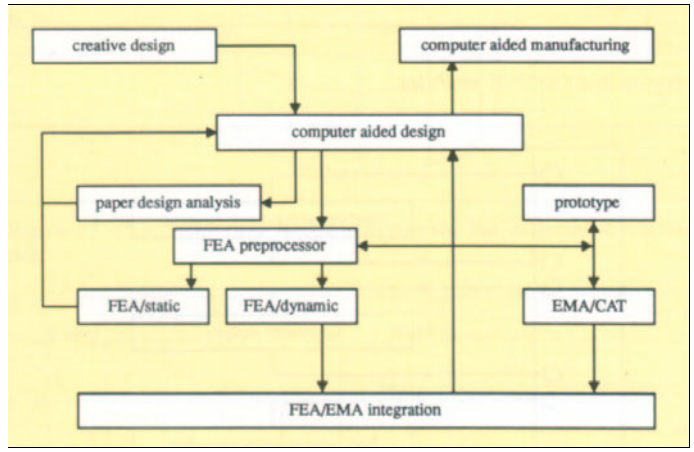

Figure 1. Product development cycle.

Analytical structural analysis methods, especially the Finite Element (FE) method have become increasingly popular. This is no surprise given the fact that FE programs are now available on the entire computer hardware platform, ranging from PCs to mainframes, and that the method has been applied to all engineering disciplines.

However, this trend holds some dangers: modeling errors, accuracy problems, program bugs or just blind faith of inexperienced engineers can lead to serious misinterpretations of analysis results. Therefore the confrontation of analysis results with experimental data should become an integral part of the design process from the very start.

During the early design stages when only a rough FE model is required, this might be excused due to lack of experimental data. However, a data base holding previously acquired results and the assistance of an expert system could improve the setup of reliable FE models.

For profound studies however, experimental verification becomes indispensable. In the field of structural dynamics, the experimental modal analysis (EMA) method offers a well accepted test solution which yields reliable values for the resonance frequencies and corresponding mode shapes.

The applications of combined FEA and EMA modeling range from pretest planning over correlation analysis and sensitivity analysis to model updating. Pretest planning includes the enhancement of test conditions by optimizing the exciter and measurement locations. Correlation analysis enables the identification of corresponding mode shapes and the establishment of relations between the analytical and experimental modal data base. Sensitivity analysis then determines the influences that mass and stiffness changes to the FE model will have on the resonance frequencies. The results of these previous analysis can be used to iteratively modify the selected FE model parameters (stiffness and/or mass related) in order to improve the correlation between test frequencies and the frequencies obtained from the updated FE model.

Integration of FEA and EMA

In order to derive maximum benefit from the combined use of test/analysis data, a system is needed that integrates all required operations. This is also necessary in order to maintain consistency among the several data types and to complete the analysis within a reasonable time span. All tools that are required have been integrated as modules of FEMtools. This program has been developed over the years by Dynamic Design Solutions and first commercially released in 1989 [1-4, 9].

FEMtools collects information from different sources, stores it in an internal data base and enables the user, by means of a high-level command language, to perform operations on the contents of the data base. All necessary functions are provided with the program. This approach allows for complete independence of external analysis packages. Figure 2 shows the different analysis modules that are currently available. The results generated by each of these modules serve as input data for the next ones. This systematic approach allows for intermediate evaluation and decision making. Graphical visualization tools assist the user in evaluating the result of a previous operation and help to decide what to do next. The complete procedure can be executed in background for continued iterations or for parametric studies. Figure 3 shows the systematic order of operations that is usually followed.

Finite element model updating is a creative, decision-based process which requires access to various analysis and diagnosis tools. This is best accomplished through an iterative environment that allows one to obtain results quickly and efficiently. Utilities are included in FEMtools for this purpose such as a powerful free-formatted command language, descriptive table, color graphics, etc.

Several current problems still limit the generalized use of combined test/analysis techniques. The fact that numerical analysis and test often use different hardware and software platforms at different locations leads one to suspect that there might be a data communication problem. This should be overcome by continuing standardization among hardware and software vendors. FEMtools includes direct interface programs with several major FE analysis codes (MSC/NASTRAN, ANSYS), EMA systems (SMS, ENTEK, LMS) or acquires data using the emerging neutral formats such as Universal File or CAD*I.

If the analytical data that are available include the element structural matrices, the integrated Lanczos solver (dynamic analysis) al-

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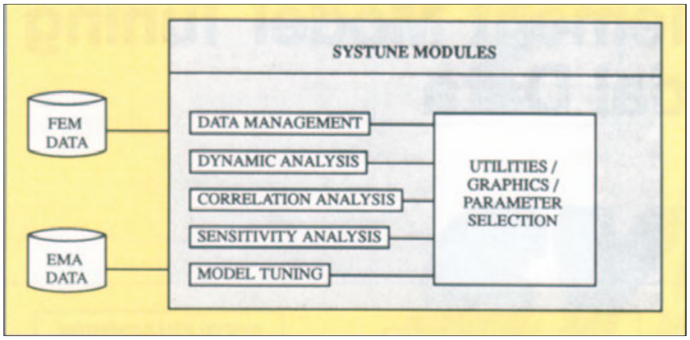

Figure 2. FEMtools modules..

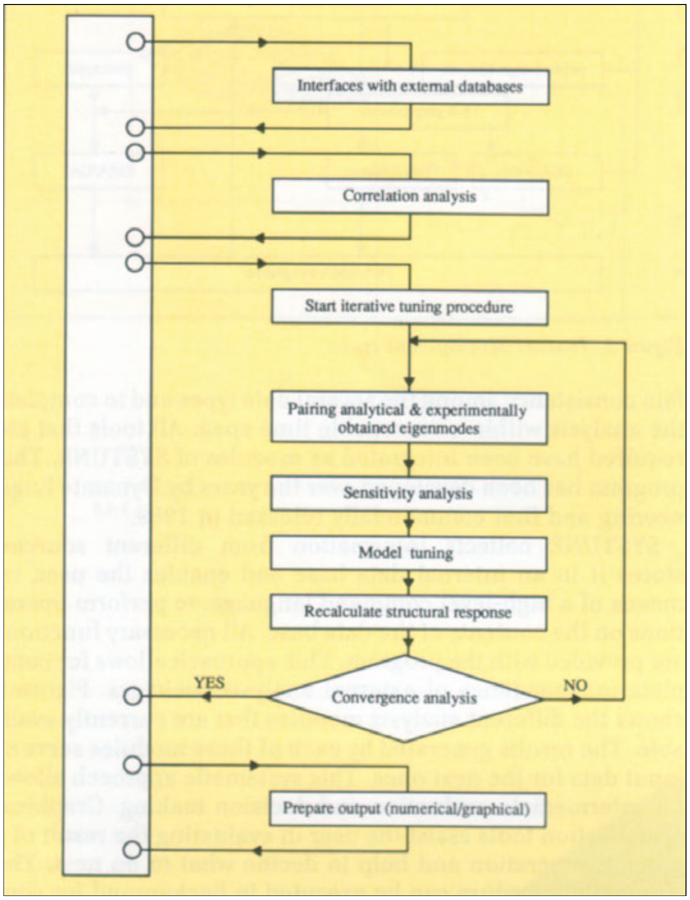

Figure 3. General updating scheme.

lows one to compute the resonance frequencies and mode shapes of the FE model. This is especially important in the iterative parameter estimation procedure (model updating) since with each iteration the modal properties of the updated model have to be recomputed and compared against the test data.

Usually the data base that is obtained from FEA is much larger than the one obtained from EMA. Moreover, both data bases may differ substantially in terms of number of degrees of freedom per node or the coordinate system that was used to describe the structure.

The basic requirement for updating is that the experimental reference data are reliable. This is generally the case for natural eigenfrequencies which can be measured within a 1% error margin. On the other hand, test estimates for mode shapes are often less reliable. The general deformation mode is likely to be reasonable (for example, when making visual comparisons), but individual values at particular measurement points may differ by 30% or more.

Results obtained from test examples show that for sensitivity analysis purposes, these experimental modal displacements can be substituted with the corresponding analytically obtained values [1, 2]. This approach eliminates the necessity to perform reduction, expansion or interpolation operations on the modal vectors in order to obtain the same number of DOF for both models. Hence, this avoids errors introduced by the expansion process or the loss of valuable information because of eigenvector reduction.

In the proposed approach, experimental mode shapes should only be used to pair experimental and analytical modes. This requires that correlation analysis results show a sufficiently high degree of correspondence between the test and analytical mode shapes.

Pretest Planning

A representative version of the FE model can be used to determine the approximate resonance frequencies that are in the frequency band of interest, their distribution and corresponding mode shapes. This examination of the mode shapes allows optimal degrees of freedom (DOF) to be used for excitation and measurement. The optimal exciter location can be found by comparing average modal displacements over all considered mode shapes for a number of candidate degrees of freedom. These average values and thus the distances to the nodal lines of the eigenmodes should be maximized.

The FE model and the eigenvectors can be reduced to a set of acceptable measurement point degrees of freedom. Correlation analysis between the reduced data set and the full FE data set allows one to examine the suitability of the selected experimental DOF. Bad correlation indicates a too limited subset of measurement DOF, resulting in incomplete estimates of the mode shapes (spatial aliasing). The reduced FE model data can be extracted to serve as the test geometry for further data acquisition and modal analysis. In addition, the influence of the transducer masses on the resonance frequencies and mode shapes can be investigated using sensitivity analysis.

Correlation Analysis

The correlation between FEA and EMA results can be expressed in terms of the resonance frequencies and the modal displacements. As indicated previously, absolute values of the modal displacements should be treated with care especially when used in a convergence criterion.

A commonly used value to compare FEA and experimental mode shapes is the Modal Assurance Criterion (MAC) [5]. It might be necessary to transform the EMA data in order to obtain compatibility with the FE data (scaling, rotations and translations). FE nodes and measurement points are then paired in order to create a set of DOF that are common to both data bases. Figure 4 shows a superimposed view of a FE model and the corresponding test geometry of an alternator hinge mount with the positions of the node/point links. In Figure 5, typical MAC-matrices are displayed before and after updating of a FE model. Other correlation tests exist (e.g. CoMAC, orthogonality analysis, mode differencing, etc) but are all based on quantitative modal displacement comparisons and are influenced by the large experimental inaccuracies of these values. It is therefore more relevant to compare the mode shape qualitatively using the graphical tools such as superimposed, side-by-side or animated viewing.

In the proposed procedure, an important step is to 'pair' experimental and analytical modes. This can be done by visual inspection or using the quantitative techniques described above. The latter allows for automation of this pairing which is particularly important when the FE modes which correspond to EMA modes do not appear in the same order. Then, during model updating, with each iteration, the pairing has to be verified since the mode shape sequence may have been changed.

Sensitivity Analysis and Model Updating

Finite element analysis becomes worthless if the several assumptions and simplifications unique to the method cannot be quantified.

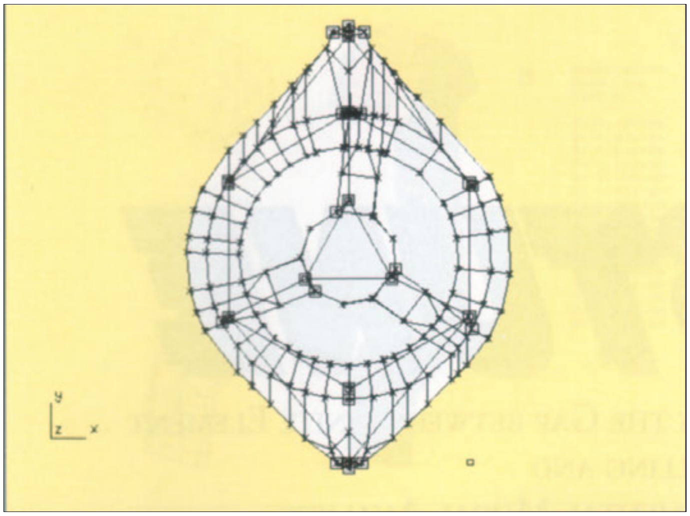

Figure 4. FE model node/measurement point pairs used for correlation analysis of an alternator hinge mount.

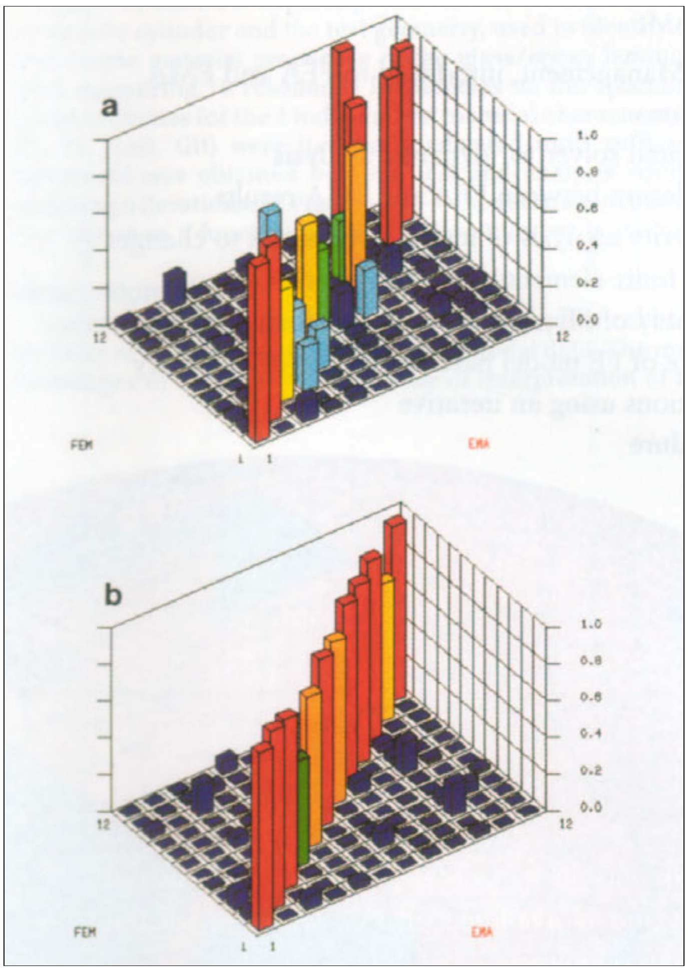

Figure 5. Correlation analysis results (MAC matrix) before (a) and after (b) updating.

Assumptions often encountered in structural dynamics practice include: approximate values for nodal constraints, joint stiffness, lumped mass properties and composite material properties.

Since a FE model usually consists of numerous parameters, making an optimum selection that will be modified to obtain better correlation between EMA and FEA is a most difficult task. FEMtools allows one to easily compare results from different parameter selections and provides tools to assist in the selection process. It is important to use design parameters only, in order to maintain physical insight of the updating process.

Parameters can be either proportional to the model matrices (e.g. Young's modulus and mass density) or nonproportional (e.g. plate thickness). They may be tuned globally or locally. Global parameter changes are typically systematic errors, for example a 2% error in Young's modulus causes a scaling of the stiffness matrix and affects all modes. Local updating refers to the individual modification of parameters associated with finite elements (material or geometrical properties) or nodes (lumped masses or elastic constraints). They may relate to simplifications used in the FE model (e.g. choosing an effective stiffness for a beam with a complex end joint) or to estimations of badly known values (e.g. composite material properties).

Sensitivities indicate how a response value (e.g. resonance frequency) is influenced by a modification of a parameter value [3]. These values are stored in a sensitivity matrix. Analyzing this matrix provides information on the sensitive and insensitive zones of the structure. Parameters for which the response value shows little sensitivity should not be selected. Also the general condition of the sensitivity matrix should be analyzed. Graphical tools allow the visualization of these different zones and a fast optimization of parameter selection.

The parameter changes required to obtain correlation between measured and analytical modal data are calculated using a generalized least squares technique [6]. Facilities to maintain the parameter between physical bounds as well as to express confidence in parameter values and experimental reference data are provided [7]. The resulting parameter changes are used to recalculate mass and stiffness matrices yielding new resonance frequencies and eigenvectors which match the experimental values more closely. An iterative process can then be continued until a convergence criterion is satisfied.

Applications

The first example illustrates the use of the program to handle the problem of approximating unknown material properties when modeling composite bicycle frames. These bicycles are especially designed for their damping characteristics and lightweight construction.

An updating procedure was applied to the FE model of a wheel fork based on a series of EMA results. The construction of a valid FE model is obstructed by the difficulties in modeling the material characteristics, which in this case is a carbon/aramid/epoxy composite. Moreover the structure also consists of a metallic (aluminum) connecting rod glued inside the composite tubes that shape the fork. These tubes are filled with a medium density foam.

The construction of the FE model required several simplifications and approximations. It was entirely constructed by using general beam elements (6 DOF/node). The material properties of the composite parts were estimated from static tensile and torsion tests. This resulted in indicative values for the Young's modulus of 62.7 GPa and a Poisson's coefficient of 0.4. Other modeling uncertainties resulted from the fabrication process that permits the carbon/aramid laminate thickness to vary slightly. This laminate thickness determines the cross-sectional properties of the beam elements. The continuously varying sections were approximated by a number of beam elements with constant cross section. The mass densities of the elements were determined from the total mass of the fork, the given densities of the foam and laminate, and the approximated values of

Table 1. Test and analysis results before and after updating the FEA of a bicycle frame.

Mode #	Test Data Frequency Hz	FEA Before Updating Frequency Hz	FEA Before Updating Difference %	FEA After Updating Frequency Hz	FEA After Updating Difference %
1	138.20	139.38	0.85	138.34	0.10
2	289.30	303.13	4.78	281.75	−2.68
3	364.00	414.44	13.86	367.23	0.88
4	448.60	499.08	11.25	456.73	1.78

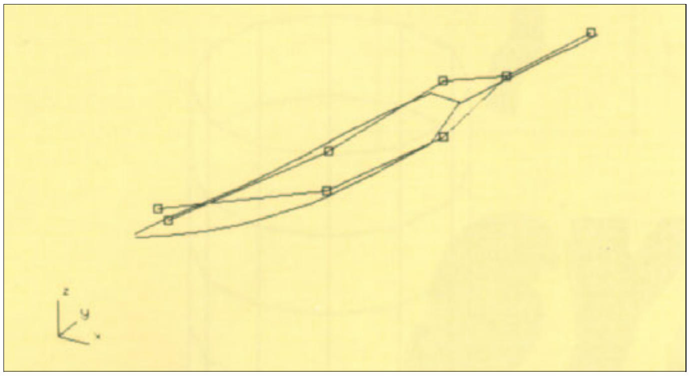

Figure 6. Superimposed view of paired mode shapes of a bicycle frame (first bending mode).

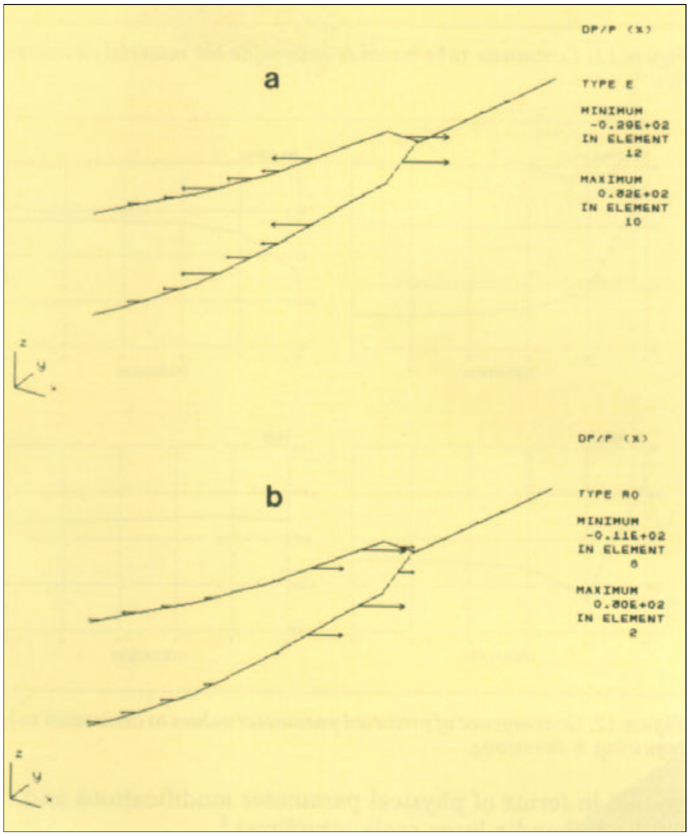

Figure 7. Required stiffness (a) and mass (b) modifications of a FE model of a bicycle frame..

the cross-sectional properties.

Tests were done on a freely suspended fork to obtain a number of resonance frequencies. In Table 1 these frequencies are compared with those predicted by the FE model. A superimposed view of the paired mode shapes is shown in Figure 6.

From the above remarks, it was concluded that the stiffness and mass densities of the beam elements modeling the composite parts of the fork were the most appropriate properties to be selected as parameters. The required parameter changes, resulting from the model updating procedure, are shown in Figure 7. The corresponding, updated frequencies (Table 1) showed a serious reduction of modeling errors. These results were obtained after 8 iterations and were based on 4 reference test frequencies. Figure 7a shows that stiffness corrections are required for the entire composite part (with a maximum of -14%). On the other hand, the mass densities (Figure 7b) were clearly underestimated for the beams, modeling the transition zone between metallic and composite parts (a maximum of +68% was required).

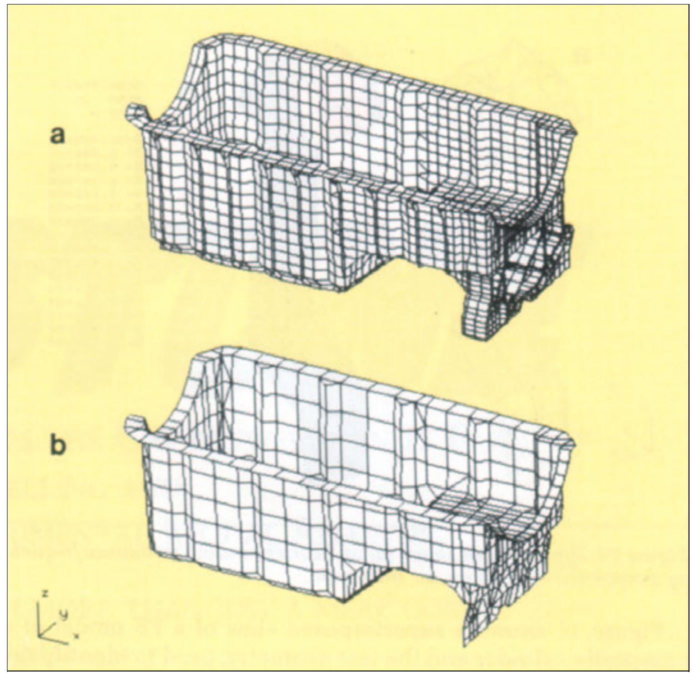

Figure 8. Original (a) and coarse (b) FE model of an oilpan.

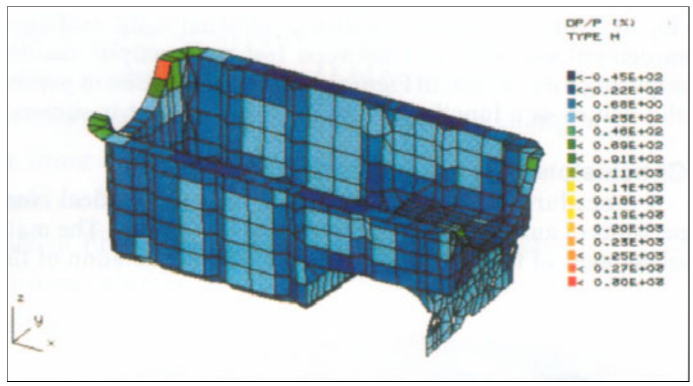

Figure 9. Shell thickness updating results to obtain a specified dynamic behavior.

Another practical problem often encountered is the modeling of very complex geometries using only a minimal number of elements while maintaining correct dynamic behavior. A simplification might be necessary to reduce calculation time or memory requirements or, because of further analysis requirements, a high density mesh is not required.

The next example discussed is an oilpan originally modeled in MSC/NASTRAN using 2988 elements (342 TRIA3, 1966 QUAD4, 564 HEXA8 and 116 PENT6) with 3252 nodes (Figure 8a). This model includes all stiffeners and represents the geometry correctly.

Table 2. Updating results for an oilpan.

Mode #	Test Data Frequency Hz	Original Coarse FEM Frequency Hz	Original Coarse FEM Difference %	Updated Coarse FEM Frequency Hz	Updated Coarse FEM Difference %
1	181.3	143.7	−20.7	172.0	−5.9
2	333.8	433.0	29.7	338.6	1.4
3	513.0	612.2	11.3	509.0	0.7
4	720.3	893.9	24.1	756.3	5.0
5	835.2	900.6	7.8	845.6	1.2
6	936.5	1058.2	13.0	924.9	−1.2
7	992.3	1167.7	17.7	992.7	0.0
8	1006.2	1219.9	21.2	1047.2	4.0
9	1118.3	1288.5	15.2	1134.2	1.4
10	1312.1	1328.7	1.2	1257.3	−4.2

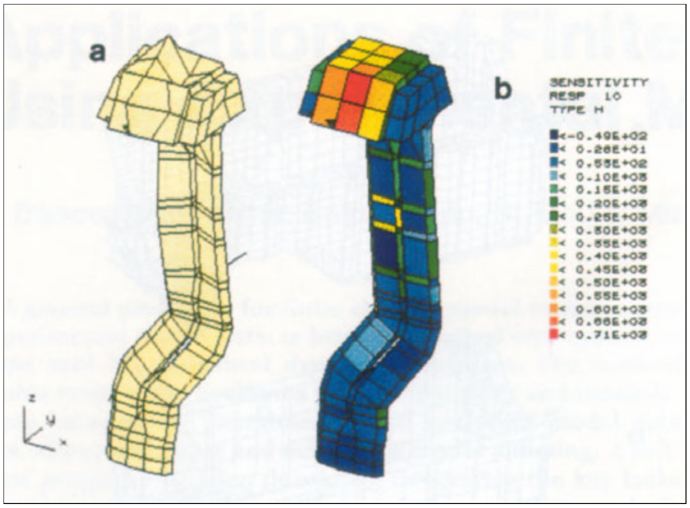

Figure 10. Typical mode shape (a) and corresponding resonance frequency sensitivities (b) for an air inlet pipe.

In order to calculate the acoustic radiation of the different modes of the oilpan, a new FE model was constructed using only 169 TRIA3 and 731 QUAD4 elements (Figure 8b). Stiffener ribs were not included in this model since they do not significantly contribute to the acoustic radiation, however, they contribute to the mass and stiffness of the oilpan. Consequently, in this application the objective was to calculate the equivalent local plate thicknesses to be used with the reduced model in order to obtain the same modal properties as predicted by the initial fine mesh. Therefore, these modal properties are used as test results that serve as reference data to be used in the model updating procedure.

Table 2 summarizes the resonance frequencies of the new coarse model, before and after the updating operation and differences with the reference data. The color-coded mesh that represents the corresponding shell thickness modifications that were required is displayed in Figure 9.

Another application is to identify the sensitive and insensitive zones in a model. This is illustrated with Figures 10a and 10b showing a typical vibration mode of an air inlet tube and the corresponding sensitivities for shell thickness modifications respectively.

Figure 11 shows a superimposed view of a FE model of a composite cylinder and the test geometry, used to identify the anisotropic material properties of the glass/epoxy laminate. After measuring 10 resonance frequencies on the specimen initial estimates for the 4 independent material characteristics (El, Et, Nult, G1t) were iteratively adjusted until sufficient agreement was obtained between test and analysis results, requiring 5 iterations. In Figures 12a-d, the variation in parameter values as a function of the iteration number is shown.

Conclusions

A procedure was demonstrated that allows practical comparisons of analytical and experimental modal data. The main advantages of the procedure are ease of interpretation of the results in terms of physical parameter modifications and ability to handle large scale structures [8].

By combining results from computer simulations and testing observations, analysts and experimentalists can evaluate and enhance their mutual benefit. This will result in a shorter design cycle at lower cost, more accurate results and the achievement of new goals.

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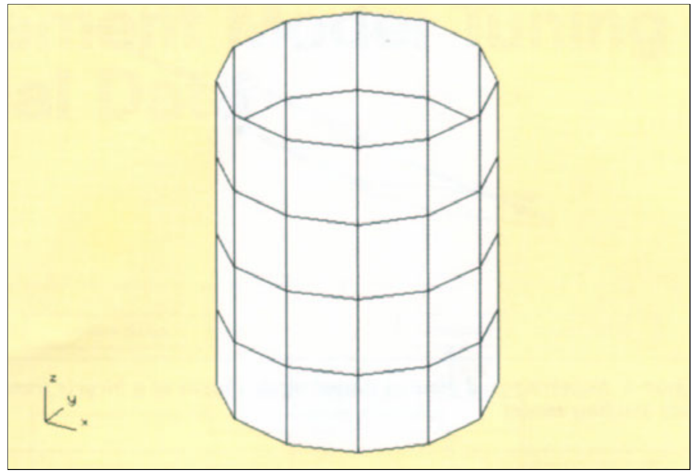

Figure 11. Composite tube tested to determine the material characteristics.

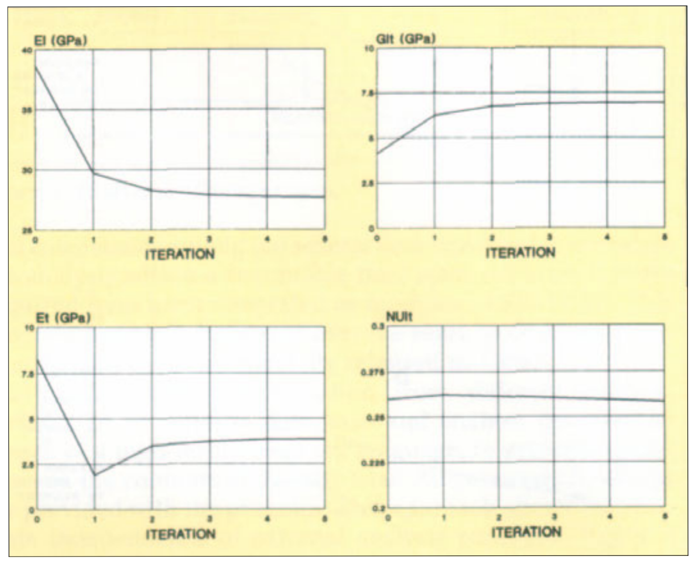

Figure 12. Convergence of predicted parameter values to their exact value, requiring 5 iterations..

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