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Modal and Stress Analysis of Differential Gearbox Casing with Optimization

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ABSTRACT
The process of casting design in the automotive industry has been significantly refined over the years through the capabilities of advanced computer aided design and engineering tools. One of the significant benefits of these computer aided capabilities is the direct access to CAD geometry data, from which finite element models can be quickly developed. Complex structures can be meshed and analyzed over a relatively short period of time. The application of advanced finite element analyses such as structural modification and optimization are often used to reduce component complexity, weight and subsequently cost. Because the level of model complexity can be high, the opportunity for error can also be high. For this reason, some form of model verification is needed before design decisions made in the FEA environment can implemented in production with high confidence. Dynamic correlation, comparison of mode shapes and natural frequencies, is a robust tool for evaluating the accuracy of a finite element model. The objective of the project is to analyze differential gearbox casing of pick up van vehicle for modal and stress analysis. The theoretical modal analysis needs to be validated with experimental results from Fourier frequency transformer analysis. The main motivation behind the work is to go for a complete FEA [1] of casing rather than empirical formulae and iterative procedures.
Keywords- Differential gearbox casing, Modal analysis, Stress analysis, Optimization, FFT analyzer, vibration

I. INTRODUCTION

The differential gearbox casing is an important part in the pick up van vehicle. The casing encloses different sets of helical gears, spur gears and three bearings to support the shafts. The bottom part is filled with oil. In a power transmission gear system, the vibrations generated at the gear mesh are transmitted to the gearbox housing through the shafts and bearings. To do the analysis of entire gearbox casing it is necessary to do the analysis of casing, set of gears and effect of oil. Optimization and structural modification is necessary to reduce component complexity, weight and subsequently cost. This project work requires the analysis of casing which is die cast ALSi132 material equipped with sets of stiffeners on either side. The use of the stiffener is to reduce vibration, stress and noise. Till recently Gear box casing torsional as well as bending vibration analysis was done by the empirical formulae and iterative procedures, but the simplifying assumption that a throw of casing has one degree of freedom is only partially true for torsional modes of vibrations. More degrees of freedom are required to get information about other modes of vibration and stress distribution. The complicated geometry of casing and the complex torque applied by cylinders make their analysis difficult. However, optimized meshing and accurate simulation of boundary conditions along with ability to apply complex torque provided by various FEM packages have helped the designer to carry torsional vibration analysis with the investigation of critical stresses. FEM enables to find critical locations and quantitative analysis of the stress distribution and deformed shapes under loads. However, detailed modeling and specialized knowledge of FEM theory are indispensable to perform this analysis with high accuracy. They also require complicated meshing strategies. Simulation of actual boundary conditions to equivalent FE boundary conditions has to be done carefully because a wrongly modeled boundary condition leads to erroneous results. The solution of such large scale FEM problem requires both large memory and disc space as computing resources.

II. DESIGN STAGE

Important points to be considered at design stage in order to reduce vibrations and noise in differential gearbox casing are as follows:
1) Shape and structure of its housing
2) Shape of stiffener
3) Thickness of stiffener
4) Layout of stiffener

Fig. 1- Existing Component

III. INDUSTRIAL RELEVANCE

Now days, for better performance or lower cost by using advanced technique like CAD/CAM/CAE, Quality inspection tool for rapid product development. The development of a product is an iterative process, which includes product design, analysis of performance, safety and reliability, product prototyping for experimental evaluation and design modification. Industries provides tailor made solution to optimize housing with own product range as per customer demand. There is inadequate documentation of the original design [5]. The original manufacturer no longer exists, but a customer needs the product. The original design documentation has been lost or never existed. Some bad features of a product need to be design out. To strengthen the good features of a product based on long-term usage of the product. To gain competitive benchmarking methods to understand competitor's products and develop better products. The original supplier is unable or unwilling to provide additional parts. The original equipment manufacturers are either unwilling or unable to supply replacement parts, or demand inflated costs for sole-source parts.

IV. METHODOLOGY

The problem under consideration will be modeled through six approaches

1. 3-D CAD Modeling
2. 3-D Finite Element Meshing
3. Analytical Calculations
4. Finite Element Analysis (Structural, Static and Dynamic)
5. Experimental Validation
6. Optimization of Structure and Re FEA for Safe Check for Operating Conditions

Based on the Nodes [6] and Element formulation, the Finite Element Model will be divided into a number of Elements where the variable of interest is located at the vertices of element called nodes.

The dissertation work will be done in six phases as follows:

Phase I:

1. Study the literature on Casing, Gear box layout.
2. Study the types of Gears, Bearings and its location.

Phase II:

1. Make 3 D modeling of Casing in Pro-E (WF3.0)
2. Mesh Generation
3. Establish boundary condition.
4. Mass and Inertia Properties calculation for Casing.

Phase III**:**

1. Analytical calculation for Gear Forces.
2. Analytical calculation for Bearing Reaction

Phase IV**:**

1) Torsional and Bending mode Calculation of Casing
2) Vonmises Stress and Deformation calculation using Bearing Reaction and Bolting Torque.

Phase V:

1) FFT equipment study.
2) Mode shape Calculation using FFT.
3) Error Estimation and Comparison with ANSYS output.

Phase VI:

1) Mode shape calculation for different layout and Nos. of Stiffeners.
2) Comparison with Resonance frequencies / Gear Meshing frequencies.
3) Selection of optimum no and layout of stiffeners based on Dynamic Vibration.
4) Re FEA for the optimum structure.
5) Report generation and future scope defining.

V. ANALYSIS METHODS

The project is divided into two domains:

1. Modal Analysis
2. Stress analysis

Modal Analysis: The natural frequencies of model in free-free conditions are calculated using Ansys10, and by applying the boundary conditions also to compare with experimental and operating frequencies.

Stress Analysis: The static analysis of the model is performed
by applying boundary conditions and forces which are calculated according to the data provided by the company.

1.1 Model analysis

Modal analysis is a term used to describe any of the processes employed to extract a structure's modal properties (natural frequencies, modal damping factors, and mode shapes) from information about the structure that is presented in a different format. When these properties are extracted from a theoretical analysis of the dynamic behavior.

For example, the tuning fork shown in Fig 1 is a very simple structure. By recording its FRFs at various points, the results shown in Fig. 1 are obtained.

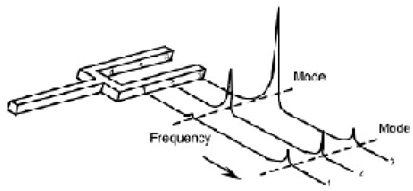

Fig 2- Modal analysis and mode shape of tuning fork

1) Modal analysis will help you reduce the noise level emitted by your product.

2) Modal analysis assists in pointing out the reasons of vibrations because cracking issues of components.
3) Modal analysis can improve the overall product performance in specific operating conditions

The model will be created in Pro-E Wildfire 3.0 and meshed using Hyper mesh 11.0. The Stress analysis and natural frequencies of model in free-free conditions are calculated using ANSYS 12.0. Experimental modal analysis (EMA) of actual component in free-free conditions will be done using FFT analyzer. The EMA determines natural frequency, mode shapes and damping ratio. From the experiment, natural frequencies of the component will be calculated. The natural frequencies will be compared with ANSYS 12.0 results which validates the model created in Pro-E Wildfire 3.0. There are five different sets of stiffeners provided on both sides of casing to reduce vibration, noise and stress. Each set of stiffener is removed and its natural frequency is evaluated**.**

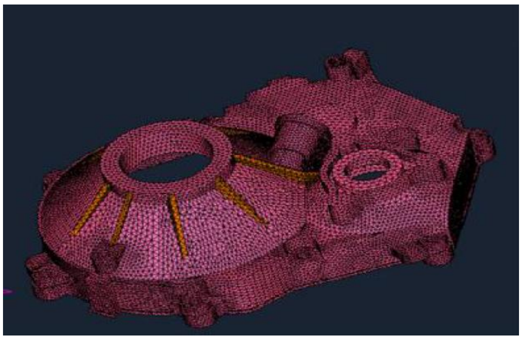

Fig. 3- Finite element model

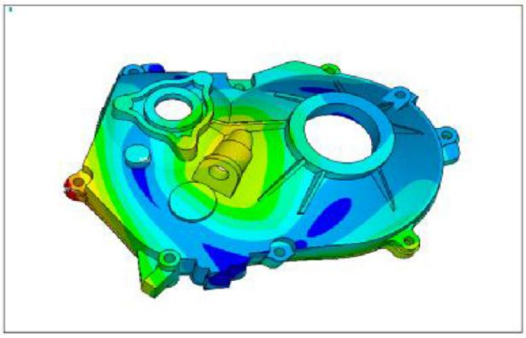

Fig. 4- Mode shape with all stiffeners

Modal analysis had performed in free-free condition, to find out first 10 natural frequencies of the model. Block Lanczos method is used to solve the basic equation.

Mode	Frequency (Hz)
1	0.0000
2	0.0000
3	0.0000
4	0.5583E-03
5	0.1389E-02
6	0.1489E-02
7	915.7078
8	1212.2173
9	1583.1551
10	1856.8609

Table 1- Frequencies of model with all stiffeners

The resonant conditions are evaluated with forcing frequencies [2]. The forcing frequencies in this case are:

1.1.1 Gear mesh frequency

This is the frequency most commonly associated with gears and is equal to the number of teeth on the gear multiplied by the actual running speed of its shaft. A typical gearbox will have multiple gears and therefore multiple gear meshing frequencies. A normal gear mesh signature will have a low-amplitude gear mesh frequency with a series of symmetrical sidebands, spaced at the exact running speed of the shaft, on each side of the mesh components. The spacing and amplitude of these side bands will be exactly symmetrical if the gearbox is operating normally. Any deviation in the symmetry of the gear mesh signature is an indication of incipient gear problems. Fig. 2 shows a diagram of a basic test system configuration.

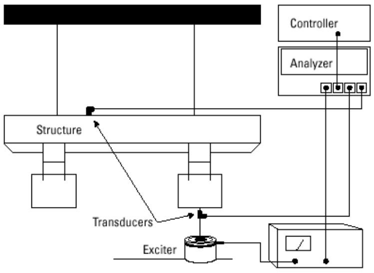

Fig .5- General test configuration

From above only the effect of gear mesh frequency is considered [3]. The gear mesh frequency is considered for following conditions:
1. Idling
2. Cruising
3. Maximum speed

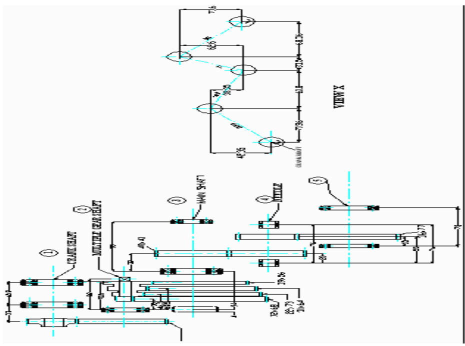

Fig.6- Gear train arrangement

Gear mesh frequency (F)= k* (N/60) Hz
Where,
k= number of teeth on gear
N= speed of the rotating shaft (on which gear was mounted)

Therefore the general expression for fundamental gear meshing frequency and higher harmonics is as follows:

Fn =nk* (N/60) Hz

Where n=1, 2, 3, 4…………..corresponding to the harmonics.

The goal of modal analysis in structural mechanics is to determine the natural mode shapes and frequencies of an object or structure during free vibration. It is common to use the finite element method (FEM)[4] to perform this analysis because, like other calculations using the FEM, the object being analyzed can have arbitrary shape and the results of the calculations are acceptable. The types of equations which arise from modal analysis are those seen in eigensystems. 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.

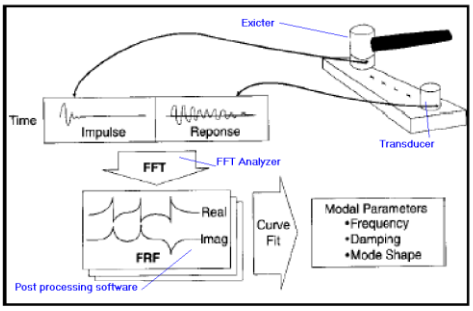

Fig. 7 Experimental Modal Analysis

It is also possible to test a physical object to determine its natural frequencies and mode shapes. This is called an Experimental Modal Analysis. The results of the physical test can be used to calibrate a finite element model to determine if the underlying assumptions made were correct (for example, correct material properties and boundary conditions were used).

1.1.2 Experimental Results

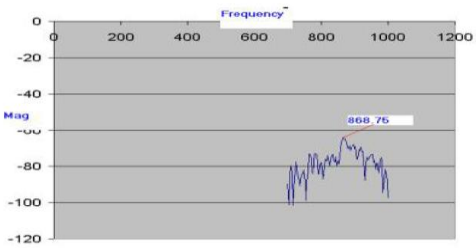

Fig. 8- Graphical results

Fig. 9- Graphical results

Mode	Ansys Frequency (Hz)	Experimental Frequency (Hz)
1	915.70	868.75
2	1212.20	1275.00
3	1583.15	1543.75
4	1856.86	1775.00

Table 2- Comparison of Ansys & Experimental Results

1.1.3. Methodology to optimize stiffeners

The second aim of this project was to find the set of stiffeners of which the natural frequency coincides with the excited frequency i.e., gear mesh frequency which may cause resonance. The set which was found will not be recommended for the design. The gear mesh frequency which was found by calculation will be multiplied with series of natural frequencies 2, 3, 4, 5 and 6 the result obtained will be considered as excited frequency.

Conditions while performing modal analysis,

1. Should not run under free-free condition.
2. Stiffeners must be removed which are only one on either side.
3. Removing stiffener must follow a proper logic as mentioned in Literature.

Fundamental frequency (f)	II 2 x f	III 3 x f	IV 4 x f	V 5 x f	VI 6 x f
457	914	1371	1828	2285	2742
838	1676	2514	3352	4190	5028
1182	2364	3546	4728	5910	7092
1486	2972	4458	5944	7430	8916

Table 3- Operating frequencies (Gear mesh frequency)

The following twelve combinations of stiffeners had been considered for the optimization of stiffeners, the method of removing stiffeners had done as per the existing literature study.

Mode	A	B	C	D
case1	8	5	5	5
case2	6	5	5	5
case3	4	5	5	5
case4	8	4	5	5
case5	6	4	5	5
case6	4	4	5	5
case7	8	5	4	5
case8	6	5	4	5
case9	4	5	4	5
case10	8	4	4	5
case11	6	4	4	5
case12	4	4	4	5

Table 4- Various set of stiffeners considered for optimization

Modal analysis had carried out for all the above twelve cases by applying the boundary conditions to find the fundamental frequencies.

Mode	1	2	3	4	5	6	7	8	9	10
8555	2113	2769	4491	4786	5394	5757	5913	6166	6808	6851
6555	2112	2768	4489	4787	5398	5759	5919	6167	6814	6852
4555	2110	2732	4478	4786	5389	5749	5918	6161	6730	6844
8455	2113	2769	4491	4786	5392	5749	5907	6161	6808	6850
6455	2110	2721	4470	4784	5344	5745	5904	6157	6728	6843
4455	2110	2721	4470	4784	5344	5745	5904	6157	6728	6843
8545	2112	2768	4484	4786	5392	5757	5911	6166	6796	6850
6545	2110	2731	4469	4786	5389	5745	5916	6160	6721	6844
4455	2110	2721	4470	4784	5344	5745	5904	6157	6728	6843
8554	2110	2761	4471	4787	5394	5758	5897	6166	6805	6850
6554	2110	2759	4462	4788	5404	5762	5903	6165	6805	6848
4554	2108	2724	4456	4787	5388	5749	5903	6160	6728	6843

Table 5- Fundamental frequencies for different cases

The fundamental frequencies in the Table 5 had compared with exited frequencies in Table 3. After comparison it is observed that cases 2, 4 and 7 are subjecting to resonance.

But one more attempt had made by performing modal analysis by removing all the stiffeners from the existing model.

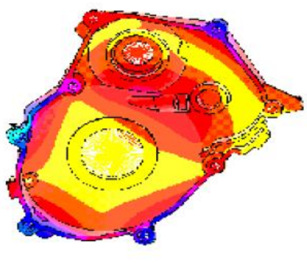

Fig. 10- Mode shape without stiffeners

Mode	Frequency (Hz)
1	0.0000
2	0.0000
3	0.0000
4	0.0000
5	0.8674E-03
6	0.1042E-02
7	843.0098
8	1152.3987
9	1429.1024
10	1696.5263

Table 6- Frequencies of model without stiffeners

It is observed that variation in the natural frequency from existing model to model without stiffeners is less and also they are not matched with the operating frequency. So, static analysis has to be carried out to confirm safety in stress parameters.

1.2 Stress analysis

Force calculations on various gears will be done. The tangential force, radial force and thrust load on the gear will be calculated and gear mesh frequency will be computed. The gear mesh frequency is considered for different conditions like idling, cruising and maximum gear speed. Natural frequency of casing with constraints is obtained using ANSYS 12.0 and compared with gear mesh frequency. Analysis will be done for actual model and proposed model and stress levels have to be compared.

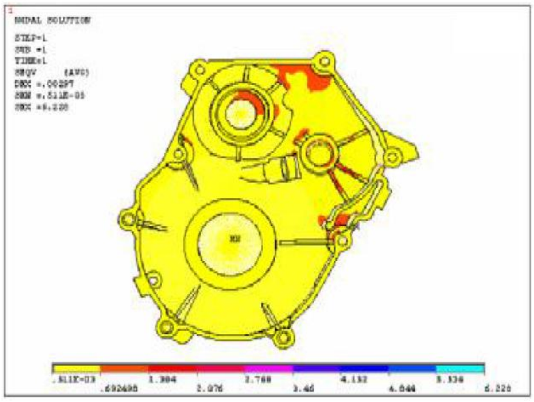

Fig. 11- Static analysis with all stiffeners

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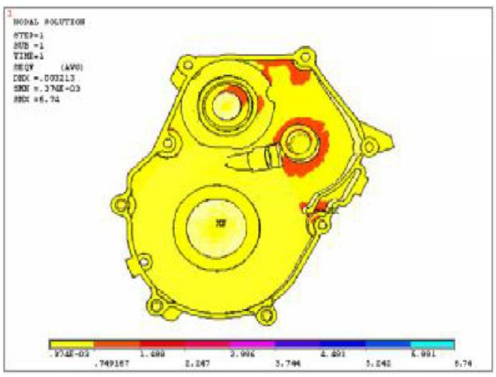

Fig. 12- Static analysis without stiffeners

VI. RESULTS AND VALIDATION

Change in Natural frequency is observed as follows for the existing model and for the proposed model:

Mode	Existing model frequency (Hz)	Proposed model frequency (Hz)
1	915.70	843.01
2	1212.21	1152.4
3	1583.15	1429.1
4	1856.86	1696.5

Table 7-Comparison of Natural frequencies for existing and proposed model.

Change in stress values are observed as follows for existing and proposed model:

Type of Model	Stress (Mpa)
Existing	6.228
Proposed	6.74

Table 8- Comparison of stress for existing and proposed model.

VII. CONCLUSION

In this work casing of differential gearbox is analyzed.

1. Then natural frequency of casing with constraints is obtained using ANSYS and compared with gear mesh frequency i.e., operating frequency. There is no resonance condition found.
2. Natural frequency of the model without stiffeners had compared with the natural frequency of model with all stiffeners and with operating frequency. Proposed model is safe from resonance point of view.
3. As the stiffeners are removed from casing there is a chance of increase in stress, from static analysis results we conclude that proposed model is in safe stress limit.
4. So, model without stiffeners had proposed for further analysis and judgments.

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