

Design and Dynamic Analysis of Fixture Behaviour Used for Vibration Testing of some Automotive Components

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Abstract. The purpose of this paper is to develop a project for fixtures used in the automotive industry for fixing of certain components such as side airbags, curtain airbags, driver airbag, airbags in the seats airbags knee protection, devices for securing the seat belts, steering wheels, in order to test to vibration. Due to the multitude of elements which are in continuous movement on the vehicle or the high forces that occur during vehicle running, mechanical vibrations phenomena are produced. These cause aging products mounted on the car, lead to fatigue of materials, and to lower initial mechanical properties and finally the complete destruction of subassemblies. To prevent such situations, car manufacturers always ask before approval to make the simulation of vibration and then to recover destructive tests and prepare a report of the simulation correctly. Therefore, after carrying out the project it will be modelled by applying the Finite Element Method (FEM), and then using Finite Element Analysis (FEA) to determine modal parameters. A fixture for the locking of the seat belt socket is the object of the analysis in this paper.

1. Introduction

Among the methods most used in machine monitoring, due to the development of sensor systems, signal acquisition systems and real-time processing, vibration analysis has become a very powerful tool. This analysis is used in dynamics qualification of newly designed structural components, defects prediction, structural diagnosis and structural aging issues [1].

Subjecting these devices to some aging tests involves knowing some mechanical characteristics that govern their dynamic behavior in order to select parameters that can lead to aging within a prescribed time. These dynamic features are given by the natural frequencies and the natural modes of the mechanical structure [2].

Progress in automotive technology has included devices that provide increased protection for vehicle occupants in the event of accidents or vehicle collisions. Primary safety features of the vehicle have always been considered as safety belts and airbags. The seat belt socket is considered one of the most important components of the seat belt [3].

Determining the regimes to which the seat belt socket is subjected by aging or destruction tests is the purpose of this paper. For this purpose, in the Catia V5 software a fixture for the seat belt socket dynamic testing was designed.

2. Finite element modelling

The finite element method is a numerical method for solving engineering problems or physics-mathematics. The analytical solution given for such problems requires the solving of differential



equations with partial derivatives, with inputs of some initial conditions and border conditions. MEF includes, in a first stage, mesh generation techniques to divide a complex problem related to the whole body's behavior into small parts called finite elements. The simple equations governing the behavior of these finite elements are assembled into a large system of equations that govern the behavior of the whole. Therefore this numerical method leads to solving a set of algebraic equations. This part of the problem is known in engineering practice as FEA, meaning it is a computational tool for performing engineering analyzes by using coded software, based on specific algorithms. In the dynamic behavior of some mechanical structures, it is very important to know the natural frequencies and natural modes of dynamic deformation, and the algorithm of this application was first given by [4] and is known as Fast Fourier Transform (FFT) or Modal Analysis (MA).

Fixture design for seat belt socket dynamic testing, two elements, Solid 186 and Solid 187 were used to generate mesh (Figure 1). The first type of finite element is shown in Figure 2.

This type of upper order 3D element is defined by 20 nodes, each having three degrees of freedom corresponding to the three translational displacements along x, y and z nodal directions. The element supports plasticity, hyper elasticity, creep, stress stiffness, large deformation and large displacement capabilities [5-6].

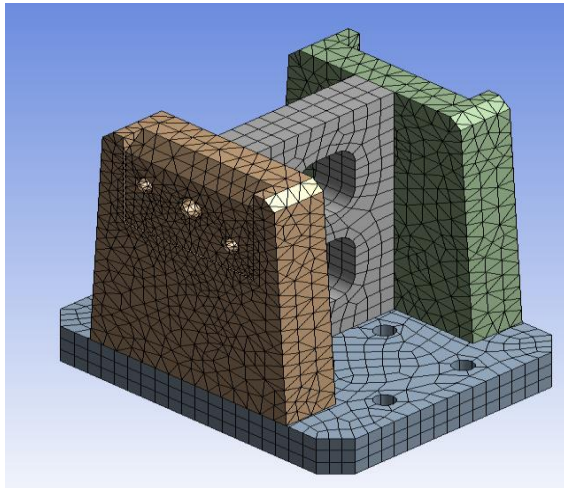


Figure 1. Finite element mesh for a fixture.

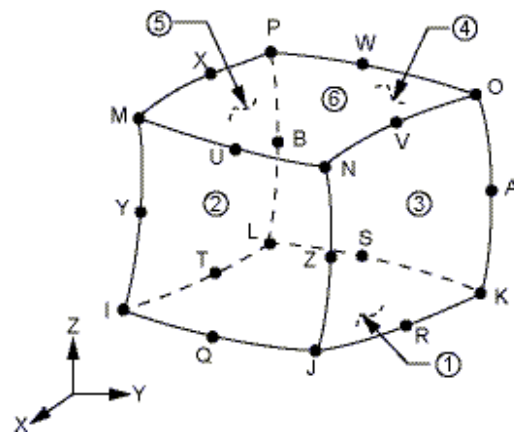


Figure 2. Spatial element SOLID186.

The second type of finite element, Solid 187, has a tetrahedral geometric shape with only 10 nodes but allows for quadratic displacement behavior as SOLID 186. This type of element is well for modeling irregular meshes, such as those made from different Catia V5 and CAD/CAM systems.

3. Modal Analysis

To determine the natural frequencies and the shape of the natural vibration modes of a fixture for a seat belt socket, the software provided by ANSYS 14.5 was used, and the finite element modeling was followed by the finite element analysis (FEA). In the construction of the mathematical model, based on finite elements, a three-dimensional model (3D), having a hexahedral geometric shape (Solid 186) and a model type having a tetrahedral geometric shape (Solid 187) were used. The first element has a number of 20 nodes, and the second one has 10 nodes, each node having three degrees of freedom. For simulation, 5 cases were analyzed. Depending on the dimensions of the geometric element, 29 767 nodes, 33 250, nodes, 38 527 nodes, 43 187 nodes and 47 383 nodes were obtained. For the modal analysis of fixture vibrations, the ANSYS module solves numerically a matrix differential equation of an undamped vibrant system given in the form:

$$[M]\{\ddot{u}\} + [K]\{u\} = \{0\}, \quad (1)$$

where $[M]$ and $[K]$ are mass and rigidity matrices, respectively $\{u\}$ are nodal displacements. The number of equations in the matrix system was three times greater than the number of nodes. In the absence of damping, the vibrations of the nodes can be expressed by harmonic functions such as:

$$\{u\} = \{\phi\}_i \cos(p_i t + \phi_i), \quad (2)$$

where $\{\phi\}_i$ are the eigenvectors and p_i are corresponding natural circular frequencies.

By imposing the solution (2) to verify the differential equation (1), this leads to a problem of eigenvalues and eigenvectors

$$(-p^2 [M] + [K])\{\phi\} = \{0\}, \quad (3)$$

from which one can determine the unknown eigenvalues p_i by imposing the condition that the equation (3) has nontrivial solutions, that means to solve the determinant:

$$|-p^2 [M] + [K]| = 0. \quad (4)$$

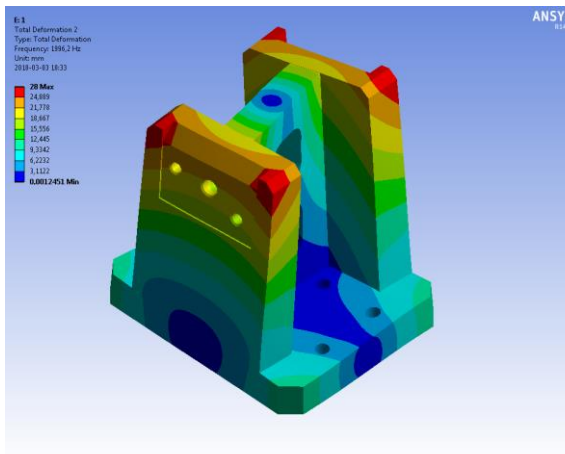


Figure 3. The first mode shape at natural frequency 1996.2 Hz.

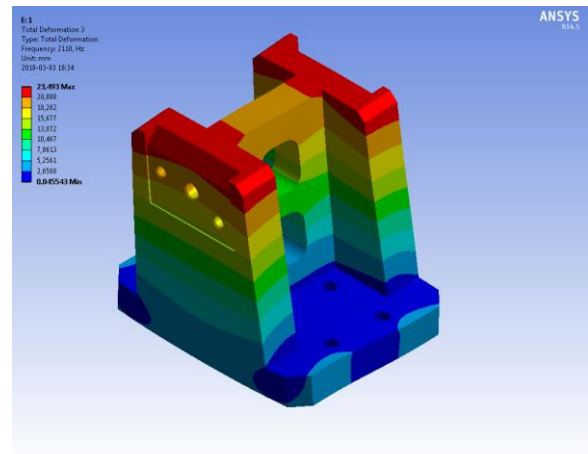


Figure 4. The second mode shape at natural frequency 2110 Hz.

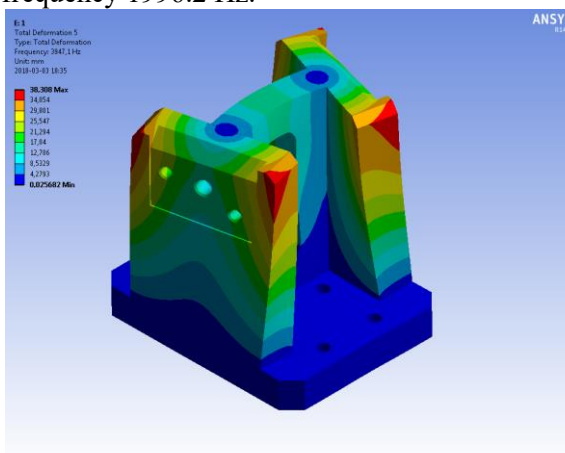


Figure 5. The fourth mode shape at natural frequency 3947.1 Hz.

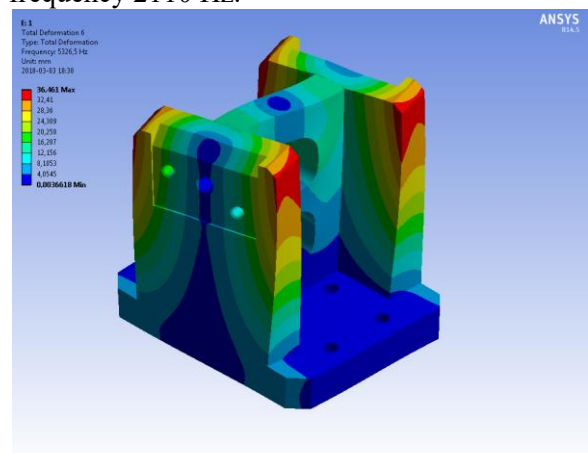


Figure 6. The fifth mode shape at natural frequency 5326.5 Hz.

Figures 3 and 4 give the forms and frequencies of the first two modes of vibration, and Figures 5 and 6 give the forms and frequencies of modes four and five.

4. Results and conclusions

The modal analysis of fixtures, used to fix auto parts subjected to dynamic tests, is made for knowing the natural frequencies, that is, those frequencies in which resonance occurs. A first objective proposed in the paper was to determine how it influences the number of elements and implicitly the number of nodes on the convergence of the ANSIS algorithm in the determination of natural frequencies. It can be seen (Figure 7) that with the increase of the number of nodes the differences obtained in the frequency of a natural vibration mode is 8-12 Hz for the low modes (first, second and third mode), respectively 26-35 Hz for the other two higher modes. In conclusion for the practical purposes of determining the frequencies to which the head shaker extender is required to act, no finite elements with dimensions of less than 5 mm are necessary. This has an effect on reducing the time allocated to dynamic analysis.

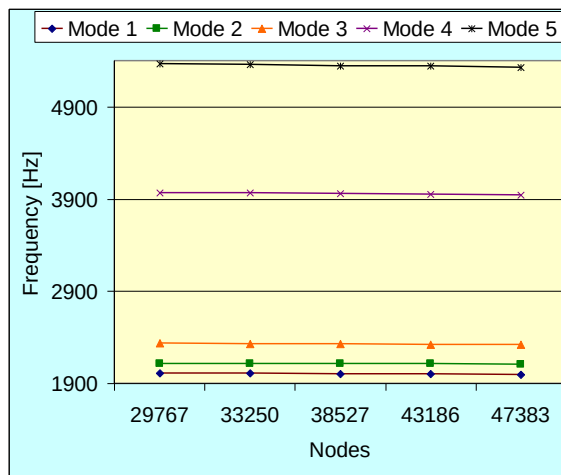


Figure 7. Variation of natural frequencies according to the number of nodes.

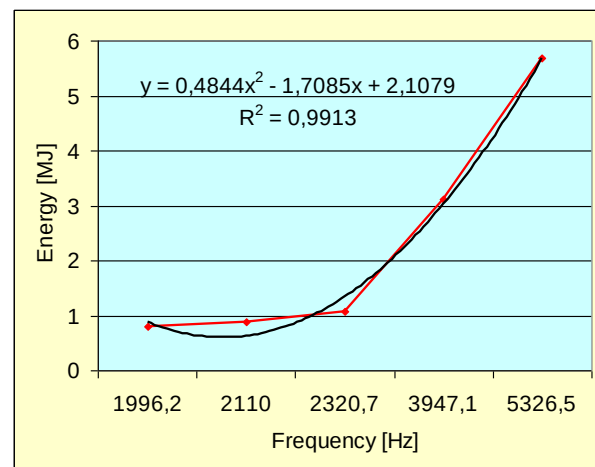


Figure 8. Variation of sample energy according to the frequencies of natural modes.

A second aspect to be taken into account in dynamic testing is that the energy transmitted to the samples is proportional to the square of the vibratory machine frequencies. The graphical representation (Figure 8) of energy variation according to the natural frequencies confirms this, and the interpolation curve is a parabola having a correlation coefficient R^2 equal to 0.9913, i.e. the mean square deviation, between the results obtained by the FEA and the theoretical ones are very good.

5. References

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