International Journal of Modern Trends in Engineering and Research
www.ijmter.com
e-ISSN No.:2349-9745, Date: 2-4 July, 2015

Natural Frequency Estimation of Headlamp Fixture and Its Co-Relation with Experimental Data

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Abstract- Modal testing is an effective technique to identify the modal parameters of structural systems under ambient or operational conditions. It can be effectively used for model validation, model updating and for determination of modal characteristics of the structures. Structural dynamics deals with the behavior of structures under dynamic loading. Dynamic analysis is used to determine operating deflection modes and fundamental frequency. This can be effectively studied by analyzing the mode shapes obtained. The objective of this work is to achieve the desired natural frequency of the headlamp fixture by developing the different designs. The natural frequencies of the fixture are obtained and respective mode shapes are estimated. Modifications in headlamp fixture are proposed based on the experimental and theoretical modal analysis. The accuracy of numerical results obtained is assured by comparison with experimental results. The proposed method can be applied to different types of fixtures of automotive electronic clusters, heating ventilation air conditioning modules etc. which undergoes vibration testing analysis.
Keywords- Natural Frequency; Modal analysis; Finite element method; Eigen value; FFT analyzer.

I. INTRODUCTION

Modes are used as a simple and efficient means of characterizing resonant vibration. The majority of structures can be made to resonate. That is, under the proper conditions, a structure can be made to vibrate with excessive, sustained, oscillatory motion.

The knowledge of the dynamic behavior of structures is of primary importance in many applications, particularly in the field of aerospace and mechanical design. Prediction of resonance frequencies and mode-shapes are essential steps in design analysis. For this purpose, numerical techniques based on finite element model are commonly used. Uncertainties in mechanical properties, tolerances in fabrication and assembly process may cause problem in correlation between numerical predictions and experimental results. To ensure that the mechanical component will survive the dynamic environment in which it is operating; vibration qualification testing is required according to international standards or manufacturer specifications.

To perform vibration testing on electro-dynamic shaker, the test item has to be properly mounted on vibrating table using a fixture. On the one hand, it is of primary importance that the fixture which supports head lamp assembly on shaker table does not modify the dynamic behavior of head lamp assembly under test. To this end, the lowest resonance frequency of the fixture should be much higher than the frequency of interest. This condition allows to avoid significant dynamic interaction between the fixture and the specimen. On the other hand, as the performance of the shaker (i.e. the maximum force that it can develop) is directly related to the mass of the moving elements, which will include the structure under test mounted on the fixture and the shaker slip-table, the fixture should be as light as possible. Thus rigidity and lightness are two contra dictionary objectives and balance between them needs to be achieved. Moreover, additional constraints have also to be accounted for. For instance, the hole pattern on the shaker slip-table that is used to mount the test item, needs to be reproduced (at least partly) on the fixture; the center of gravity of the assembled system (i.e. the test item on its support) has also to be as close as possible to the center of the shaker slip-table in order to minimize the effect of resulting moments. All the previous considerations show that the structural design of fixtures corresponds to a constrained optimization problem.

The automotive headlamp shall be subjected to vibration testing of the frequency range 70 to 80 Hz .For this purpose first natural frequency of the testing fixture shall be at least 210Hz.To obtain the desired frequency for the fixture various iterations are required which will be carried out in Nastran solver to determine the exact locations in structure that require structural modifications to increase the frequency of the fixture effectively.

II. EIGEN VALUE CALCULATION

If a system is given some initial disturbance, then it will vibrate at some frequency known as natural frequency. The lowest natural frequency, usually referred to as the fundamental frequency, has the lowest strain energy.

The modal analysis has been done to determine the natural frequencies of an object or structure during free vibration. For natural frequency calculation the component is discretized into elements and described the variation of the displacement with each element, the kinetic and potential energies of the structures are calculated to find the natural frequencies and mode shapes in terms of various nodal values. Numerically this equates to solving an eigen value problem expressed in terms of mass and stiffness matrices. From elementary engineering mathematics eigenvalues and eigenvectors may be evaluated. The physical interpretation of the eigenvalues and eigenvectors which come from solving the system are that they represent the frequencies and corresponding mode shapes. Sometimes, the only desired modes are the lowest frequencies because they can be the most prominent modes at which the object will vibrate, dominating all the higher frequency modes.

For the most basic problem involving a linear elastic material which obeys Hooke's Law, the matrix equations take the form of a dynamic three-dimensional spring mass system. The generalized equation of motion is given as,

$$[M][\ddot{U}] + [C][\dot{U}] + [K][U] = [F]$$

Here $[M]$ is the mass matrix, $[\ddot{U}]$ is the 2^{nd} time derivative of the displacement $[U]$ (i.e., the acceleration), $[\dot{U}]$ is the velocity, $[C]$ is a damping matrix, $[K]$ is the stiffness matrix, and $[F]$ is the force vector. The general problem, with non- zero damping, is a quadratic eigenvalue problem. However, for vibrational modal analysis, the damping is generally ignored, leaving only the 1st and 3rd terms on the left hand side,

$$[M][\ddot{U}] + [K][U] = [0]$$

This is the general form of the eigen system encountered in structural engineering using the FEM. To represent the free-vibration solutions of the structure harmonic motion is assumed, so that $[\ddot{U}]$ is taken to equal $\lambda[U]$, where λ is an eigen value (with units of reciprocal time squared S^{-2}) and the equation reduces to,

$$[M][U]\lambda + [K][U] = [0]$$

In contrast, the equation for static problems is,

$$[K][U] = [F]$$

which is expected when all terms having a time derivative are set to zero.
In linear algebra, it is more common to see the standard form of an eigen system which is expressed as,

$$[A][x] = [x]\lambda$$

Both equations can be seen as the same because if the general equation is multiplied through by the inverse of the mass $[M]$-1 it will take the form of the latter. Because the lower modes are desired, solving the system more likely involves the equivalent of multiplying through by the inverse of the stiffness $[K]$-1 a process called inverse iteration. When this is done, the resulting eigen values μ , relate to that of the original by,

$$\mu = \frac{1}{\lambda}$$

but the eigenvectors are the same.

III.MODAL ANALYSIS OF AUTOMOTIVE HEADLAMPFIXTURE USING FEA

The automotive headlamp is subjected to vibration testing of the frequency range 70 to 80 Hz .For this purpose first natural frequency of the testing fixture shall be at least 210Hz. Various designs of headlamp fixture are analyzed for the fundamental frequency calculation which are carried out in Nastran solver to determine the exact locations in structure that require structural modifications to increase the frequency of the fixture effectively.

Finite element model of headlamp fixture is prepared using commercially available FEA tool.3D CAD model of fixture is discretized using tetrahedron element geometry and components like M6, M10 screw is modelled with rigid elements. The total number of elements in the model is 25719 and rigid elements are 42.

The base plate of fixture is fixed at 6 locations in all degrees of freedom as these locations are fixed with shaker table in physical testing. The modal analysis is carried out with these fixations to get the natural frequency. Material properties are taken as per manufacturer's specifications of the laboratory.

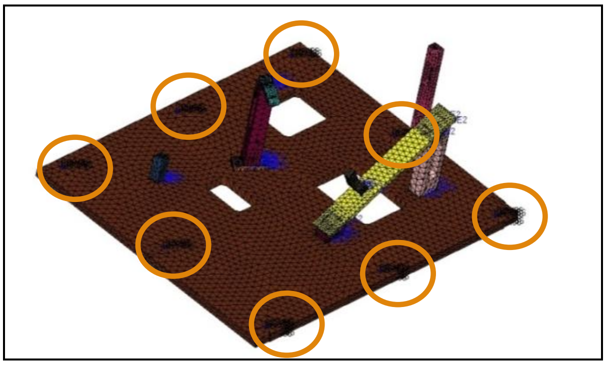

Fig. 1: Boundary conditions used in FEA

Post-processing of the results obtained has been done in commercially available FEA tool. Multiple designs are analyzed to achieve the target frequency of headlamp fixture i.e. 200 Hz. Frequency achieved for final design is 229.7 Hz. Mode shape for the final design of headlamp fixture is shown in fig. 2. Here, the base plate thickness of the final design of the fixture was 25 mm this was difficult to manufacture due to cost constraints. Instead, thickness of base plate of the fixture was taken as

14.5 mm and the same was applied to the FEA model with the assumption that if the experimental and FEA results for 14.5 mm thickness could be correlated then the same will hold true for 25 mm thickness.

A similar assumption has also been made in terms of the degree of freedom applied to the fixture. The experiment was performed by placing the fixture on the floor. In other words the Z direction movement of the fixture was arrested and the same was applied to the FEA model as well. Here it has been assumed that if the experimental and FEA results for arrested Z direction analysis match then the same would also be true for all dof constraint at the 8 mounting locations. For 14.5 mm base plate thickness the frequency obtained was 120.3Hz, it is expected that experimental frequency should also be close to this value.

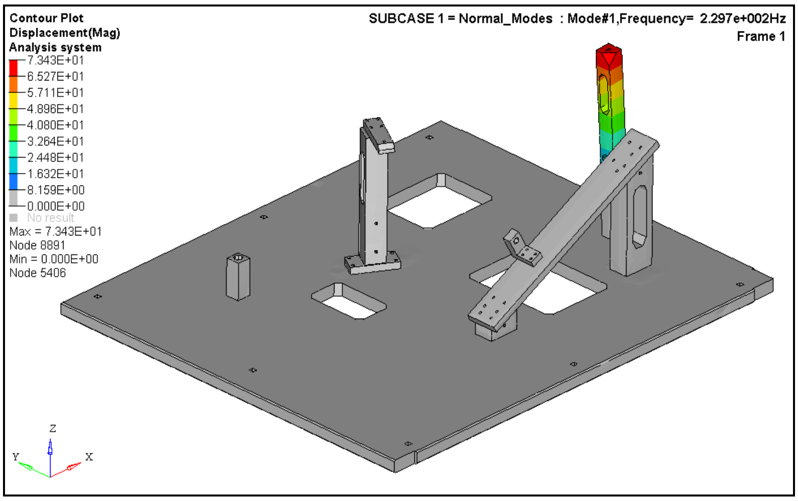

Fig. 2: Mode shape for Final design of Headlamp fixture

IV. EXPERIMENTAL MODAL ANALYSIS OF HEADLAMP FIXTURE

Impact testing has become widespread as a fast and economical means of finding the modes of vibration of a machine or structure. Modal testing and experimental modal analysis is the process of characterizing the dynamic properties of a test structure by exciting the structure artificially and identifying its modes of vibration. When a structure is damaged e.g. its geometrical properties change, its boundary conditions modify or its material properties alter the dynamic characteristic of the structure change. These changes are the basic for the damage identification methods and modal testing. Experimental modal analysis serves as a means to extract dynamic properties for a given structure.

In modal testing and experimental modal analysis the dynamic characteristics of a structure are identified from its vibrational responses. While modal testing describes the performance of the testing and acquisition of the modal data from test structure, experimental modal analysis is the process of determining the modal parameters (natural frequencies, damping ratios and mode shapes) from the acquired data. Generally modal testing and experimental modal analysis can be divided in to three major steps: signal processing, frequency response function (FRF), modal parameter estimation. One common frequency response measurement method utilizes an impact hammer to excite the structure and an accelerometer to measure the response. An impact test generates an impulse excitation, which will result in a transient response. The frequency content of the energy applied to a structure is a function of the stiffness and masses of the contact surfaces. This in turn controls the shape of the force input and the frequency content of the measurement. Because it is not feasible to alter the structure's surface or mass, impact hammers are available with a variety of interchangeable contact surfaces and masses. Altering either the tip or the mass of the impact hammer will result in a change in the frequency content of the measurement. If a hard tip is used, the structure is impacted a shorter duration of time, resulting in an excitation of a higher frequency content. Generally, a tip should be chosen so that the amplitude of the force spectrum is no more than 10 to 20 dB down at the maximum frequency of interest.

Both noise and leakage are problems which may arise when an impact test is conducted. Noise can be reduced by using a Force Window, which literally forces the sample window to zero after a defined amount of time. Leakage can be found in a system's response if the acquisition software performs a Fourier Transformation on a non-periodic sample window. This error can be reduced by either applying an exponential window to the response signal or adjusting the sample window so that the system decays to a constant displacement of zero. However, if an exponential window is used, the system being analyzed may seem to have a higher damping ratio.

Experimental set-up consists of headlamp fixture, impact hammer, FFT analyzer and the accelerometer. Ideally it is required to perform natural frequency calculation in free-free condition i.e. all dof are kept open this is very difficult to perform since ideal free-free condition would consist of placing the component in space. So the headlamp fixture is placed directly on the floor for natural frequency calculation for the following reasons. Placing the headlamp fixture essentially arrests the Z direction movement. However it also simplifies the experimental procedure. Also such a set up can be easily reproduced using FEM by providing boundary conditions.

The accelerometer is attached to the headlamp fixture at location where the acceleration is to be measured. The accelerometer is connected to the FFT analyzer by means of a cable. Another cable connects the impact hammer to the FFT analyzer. Input force from the impact hammer is provided to the FFT analyzer whereas the corresponding acceleration of the headlamp fixture is provided by the accelerometer. Using this data the FFT analyzer calculates the frequency response.

Table No. 1 Material Properties

Properties of material used for automotive headlamp fixture.	
Material type	***Isotropic material***
Modulus of elasticity,E	70 e+09 N/m2
Density,ρ	2700 kg/m3
Poisson's ratio, ν	0.3

After carrying out the experimentation following results are obtained. The frequency response functions obtained were curve-fitted. The resonance frequencies corresponding to the peaks of the FRFs are taken as the natural frequencies. By moving the cursor along the FRF, the peak where the phase is 900 that value is taken as natural frequency at that corresponding mode.

The MATLAB code is used for the calculation of natural frequency of the specimen. Matlab is generally programming software unlike other programming languages it is problem-solution kind of software which is much useful to evaluate results instantly.

Test data for Natural frequency calculation

The Headlamp fixture is placed directly on the floor and modal analysis is performed. The first natural frequency of headlamp fixture is 110 Hz. Input of hammer with respect to time is shown in Fig. 3. Output of Acceleration v/s Time is shown in Fig. 4. Fig.5 shows the FRF test data i.e. Amplitude v/s Frequency. Amplitude v/s Frequency plot is generated by using the Matlab code.

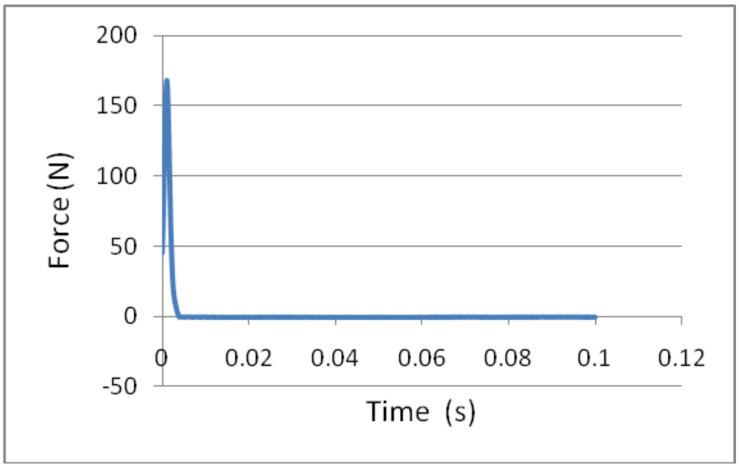

Fig. 3: Force v/s Time plot

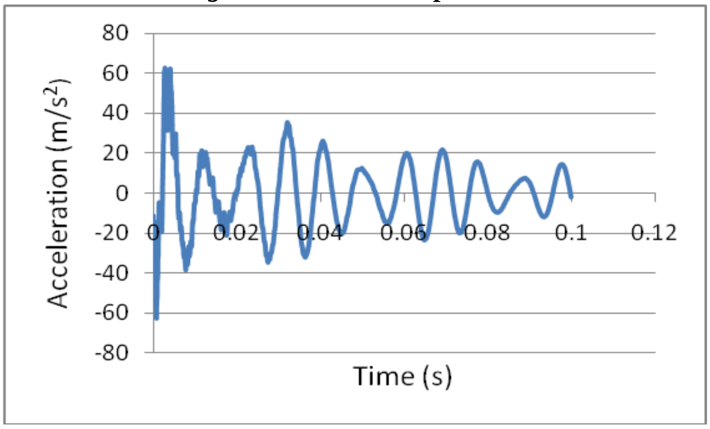

Fig. 4: Acceleration v/s Time Plot

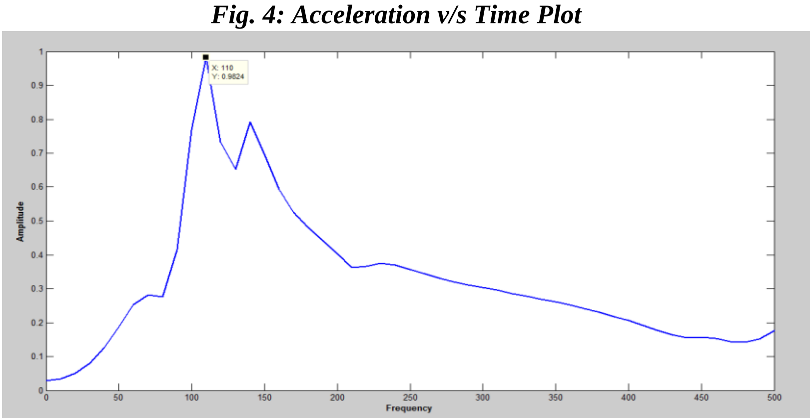

Fig. 5: Amplitude v/s Frequency Plot

Ideally it is required to perform natural frequency calculation in free-free condition i.e. all dof are kept open this is very difficult to perform since ideal free-free condition would consist of placing the component in space. So the headlamp fixture is placed directly on the floor for natural frequency calculation for the following reasons. Placing the headlamp fixture essentially arrests the Z direction movement. However it also simplifies the experimental procedure. With this set up, first natural frequency of the headlamp fixture observed is 110 Hz. The same kind of set up reproduced using FEM by providing appropriate boundary conditions. The first natural frequency with FEA calculations is 120.3 Hz.

The correlation between physical test data and CAE result is around 90%.So it can be concluded that the experimental value of natural frequency and FEA match very well with the FEA model. So if the experimental and FEA results for arrested Z direction analysis match then the same would also be true for all dof constraint at the 8 mounting locations. The frequency calculated in FEA for headlamp fixture with all dof constrained at 8 locations is 229.7 Hz. Thus the objective of the work is achieved as the natural frequency of the testing fixture is above 210 Hz in addition to that the mass of the fixture is reduced by 18.4 Kg.

CONCLUSION

Modern experimental modal analysis techniques are reviewed in this paper. FRF based modal testing has been used to determine natural frequency of the Headlamp Fixture experimentally. FEA tools are utilized for the natural frequency calculation in intermediate stages of different designs of headlamp fixture.

Dynamic characteristics mass and stiffness of structure have been optimized in the given frequency range. Laboratory investigation (impact testing) was made to validate experimentally modal natural frequencies of Headlamp fixture. The finite element model of fixture was prepared and correlated with the experimental results. The FEM and experimental results showed good compliance.

Finally the fixture that fulfills the desired frequency criteria is achieved with multiple designs and the same is validated with experimentation. In addition to that mass of headlamp fixture is reduced by 18.4 kg. The final design is the best combination of frequency and mass.

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