Defence Science Journal, Vol. 68, No. 6, November 2018, pp. 604-609, DOI : 10.14429/dsj.68.12705

Experimental and Numerical Investigation of Engine Foundation for Vibration Reduction

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ABSTRACT

The purpose of this study is to minimise frequency response of engine foundation using topology optimisation. The study involves vibration response estimation of an existing marine engine foundation, validation of estimations with measurements and estimation of reduction in vibration response after optimisation. Initially, solid model of baseline model is generated using dimensions of the existing foundation measured by a laser line probe coordinate measuring machine. Harmonic analysis is used to find the vibration response of the foundation. These results are experimentally validated by the measurements on the foundation using the vibration testing. Frequency response topology optimisation is then carried out on the baseline model to reduce vibration response with specified constraints and objective function. Subsequently, harmonic analysis is performed on the topology optimised design to verify the reduction in vibration response. From these results, it is observed that considerable frequency response is reduced with modified design compared to baseline model.

Keywords: Engine foundation; Frequency response; Topology optimisation; Vibration reduction

1. INTRODUCTION

Vibrations of an engine transmit through its foundation to the hull on a marine platform. Hull vibration of a marine vessel results in underwater radiated noise leading to its vulnerability. It is therefore important to optimise the foundation design for reduction of vibration transmission. Various algorithms[1-4] were developed such as Kriging-interpolated level-set approach and the multi-objective genetic algorithm, gradient-based optimisation and fruit fly optimisation algorithm for structural optimisation. Yang[5] used video-based remote vibration measurement system to understand the structural behaviour and included the motion and strain distribution. Daee[6] presented finite element analysis of structural performance of an innovative noise barrier consisting of poly-block, rigid polyurethane foam and polyuria and these results were validated using measurements. Xu & Jin[7] investigated the static and dynamic multi objective topology optimisation of trusses with interval parameters. Pareto constrained multi-objective genetic algorithm was adopted to solve the constrained multi objective optimisation problem. Liu and Novak[8] investigated the dynamic behaviour of a turbine generator foundation system. Shu[9], *et al*. studied minimisation of frequency response at the specified points or surfaces on the structure within an excitation frequency range, subject to the constraint of given amount of the material over the admissible design domain.

Topology optimisation[10-14] is used to suppress the vibrations of an elastic aero elastic structures, diesel generator foundation and engineering structures. Zhang[15], *et al*. Proposed active control of dynamic frequency response to control mechanical vibration. Ramakrishna[16-18], *et al*. studied the free size optimisation, rib optimisation and frequency response optimisation of the marine engine foundation using genetic algorithm optimisation tool. Review of recent literature related to vibration reduction suggests that it is possible to reduce vibrations by modification of baseline design using structural topology optimisation. However, have been carried out not many research studies about frequency response topology optimisation for reducing vibration transmission. The objective of the present study is to improve the stiffness of the baseline model by modification of the design so as to minimise vibrations using frequency response topology optimisation. A geometric surface model with shell finite elements is used for frequency response topology optimisation with given constraints to determine the optimal distribution of material for reducing the frequency response.

The optimisation of baseline marine engine foundation is performed in three stages in order to reduce the frequency response, improve the structural stiffness and reduce the weight. In the first stage, a surface model is used for free-size optimisation to determine the material thickness distribution of marine engine foundation. In the second stage, a solid model of the reference marine engine foundation is used for

Received : 14 February 2018, Revised : 19 September 2018
Accepted : 09 October 2018, Online published : 31 October 2018

optimal distribution of supporting ribs for improving the stiffness in the marine engine foundation. In the final stage, a geometric surface model with shell finite elements is used for frequency response topology optimisation with given constraints to determine the optimal distribution of material for reducing the vibration response.

2. MODEL OF EXISTING MARINE ENGINE FOUNDATION

The dimensions (L =3140 mm, W =1330 mm, H = 926 mm) of the existing marine engine foundation (baseline model) were obtained from the reverse engineering method. Co-ordinate measuring machine (Cmm) was used to obtain the actual dimensions of the marine engine foundation. Cmm is directly coupled to a data acquisition system to store all data and to generate solid models using in built software in the data acquisition system. Figure 1 shows a solid model of baseline foundation obtained from CMM. The solid model was exported to finite element analysis package to carry out static and dynamic analysis on the baseline model to know the strength and vibration characteristics. With the density specified, the overall mass of solid model of the baseline model was estimated as 790.5 kg, which compares closely with the measured mass of 787.5 kg of the existing marine engine foundation.

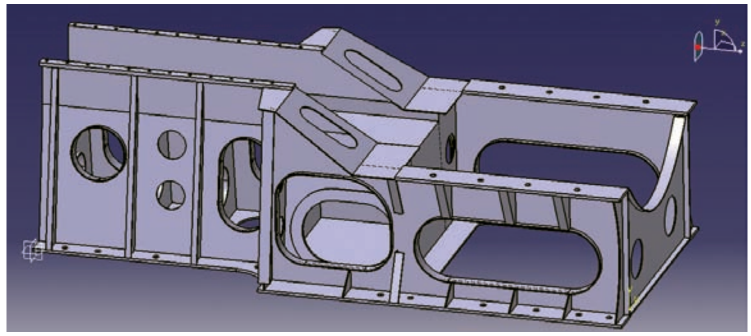

Figure 1. Solid model of existing marine engine foundation.

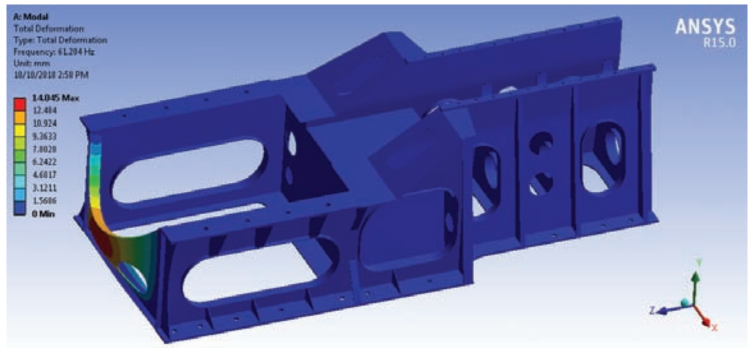

Figure 2. First mode of baseline marine engine foundation.

3. STATIC AND MODAL ANALYSIS ON BASELINE MODEL

ANSYS Workbench was used to carry out the static and modal analysis on the solid model of existing marine engine foundation. This model was exported into ANSYS in IGES format. Meshing is an important step in the finite element analysis. This model was meshed with four different mid side node tetra elements (sizes: 50, 45, 40 and 35) for grid independence study. From this study convergence was observed with element size 35. Generator (3,960 kg) and diesel engine (3,100 kg) were applied on the top face of the baseline model as masses. The bottom of the baseline model was fixed to another frame on the hull of the ship. From the static structural analysis, maximum von-mises stress obtained was 83.7 mPa on the plate near the engine. modal analysis was carried out using Block Lanczos iteration on the baseline model to find out the natural frequencies and mode shapes. modal analysis was carried out using fixed boundary condition at the bottom of the foundation. From the modal analysis, it was observed that the translation mode along the length (No 2 at 126.1 Hz) is more predominant mode compared to all other modes. First tension mode along the length axis (No 1 at 61.2 Hz) and twisting mode (No 3 at 171.2 Hz) were observed to be less responsive when compared to the second mode of the structure. Figure 2 shows the first mode of baseline marine engine foundation.

4. HARMONIC ANALYSIS ON BASELINE MODEL

Harmonic analysis was carried out on the baseline model to obtain frequency response of engine foundation. The excitation force from diesel engine (having 12 cylinders and rotating at 1,500 rpm) was obtained according to ISO 1940/1 grades and manufactures catalogue. For marine diesel engines for more than six cylinder G40 grade is to be used as per ISO 1940/1 balancing grades. The total mass of piston, piston rod and crankshaft was taken as 5 kg per cylinder.

The equation for the unbalanced force as per ISO 1940/1 Grades

$$U = \left[\frac{9549\times G\times M(kg)}{N}\right]\times\omega^{2} = 37,660\ N \qquad (1)$$

A force of 37,660 N was thus applied on the top of the foundation. Figure 3 shows the frequency response of the baseline model at the bottom surface of engine in the range of 0 – 2 kHz. From Fig. 3, the maximum peak amplitude was observed at 126.1 Hz (predominant modal frequency observed in modal analysis). The response of the structure is more predominant in the frequency range of 0-750 Hz and gradually reduces after 750 Hz.

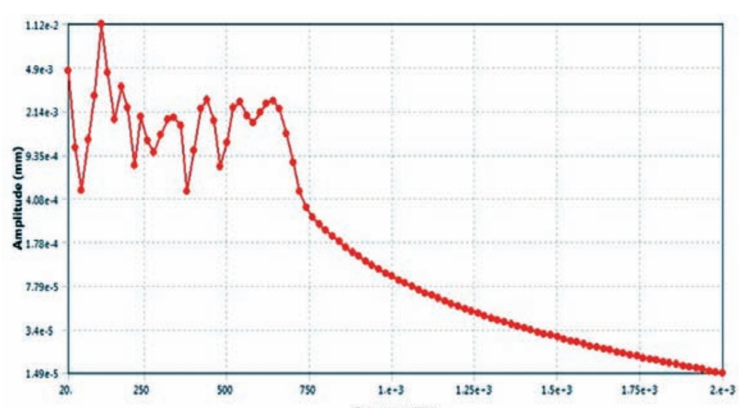

Figure 3. Frequency response of baseline model at bottom surface of engine.

5. EXPERIMENTAL STUDY

Experimental setup was used to validate the modal analysis results by sine test. Vibration testing was carried out by fixing the foundation on 16 T electro dynamic exciter system. ICP 352C04 of PCb make accelerometers were used for vibration excitation control and measurement of vibration response. They were aligned on the top and bottom surface of the baseline model in vertical direction. One accelerometer was used for control of excitation and two other accelerometers were used as response accelerometers. The methodology followed in carrying out the test is as follows:

The input excitation is generated in the controller of the exciter system. These excitations are then transmitted from the controller to power amplifier and then to the shaker armature body. From the armature, these excitations are transmitted to the foundation. The response from control accelerometer is fed back to the controller for error correction in control. Response of other two accelerometers is used for recording the frequency response of the foundation.

Figure 4 shows the frequency response plot obtained from the measurements. Figure 4 clearly shows that the first fundamental mode is at 69. 8 Hz. Table 1 shows the comparison of experimental and numerical analysis results. From these results it is observed that predicted results and experimental values are close.

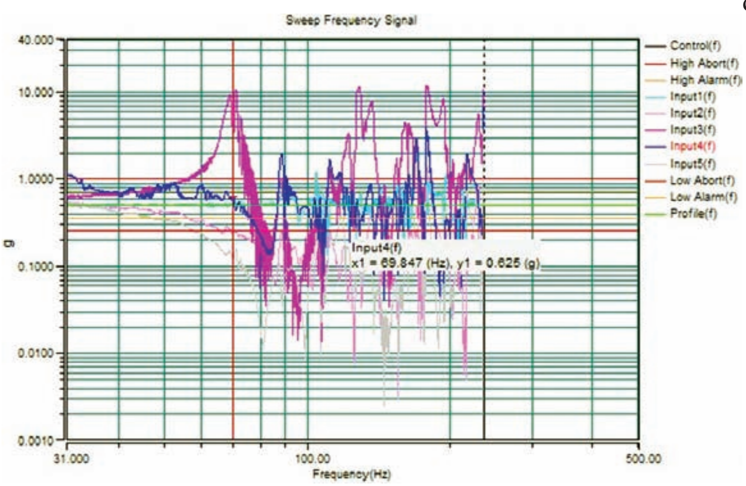

Figure 4. Frequency response plot obtained from the measurements.

Table 1. Comparison of experimental and numerical analysis results

Modes along length direction	Measurements		Prediction	
	Modes (Hz)	Acceleration (g)	Modes (Hz)	Acceleration (g)
First mode	69.8	0.625	61.2	0.524
Second mode	132.3	1.124	126.1	1.012
Third mode	183.4	0.524	171.2	0.428

6. FREE SIZE OPTIMISATION[16]

The free-size optimisation in OptiStruct software of Altair deals with the element thicknesses of the structure within a given design space. The design variable for free-size optimisation is the thickness of the shell elements on the surfaces. Thickness is allowed to vary between 3 mm and 20 mm. The lower bound for the thickness is given as 3 mm, which is the smallest thickness that can withstand the given direct loads. The upper bound for the thickness is the thickness of the reference base frame, 20 mm. A stress constraint is applied to the optimisation problem such that the stress on the base frame should not exceed the Yield Strength, which is 60 mPa. In this study a displacement constraint is also used to constrain the top most node on the reference frame to not deflect more than 0.14 mm in the direction of applied load.

Figure 5 shows the finite element model and optimum thickness distribution of surface model of baseline marine engine foundation after free size optimisation. It shows that the thickness is only 5 mm on all surfaces except the surfaces under load applied. The thickness near the load applied is 16 mm and 18 mm under the engine and the generator. It was observed that the maximum displacement obtained is 0.28 mm and maximum von-mises stress is 41.5 mPa. The total deformation and von-mises stress on baseline marine engine foundation, after free size optimisation is thus greatly reduced after optimisation. With the density specified, the overall mass of marine engine foundation, after free size optimisation in the finite element model is estimated as 695 kg, which is 92.5 Kg less than the baseline marine engine foundation, implying a reduction in weight.

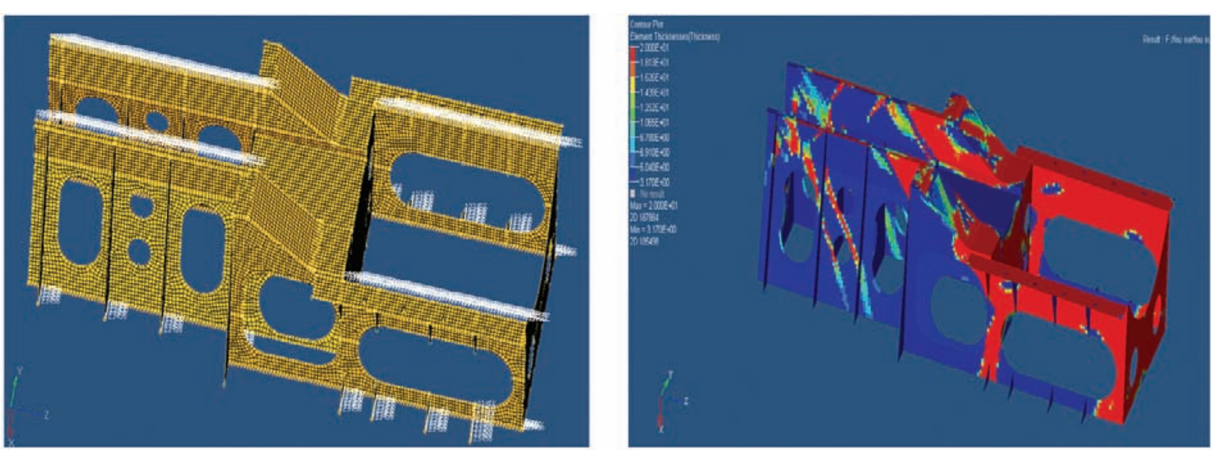

Figure 5. Finite element model and optimum material distribution after free size optimisation.

7. RIB OPTIMISATION[17]

To determine the optimal distribution of supporting ribs in the existing marine engine foundation, a solid model of the reference marine engine foundation was created in CATIA, subjected to automatic topology optimisation in OptiStruct. As per the design constraints, the areas of the mounting and certain other designated areas should not be modified after optimisation. For this purpose desirable and non-desirable areas were created in hypermesh. The objective function is set to find the minimum compliance that is the maximum stiffness of the domain. The constraint was set to use only 15 per cent of the volume of the design space, by defining a volume fraction of 0.15 to maintain the same weight after optimisation. A stress constraint was applied to the optimisation problem such that the stress in the baseline model should not exceed the yield strength, which is 60 mPa. All degrees of freedom of the nodes present at the bottom surface of the baseline model were constrained. The minimum member size was fixed at 5 mm and the baseline model was then subjected to topology optimisation. In optimised model, four and three ribs were suggested as per findings under the engine and the alternator. The middle portion between the engine and alternator was specified as non-design area in topology optimisation, so it remains same as baseline model. The final overall dimensions of the redesigned foundation are same as the existing marine engine foundation. Figure 6 shows the finite element model and optimal rib distribution after topology optimisation.

8. FREQUENCY RESPONSE TOPOLOGY OPTIMISATION[18]

Structural optimisation was performed in this study to improve the structural stiffness of marine engine foundation for vibration reduction. A stress constraint was applied to the optimisation problem such that the stress in the baseline model should not exceed the yield strength (60 mPa). The frequency range was taken as 0 to 1000 Hz for the analysis and also assumed that that load is constant in the entire frequency range. A frequency dependent dynamic load of 37,660 N was applied to the foundation. All degrees of freedom of the nodes present at the bottom surface of the baseline model were constrained. RADIOSS software was used to get the frequency response of the baseline model. From the analysis, it was observed that maximum displacement is 2.97 mm in Y-direction which is the direction applied load. Volume response was defined as the objective function and it was set to minimise the volume. In order to reduce the frequency response deflection, the deflection constraints (2.25 mm in Y direction, taking 25 per cent of that of baseline model and 1 mm for X and Z directions) were set. This model was subjected to frequency response topology optimisation.

Figure 7 shows the finite element model and element density after frequency response topology optimisation. It was observed the maximum displacement obtained is 1 mm coming down from 2.97 mm corresponding to baseline foundation in Y-direction (applied load direction). It has been found that the overall average structural stiffness of 66 per cent has been

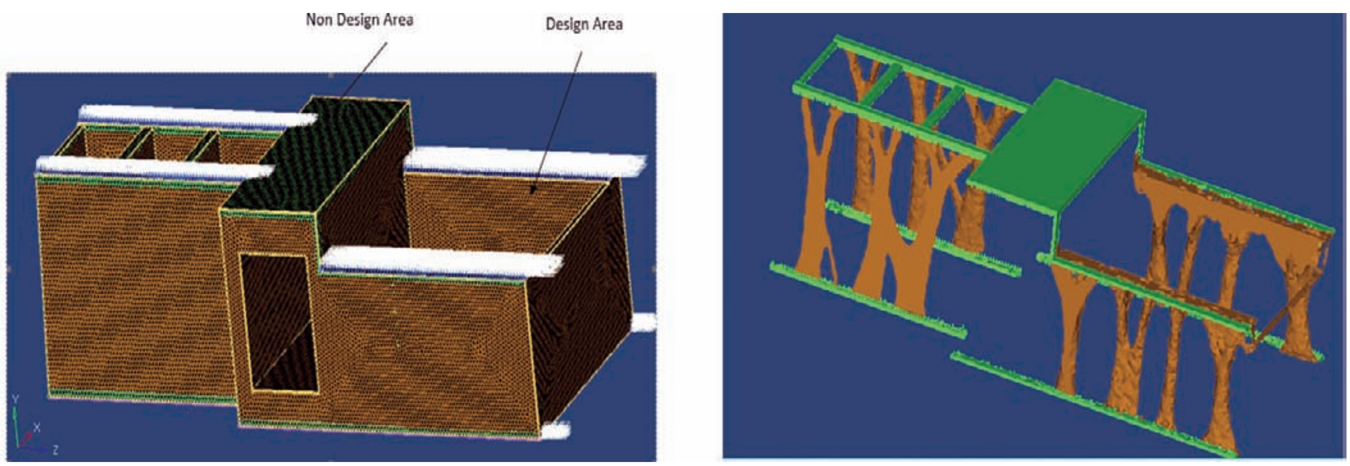

Figure 6. Finite element model and optimal rib distribution after topology optimisation.

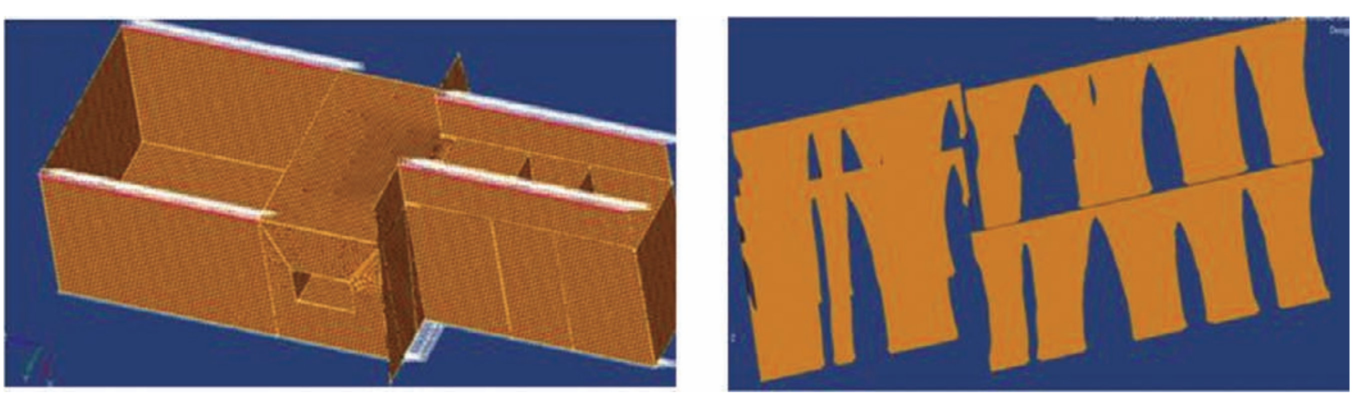

Figure 7. Finite element model and element density after frequency response topology optimisation.

improved with redesigned foundation when compared with baseline design.

9. STATIC AND MODAL ANALYSIS

After carrying out free size, rib and frequency response topology optimisation, redesigned foundation was generated considering manufacturing feasibility. The static and dynamic analysis was carried on solid model of redesigned marine engine foundation using ANSYS Workbench. The loads and boundary conditions applied to the redesigned foundation are same as the baseline model. Figure 8 shows the final model of redesigned marine engine foundation. From the static structural analysis of the redesigned foundation, the maximum von-mises stress of 46.3 mPa was observed on the plate near the engine. From the comparison of static analysis results of both baseline model and redesigned model, it is evident that the new design is safer than the baseline model under the static load conditions. From the modal analysis results, it was observed that the translation modes (mode no. 3 at 130.5 Hz and mode no. 1 at 72.3 Hz) in length direction are more predominant modes than all other modes. The translational mode (mode no. 2 at 109 Hz) along the width direction is the next predominated mode compared to other modes. Figure 9 shows the first mode of redesigned marine engine foundation. The modal displacements of all modes of redesigned marine engine foundation are considerably reduced when compared to similar modes of baseline marine engine foundation. Thus, it can be stated that the redesigned foundation has lesser modal response.

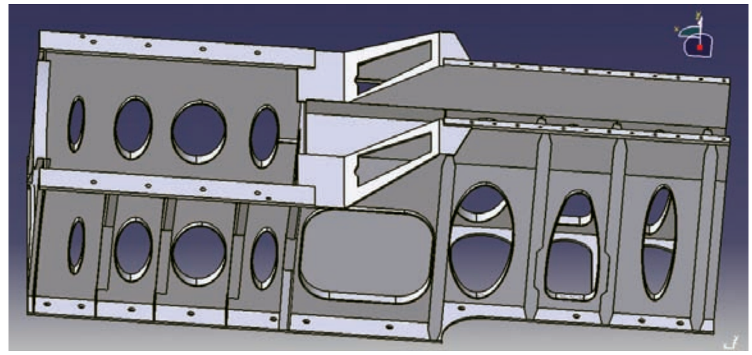

Figure 8. Final model of redesigned marine engine foundation.

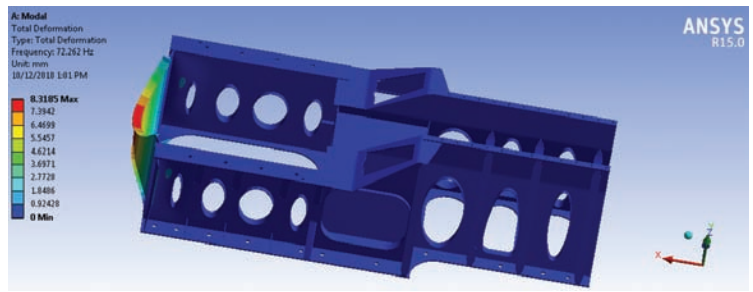

Figure 9. First mode of redesigned marine engine foundation.

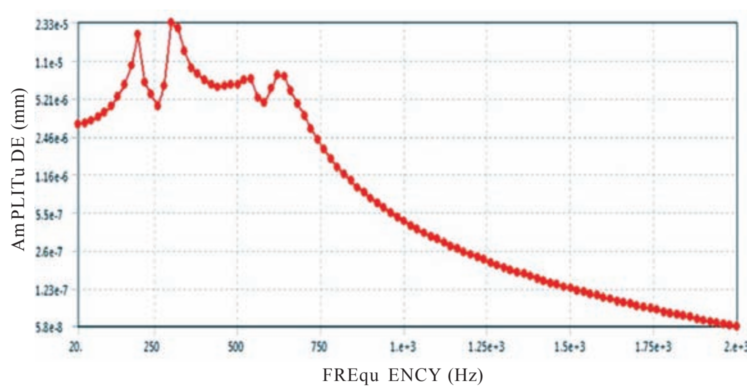

Figure 10. Frequency response of redesigned marine engine foundation at the bottom surface.

10. HARMONIC ANALYSIS

Harmonic analysis was carried out on the redesigned marine engine foundation applying the same boundary conditions as applied on existing marine engine foundation. Figure 10 shows the frequency response of redesigned marine engine foundation at the bottom surface of engine in the range of 0 – 2 kHz. From Fig. 10, maximum peak response is observed at 320 Hz (predominant mode observed in modal analysis). The response of the structure is more predominant in the frequency range of 0 to 600 Hz and gradually decreases after 600 Hz. The results show that all frequency response deflections of redesigned marine engine foundation are well below those of the baseline foundation. This response is specific to the same node considered in the baseline model at the bottom surface of the engine. Other nodes were also analysed, and they showed similar response and hence redesigned marine engine foundation can be considered safe.

11. CONCLUSIONS

A new design for the marine engine foundation was generated based on frequency response topology optimisation. methodology was developed for material distribution to minimise the frequency response deflection, improve the structural stiffness and minimise weight of the baseline model subjected to given loads and constraints. It has been found that the overall average structural stiffness has been improved by 66 per cent with redesigned foundation when compared with baseline design. Also, 12 per cent of the mass of the redesigned foundation has been reduced when compared with baseline design. The maximum von-mises stress and vibration response on the redesigned foundation has also come down when compared with baseline design.

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ACKNOWLEDGEMENTS

The authors wish to express their sincere gratitude to Dr O.R. Nandagopan, Outstanding Scientist and Director, NSTL Visakhapatnam for permission to publish this paper. The authors also wish to acknowledge the support of mr Ch. Sankara Rao, Scientist 'G' and mr D. Apparao, Scientist 'F' by providing the facilities at Environmental Test Centre, NSTL, Visakhapatnam.

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In the present paper, he has contributed for the static, modal and harmonic analysis and experimental work carried out on existing marine engine foundation

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In the present paper, he has contributed to the formulation of problem, guidance in applying boundary conditions in numerical analysis, suggestions for harmonic analysis and technical guidance in writing paper.