Vibration Analysis of Gearbox Top Cover

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Abstract- Gearbox top cover is a vital component of horizontally split integrally geared centrifugal compressor. Reliability of top cover is directly related to the structural rigidity and structural integrity of compressor. Assuming that the top cover is designed within the mechanical limits of its construction material, one of the most common causes of top cover failure is resonance. The aim of the designer while designing top cover is to have minimum weight and still have sufficient rigidity to avoid resonance. It is imperative; however, that rigorous engineering analysis is applied in the initial design phase to prevent resonance that will result in failure of gearbox top cover.

To meet the objectives of designing a reliable integrally geared centrifugal compressor, gearbox top cover modal analysis is of utmost importance. Numerical modal analysis, usually based on Finite Element Method, is commonly used to determine the vibration characteristics, such as natural frequencies and the associated mode shapes of a structure.

This paper addresses finite element analysis and modal testing of gearbox top cover of an integrally geared centrifugal compressor. The goal is to determine, verify and validate the vibration characteristics of the gearbox top cover using both analytical and experimental techniques.

The 3-D solid model of the gearbox top cover was built using Autodesk Inventor®. ANSYS workbench 13® was used for preprocessing, solving and post processing. Modal analysis involved extraction of mode shapes and natural frequencies. Impact testing using hammer excitation was used to experimentally determine the vibration properties of the gearbox top cover.

The results obtained by FEA modal analysis are compared and validated against modal test data. The results indicate that the natural frequencies of gearbox top cover predicted by FEA are within 8 percent of the measured natural frequencies of the modal test data, thus confirming the close agreement between FEA and experimental data.

Finally a parametric study was conducted using FEA, to study the effect of wall thickness and rib thickness of gearbox top cover on its natural frequencies and associated mode shapes. It was found that the effect of varying casing wall thickness on the natural frequencies of the gearbox top cover was not significant, as compared to increase in the weight of the gearbox top cover. However, the effect of addition of rib on the natural frequencies of the gearbox top cover was significant, as compared to increase in the weight of the gearbox top cover.

Key Words: FEA, Modal Analysis, Gearbox top cover, Natural Frequency, Mode Shape, Modal test, Centrifugal Compressor.

I. Introduction

The top cover of gearbox is an important component in an integrally geared centrifugal compressor. The function of gearbox casing is to accommodate and support power train and aero components of centrifugal compressor. Since the gearbox application is for very high speed (typically 50,000 rpm to 70,000 rpm), failure of casing may lead to major primary damage of the compressor. Analysis of compressor top cover is very essential in order to decide appropriate dimensions and to predict the behavior of top cover under different operating conditions. Several authors have analyzed using commercial softwares as well as experimentally, few of them presented as follows. Choy F.K et. al. [1] have compared gearbox mode shapes and vibration predictions obtained from analytical model with those of experimental results obtained from the test rig at the NASA Lewis Research Center. 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.

Khobragade T.N. et. al. [2], have statically analyzed gearbox casing using two simulation software Altair Hypermesh / OptiStruct and ANSYS 11.0. They have compared the results obtained from both the software. The comparisons obtained were closely matching. The results obtained from simulation were used in developing an optimum design of gearbox casing. It has not only helped in building cost effective design, but also have helped reduced prototype development and testing time. This explains why the vibration characteristics of casing have been studied in such great detail. However, greater knowledge of the vibration behaviour of gearbox top cover will be useful for optimising the top cover, possibly resulting in improved reliability and reduced downtime of compressor.

The particular case study used for this paper is a gearbox top cover as shown in Fig. 1(a). The top cover, which is part of the horizontal split, intergrally geared centrifugal compressor, is cast in spheroidal graphite cast iron i.e. Nodular Iron. In general terms, the investigation has been divided into two main sections: 1- Modal analysis using FEA, and 2 – Modal testing.

The finite element method, which originated in the field of structural analysis, was widely developed and exploited in the aerospace industry during 1960s and 1970s. Information on this and other advanced topics may be found in [3-7]. Due to lack of confidence in FE models, the dynamic testing of structures has become standard procedure for model validation and updating [8]. Over the past fourty years, modal testing and analysis have become a fast developing technique for the experimental evaluation of the dynamic properties of structures [8]. The experimental studies using modal analysis were mostly used to validate the Finite element analysis results.

Before a modal test is performed, there are some features that should be carefully selected [9]. In this way, the test can be carried out effectively as data from the test contain more useful information. Among these considerations, selecting suspension points, driving point(s) and response points are most important, especially for a modal test for model validation [10]. Test planning is undertaken to help decide the optimum conditions for the test set-up i.e. to determine how and where to suspend the test structure for excitation, and how to choose the response DOFs to be measured. Sometimes all these choices can be made using test engineer's experience or intuition.However, for a test structure with complex shape and complex configuration, relying on experience and intutition is not enough to achieve the optimum conditions especially when encompassing a wide range of frequency.

II. Modeling and Finite Element Analysis of Gerabox Top Cover.

The 3-D solid model of the gearbox top cover was build using Autodesk Inventor®. ANSYS workbench 13® was used for preprocessing, solving and post processing. Material properties of spheroidal graphite cast iron, grade EN-GJS 600-3 were selected from DIN EN 1563:1977 [11]. Solid 3-D model of the gearbox top cover was meshed using Solid 187 element (3-D 10-Node Tetrahedral Structural solid element). The FE model consists of 78,749 elements. Next, Block Lanczos technique was used to extract first 11 natural frequencies and mode shapes of the model.

(a) (b)

Figure 1. (a) 3D model of Top Cover (b) Top cover mesh using solid element

Table 1 Lists the results of FEA obtained for the gearbox top cover in a free-free boundary condition (rigid body modes were excluded), while the modes shapes are depicted in Fig. 2.

Table -1 Natural frequencies of the gearbox top cover

Mode No.	Mode Description	Natural frequency (Hz)	Figure
1	**Bending**	**125.47**	**a**
2	**Combined Bending And Twisting**	**165.12**	**b**
3	**Bending**	**216.65**	**c**
4	**Combined Bending And Twisting**	**320.50**	**d**

5	**Bending**	**367.97**	**e**

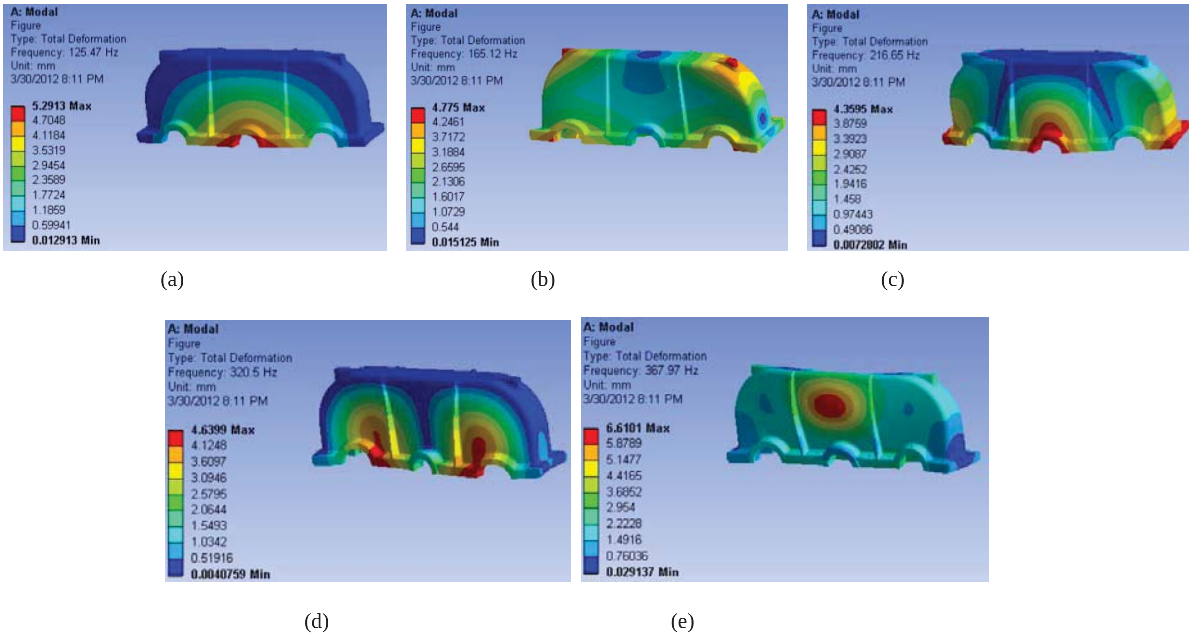


Figure 2.(a) to (e) - Mode shapes of top cover obtained from ANSYS workbench 13®

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 [12], excitation due to mechanical looseness and unbalance is not considered.

The second type of excitation is due to gear tooth meshing. However as excitation frequency due to gear tooth meshing is 21730 Hz, which is very high and away from natural frequency, hence excitation corresponding to gear mesh frequency is not considered.

Third type of excitation force is due to electrical motor, which is running at 2970 rpm. In order to prevent resonance of the gearbox top cover, it is expected that natural frequency of the gearbox top cover should have a minimum seperation margin of 20% [13], from first two harmonics of the exciting frequency, i.e. of electrical motor speed. The frequencies of first and second harmonics of motor speed considering 5% frequency variation of input power are 51.95 Hz and 103.95 Hz.

In case of modal analysis, for any one harmonic of excitation source, there can be three situations where first two natural frequencies of the component can lie. First case is first two natural frequencies are below the harmonic of excitation source. Second case is first natural frequency above and second natural frequency below the harmonic of excitation source. Third case is first two natural frequencies are above the harmonic of excitation source. Here, all the cases are acceptable, as far as these natural frequencies of the gearbox top cover are separated by 20 % from first two harmonics of excitation frequencies.

From the FEA results, it is evident that the natural frequencies of the mode shape 1 and mode shape 2 of the gearbox top cover is seperated from first and second harmonics of the excitation frequency of motor speed by more than 20%. Hence resonance is not likely to occur.

III. Modal 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 dymanic properties.

(i) 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 [8, 10, 14-15]. 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 [8, 10, 14-15]. 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 [16]. 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.

(ii) 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.

(iii) 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. On the other hand, the number of measurements cannot be enlarged without reaching practical limits. In order to fullfill 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 14 points - Impact points 35 to 48, were selected as measurement locations as shown in Fig. 3. Accelerometer was mounted on point number 43.

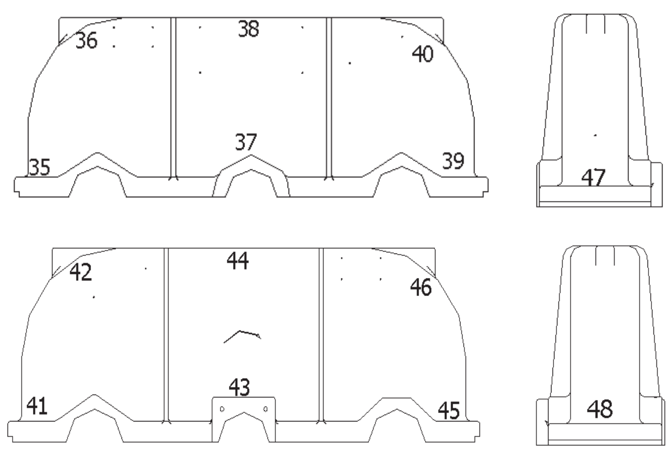


Figure 3. Measurement locations on gearbox top cover.

3.2 Modal testing and modal analysis

(i). Hammer Testing: The gearbox top cover was suspended by using the flexible ropes having least stiffness to achieve nearly free-free condition. The experimental data were obtained via hammer (Model 2302-10, S/N 3588, make – Megigit Endevco) testing using a B&K 3560 analyser connected to a PC. One B&K piezoelectric accelerometer was attached to point 1 on the structure. The overall arrangement was as shown in Fig. 4

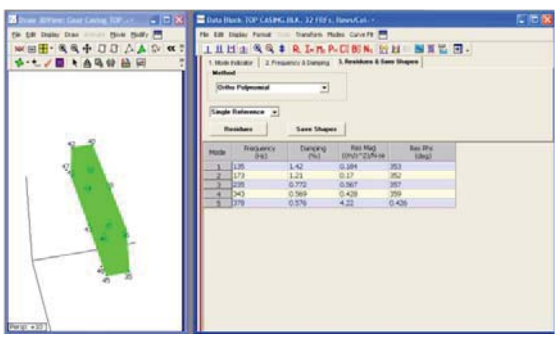
(a)

(b)

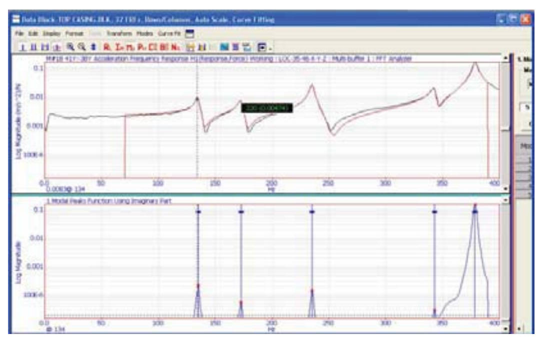
Figure 4. (a) Photograph of Modal Testing (b) Photograph of Data Aquistion System.

The measurements were carried out by impacting the structure at point 35 to 48 in the X/Y/Z direction as per predefined location / co-ordinate system and acquiring the response data at point number 43.

(ii). Modal Analysis Results: The impact points are further plotted in MEscope® to generate the structure (The MEscope® 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 MEscope, with the location of points, as that of the actual component test points i.e. points tested on actual structure by using FFT analyzer along with data acquisition system. During the analysis first five natural frequencies of the gearbox casing are extracted from the data as seen in Figure 5.



(a)



(b)

Figure 5. (a) Surface Model from MEscope® for top cover (b) Natural Frequencies Plot from MEscope for top cover.

Natural Frequencies are as shown in Table 2.

Table - 2 Experimental Modal Results for top cover

Mode No.	Mode Description	Frequency (Hz)	Figure
1	**Bending**	**135**	**a**
2	**Combined Bending And Twisting**	**173**	**b**
3	**Bending**	**235**	**c**
4	**Combined Bending And Twisting**	**343**	**d**
5	**Bending**	**378**	**e**

The mode shapes obtained from MEscope is as shown in Figure 6

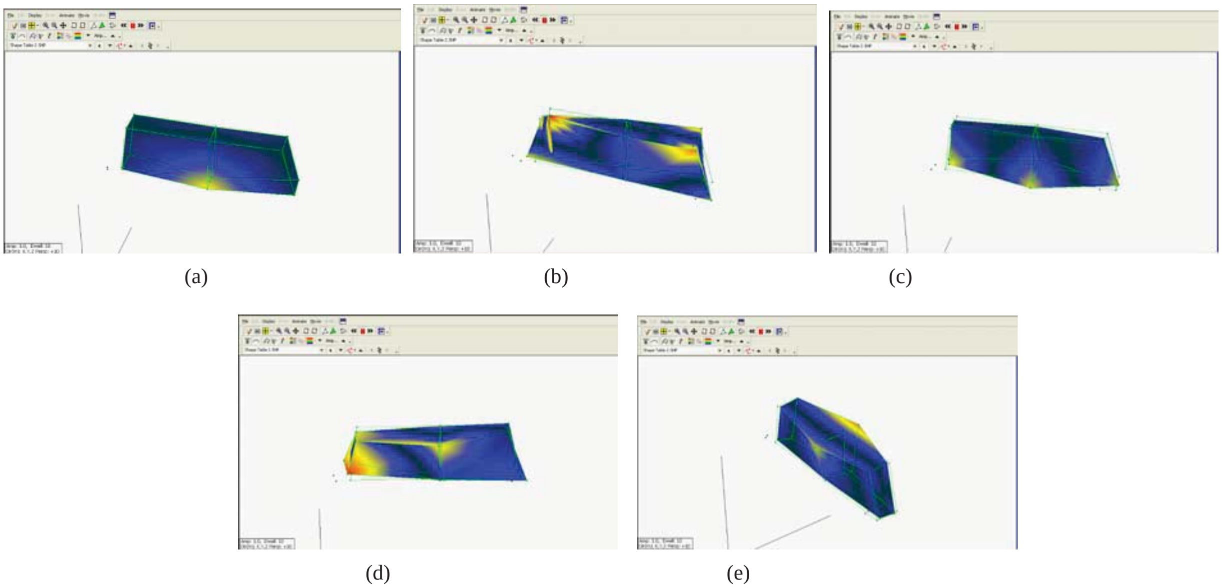


Figure 6. (a) to (e) Mode Shapes as obtained from MEscope software.

(iii). Correlation of experimental and FEA results: In FEM modal analysis mass of the structure is very important, which decides the frequency of structure. 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 top cover was 182.60 kg and that of model in the ANSYS® was 180.97. 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 gearbox top cover, with respect to the mass.

The comparison of the natural frequencies is shown in Table 3.

Table - 3 Comparison of Frequencies for Gearbox Top Cover (Modal Test data with FEA results)

Sr. No.	Natural Frequency – Modal Test data (Hz)	Natural Frequency - FEA (Hz)	Percentage Deviation
1	135	125.47	7.05
2	173	165.12	4.55
3	235	216.65	7.80
4	343	320.50	6.55
5	378	367.97	2.65

The above table 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 13 ® are within 8 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.

IV. 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 needs to be updated. The frequency can be altered by following two ways:-

1) Changing the mass of the casing:-

$$Frequency \ \alpha \ \frac{1}{\sqrt{mass}}$$

2) Changing the stiffness:-

$$Frequency \ \alpha \ \sqrt{stiffness}$$

An investigation was carried out to study the effect of natural frequency and weight due to modification of top cover wall thickness, addition of rib and modification of rib thickness.Such sensitivity analysis is necessary for design stage of the gearbox top cover.

a) Effect of varying Top cover wall thickness: - The first parametric study was devoted to the gearbox top cover wall thickness. For this case, the top cover wall thickness was changed to 1.3 times the normal thickness. It was found that the variation of frequency is not significant (3 %), as compared to significant changes in weight (13 %) of the gearbox top cover.

b) Effect of addition of extra rib: - In this case an additional rib was added on the gearbox top cover. It was found that the variation of frequency is significant (3 %), as compared to not so significant changes in weight (7 %) of the gearbox top cover.

c) Effect of varying rib thickness: - In this case, rib thickness was varied upto 2 times the normal thickness. It was found that the variation of frequency is not significant (3 %), as compared to significant changes in weight (9 %) of the top cover.

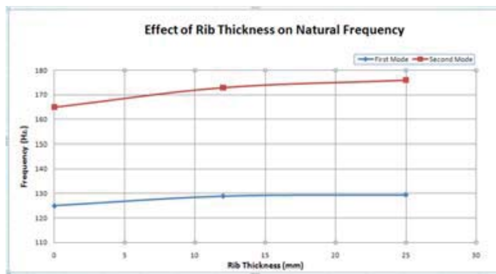


Figure 7. Graph Showing Effect of Rib Thickness on Natural Frequency.

Table – 4 Sensitivity of gearbox top cover natural frequencies with respect to differenet parameters

Methodology 1 :- Carrying Modal Analysis after Increasing Thickness of the Casing			
Sr. No.	**Description**	**Current**	**Proposed**
1	**Casing Thickness (mm)**	**40**	**52**
2	**First Mode Frequency (Hz)**	**125**	**129**
3	**Second Mode Frequency (Hz)**	**165**	**176**
4	**Weight (Kg)**	**181**	**204**

Methodology 2 :- Carrying Modal Analysis after Addition of 12 mm Thickness Rib Above the Bore			
Sr. No.	**Description**	**Current**	**Proposed**
1	**Rib thickness above bore (mm)**	**-**	**12**
2	**First Mode Frequency (Hz)**	**125**	**129**
3	**Second Mode Frequency (Hz)**	**165**	**173**
4	**Weight (Kg)**	**181**	**194**

Methodology 3:- Carrying Modal Analysis after increasing the rib thickness.			
Sr. No.	**Description**	**Current**	**Proposed**
1	**Rib thickness above bore (mm)**	**-**	**25**
2	**First Mode Frequency (Hz)**	**125**	**129**
3	**Second Mode Frequency (Hz)**	**165**	**175**
4	**Weight (Kg)**	**181**	**198**

In order to study the effect of selected element size in ANSYS workbench 13® on natural frequency of structure and required computational time, modal analysis was carried out consiering default, 5 mm, 7 mm and 10 mm element size. The results from this mesh sensitivity study revealed that for less than 1 % change in frequency results, computational time was increased by about 35 times, as seen in Table 6.

Table - 6: Effect of selected element Size in ANSYS workbench 13 on frequency and computational time for gearbox top cover.

Element Size (mm)	**Frequency (Hz)**		**Computational Time (Minutes)**
	Mode 1	**Mode 2**	
Default (Reference)	**125.47**	**165.12**	**1.1**
10	**124.83**	**164.12**	**7.3**
7	**124.75**	**164.04**	**13.1**
5	**124.70**	**163.99**	**35.6**

V. CONCLUSIONS

Analysis has been carried out to examine in detail the vibration characteristics of the Top cover of intergrally geared centrifugal compressor. The findings are as follows:-

(a) FEM Modal Analysis for Gearbox Casing:

FEM Modal analysis for gearbox casing is carried out using ANSYS Workbench 13® 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 8 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.

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 top cover for centrifugal compressor shows close agreement with the experimental modal test data.

A parametric study was conducted on casing thickess, addition of rib, increasing rib thickess and changing element size in ANSYS workbench 13®

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

(ii) The effect of addition of rib on the natural frequencies of the gearbox top cover was significant, as compared to increase in the weight of the gearbox top cover.

(iii) The effect of increase of rib thickness on the natural frequencies of the gearbox top cover was not significant, as compared to increase in the weight of the gearbox top cover.

(iv) The effect of use of smaller element size in ANSYS Workbench 13®, 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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