Vibrational Analysis of Gearbox Casing Component using FEA Tool ANSYS and FFT Analyser

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***Abstract*-This Paper is concerned with the application of ANSYS software and also FFT analyser to determine the natural vibration modes and find the free frequency of the Gearbox casing. By finding natural frequency of the Gearbox casing component in order to prevent resonance for gearbox casing component. From the result this analysis can show the range of natural frequencies of gearbox casing component with maximum amplitude of it.**

This is representative of certain key components of complex structures used in Automotive and Production Industries. The mass and stiffness matrices are then determined by exact analytical integration. The results are in close agreement with both experimental and results using analytical approach.

Key Words: FEA, Modal Analysis, Gearbox Casing, Natural Frequency, Mode Shape, Modal test.

1. INTRODUCTION

The casing of gearbox is an important component in a synchromesh gear box. The function of gearbox casing is to accommodate and support power train. Since the gearbox application is for high speed (typically 800RPM to 3200RPM), failure of casing may lead to major primary damage of the gearbox. Analysis of gearbox casing is very essential in order to decide appropriate dimensions and to predict the behaviour of casing under different operating conditions.

Several authors have analysed using commercial software as well as experimentally, few of them presented as follows R. V. Nigade [1] have compared gearbox casing mode shapes and vibration predictions obtained from analytical model with those of experimental results obtained from the test rig at the Kirloskar Pneumatic Co Ltd., Pune, Maharashtra, India. It was found that the natural frequencies of the simulated results were within 5 percent of the experimental values. Also, the simulated mode shapes were very similar to the experimental modes shapes.

The good agreement between the analytical model and the experimental measurements confirmed the accuracy of the dynamic representation of the test gearbox.

2. MODELLING AND FINITE ELEMENT ANALYSIS OF GEARBOX CASING COMPONENT.

The 3-D solid model of the gearbox casing component was build using PTC ProEngineer 5®. ANSYS workbench 12® wasused for pre-processing, solving and post processing. Material properties of Aluminium alloy, gradeIS 4223 & BS LM 4 were selected from PSG Coimbatore 1978 having Fatigue properties from MIL-HNBK-5H pages 3-277. Solid 3-D model of the gearbox casing component was meshedusing sizing of 2 mm (3-D 10-Node Tetrahedral Structural solid element). The FE model consists of 146,164elements. Next, Block Lanczos technique was used to extract first 4 natural frequencies and mode shapes of themodel.

Figure 1. Gearbox Casing Component

2.1. Excitation Forces

The excitation force can be divided into three types.

First type is the mechanical looseness and unbalance at running speed. As power transmitting components i.e. rotating components are balanced to grade G-2.5 as per ISO 1940-Part I, excitation due to mechanical looseness and unbalance is not considered.

The Second type of excitation force is due to engine, which is running at 3200RPM. In order to prevent resonance of the gearbox casing component, it is expected that natural frequency of the gearbox casing component should have a minimum separation margin of 20%, from harmonics of the exciting frequency, i.e. of engine speed. The frequency of harmonics of engine speed considering 5% frequency variation of input power is 27 Hz which is not considered.

The Third type of excitation is due to gear tooth meshing. However as excitation frequency due to gear tooth meshing of 1st, 2nd, 3rd and 4th are 508Hz, 884Hz, 1147Hz and 1447Hz respectively where input shaft is rotating at 3200 RPM, which is high and very near from natural frequency, hence excitation corresponding to gear mesh frequency is considered.

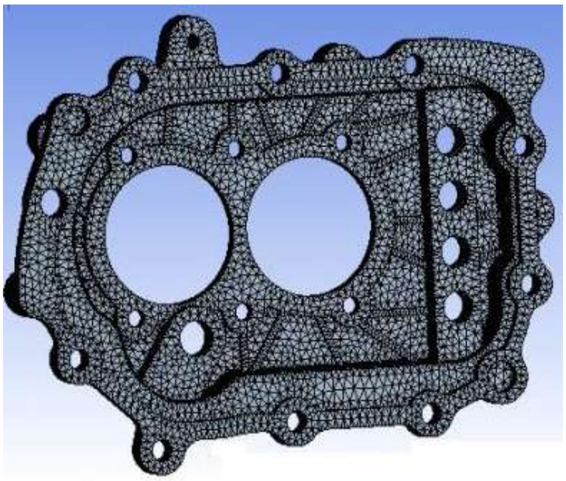

Figure 2. Meshing Model of Gearbox Casing Component

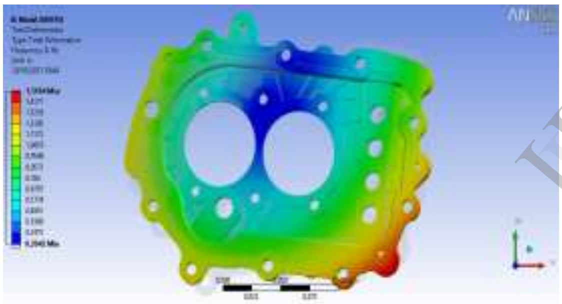

Figure 3. Mode Shape of Component obtained from ANSYS Workbench 12®

Table 1. Frequency from ANSYS Workbench 12®

Modes	Frequency Set
1	819.4
2	1475.5
3	2372.7
4	2562.9

3. MODEL TESTING

3.1. Pre-test planning

In order to ensure that the FE model of a structure can be used with confidence for the prediction of the dynamic behaviour of that structure, the model must be validated by tests. In the validation process, several types of tests should be performed to obtain measured data, and then used for comparison with data predicted by the FE model.

Before a modal test is performed there are some features that should be carefully selected. For the process of FE model validation, there must be initial FE model. Although this model may not be reliable enough to predict the dynamic properties of the structure accurately, it must contain some useful information about the structure's dynamic properties.

3.1.1. Optimum suspension point(s) selection: First, the suspension arrangements should be considered to make sure that the test structure is supported in the desired condition. In many modal tests, free-free conditions are required. However, providing such a condition in practice is difficult. Therefore, the suspension points in such test should be selected so that suspension has a little effect as possible on any mode of vibration in the frequency range of interest.

In most modal tests, relatively soft springs / flexible ropes are used to connect the test structure to ground [1]. Thus it can be assumed that there is no mass, but only stiffness attached to the suspension points so that any additional forces will result from displacement of the suspension points. The stiffness of the suspension should be as low as possible so that the natural frequency of the highest rigid-body mode of the test structure is well below the natural frequency of its first flexible mode [1]. In addition to this consideration, selecting the optimum suspension points can be helpful in reducing the additional forces to the test structure during the test.

The optimum suspension points can be selected on the basis of two criteria [5]. The first criterion is that the total displacement amplitude at all of the selected points for all modes in the specified frequency range be as low as possible, so that the additional forces caused by the displacement at these points during the test will be negligible.

The second criterion is that the vibration movement at a suspension point during the test is mainly in the plane that is normal to the suspension spring axis. Of course, there are some other limitations for suspension points selection: for example, the points selected must be accessible and have no need to drill through the specimen or attach additional items.

3.1.2. Optimum driving point(s) selection: In the given frequency range, every mode of the test structure has a different mode shape. If the chosen driving point is in the vicinity of less excitation point of any individual mode, that mode cannot be excited to a sufficient level to ensure that the measurement of its properties will be reliable.

Different excitation methods can also influence the selection of the driving point. If the hammer excitation method is to be used in a modal test, the vibration velocity amplitude at the driving point should not be so large as to give a high possibility of a double hit. If shakers are to be used to excite the test structure, the vibration acceleration amplitude at the driving point should be limited in order to eliminate the inertial effects caused by the additional mass of excitation equipment and force sensor(s).

The optimum driving point(s) selection process is based on two criteria. The first criterion is to avoid selecting a point near to any nodal line of any of the modes in the specified frequency range. The second criterion is to avoid selecting the points with excessively large vibration amplitude.

3.1.3. Optimum response DOFs selection: In the model validation process, the measured data should contain sufficient information to positively identify each mode in the specified frequency range. That means that all the measured modes should be linearly independent at DOFs where the response is measured. In pursuit of obtaining accurate measurements, the DOFs with large vibration amplitudes are usually referred for measurement.

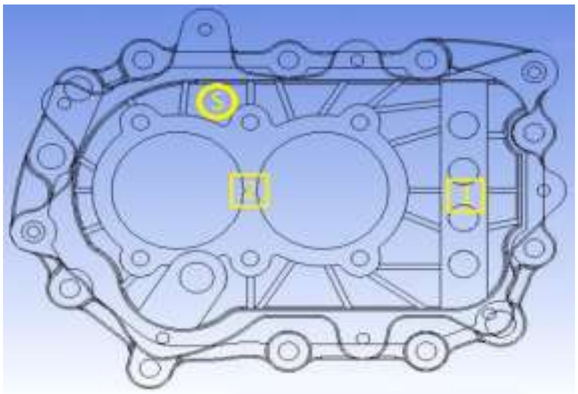

Figure 4. Measurement locations on gearbox top cover.

On the other hand, the number of measurements cannot be enlarged without reaching practical limits. In order to fulfil both the requirements, the response DOFs should be selected so that they contain enough information to distinguish all the modes, while at same time the number of measurements should be kept as low as possible (less than 5% of total model DOFs).

Here, total 3 points - Impact points 1and 2, were selected as measurement locations as shown in Figure. 4. Accelerometer was mounted on point number 'S'.

3.2 Modal testing and modal analysis

*3.2.1. Hammer Testing:*The gearbox top cover was suspended by using the flexible ropes having least stiffness to achieve nearly free-free condition. Multichannel (four) spectrum Analyzer, data collector and balance with software along with acceleration sensor sensitivity 100 m V/g (g=9.81 m/s2). Impact hammer (dynamic quartz sensor -9722A- for light to medium structure at med. To high frequencies). Measuring range upto 2000N with cable and other accessories, sensitivity at 100 Hz=2mV/N, overload capacity= 500N, Resonance frequency = 27 KHz, hammer mass=100 gm., Rigidity=0.8 KN/Micron, Microphone with cable, sensitivity = 46 .17 mV/Pa.One B&K piezoelectric accelerometer was attached to point 'S' on the structure. The overall arrangement was as shown in Fig. 5 (a). The measurements were carried out by impacting the structure at point 1 and 2as per predefined location / co-ordinate system and acquiring the response data at point number 'S'.

Figure 5. (a) Photograph of Modal Testing

Figure 5. (b) Photograph of Modal Testing

*3.2.2. Modal Analysis Results:*The impact points are further plotted in graph. (The software is used to post-process the data, which is stored during the modal test of the structure). Natural frequencies & mode shapes are further obtained as output. The testing (surface) model is prepared in Software, with the location of points, as that of the actual component test points i.e. points tested on actual structure by using FFT analyser along with data acquisition system. During the analysis four natural frequencies of the gearbox casing component are extracted from the data as seen in Figure 5 (b).

Table 2. Frequency from FFT Analyser Test

Mode	First Test	Second Test
1	834	840
2	1428	1440
3	2340	2308
4	2558	2556

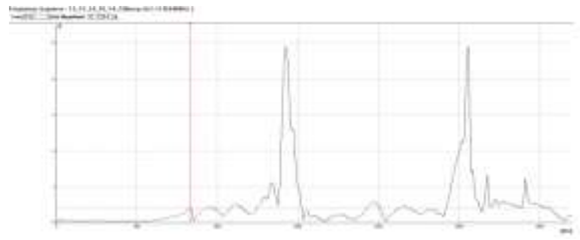

Figure 6. (a) Natural Frequencies Plot from first hammering position.

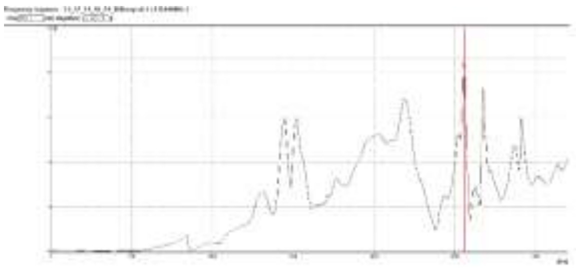

Figure 6. (a) Natural Frequencies Plot from second hammering position.

3.2.3. Correlation of experimental and FEA results: In FEM modal analysis mass of the structure is very important, which decides the frequency of structure.

Modes	ANSYS Reading	FFT Reading 1	FFT Reading 2	Percentage Deviation
1	819.4	834	840	1.75
2	1475.5	1428	1440	3.21
3	2372.7	2340	2308	1.35
4	2562.9	2535	2556	1.05

Figure 7. Graph of all readings set obtained from ANSYS and FFT Analyser

For small variation in mass, modal test data is affected. Hence, mass obtained in FEM should be equal to actual physical mass of structure. The actual mass of the casing component was 1.322 kg and that of model in the ANSYS® was 1.36297. If we compare the mass of both the model, it shows that it is quite a good match between the above mentioned models. Hence it can be said that the model in the ANSYS® is validated to actual manufactured gearboxcasing component, with respect to the mass.

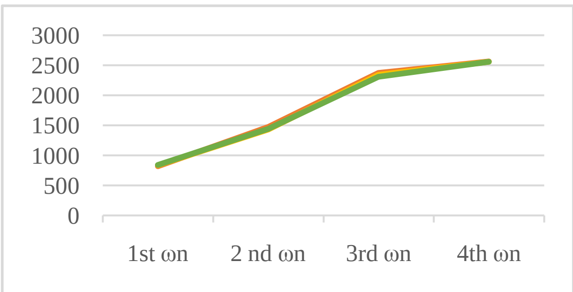

Table 3. Comparison of Frequencies for Gearbox Casing Component (Modal Tests data with FEA results)

The above figure 7 and table 3 shows the percentage variation in natural frequencies between the Modal test data and FEA results. The natural frequencies of the predicted modes in ANSYS workbench 12 ® are within 4 percent of the measured modes of modal test data. Also, the predicted mode shapes are very similar to the experimental mode shapes.

Therefore, it can be concluded that the modal test data is in good agreement with ANSYS® modal analysis results.

4 SENSITIVITY ANALYSIS

Having finished the measurement and finite element validation, it was decided to accept the model without any further correction. Most of the times it is generally found that, even after satisfying stress criteria, separation margin of natural frequency of the gearbox casing from that of the excitation frequency, is not more than 20%. In order to achieve the separation margin, 3D model of the casing component needs not to be updated.

5 CONCLUSIONS

Analysis has been carried out to examine in detail the vibration characteristics of the Casing of integrally geared Synchromesh Gearbox. The findings are as follows:-

(a) *FEM Modal Analysis for Gearbox Casing:*FEM Modal analysis for gearbox casing component is carried out using ANSYS Workbench 12® software. It is observed that the obtained natural frequencies are separated by 20% from first and second harmonics of the excitation frequency. Thus the results are within acceptable limits.

*(b) Experimental Modal Analysis for Gearbox Casing:*Experimental validation results show close agreement with FEA results of the existing casing. Natural frequencies of the predicted modes are within 4 percent of the measured modes.

The difference in the experimentation results and FEA results may be mainly because of difference of material properties especially density, Poisson's ratio, young's modulus etc. and uneven thickness of the casing component. In addition, the patterns of the predicted mode shapes are similar to the experimental mode shapes. Thus it can be concluded that the FEA results for the gearbox casing component for four speed G65-4 speed with full remote shifting shows close agreement with the experimental modal test data. A parametric study was conducted on casing thickness, element size in ANSYS workbench 13®

(i) The effect of varying casing thickness on the natural frequencies of the gearbox casing component was not significant, as compared to increase in the weight of the gearbox casing component.

(ii) The effect of use of smaller element size in ANSYS Workbench 12®, increases FEA computational time drastically, without significant changes in natural frequencies. However, the use of default setting gives the optimum results with less time.

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