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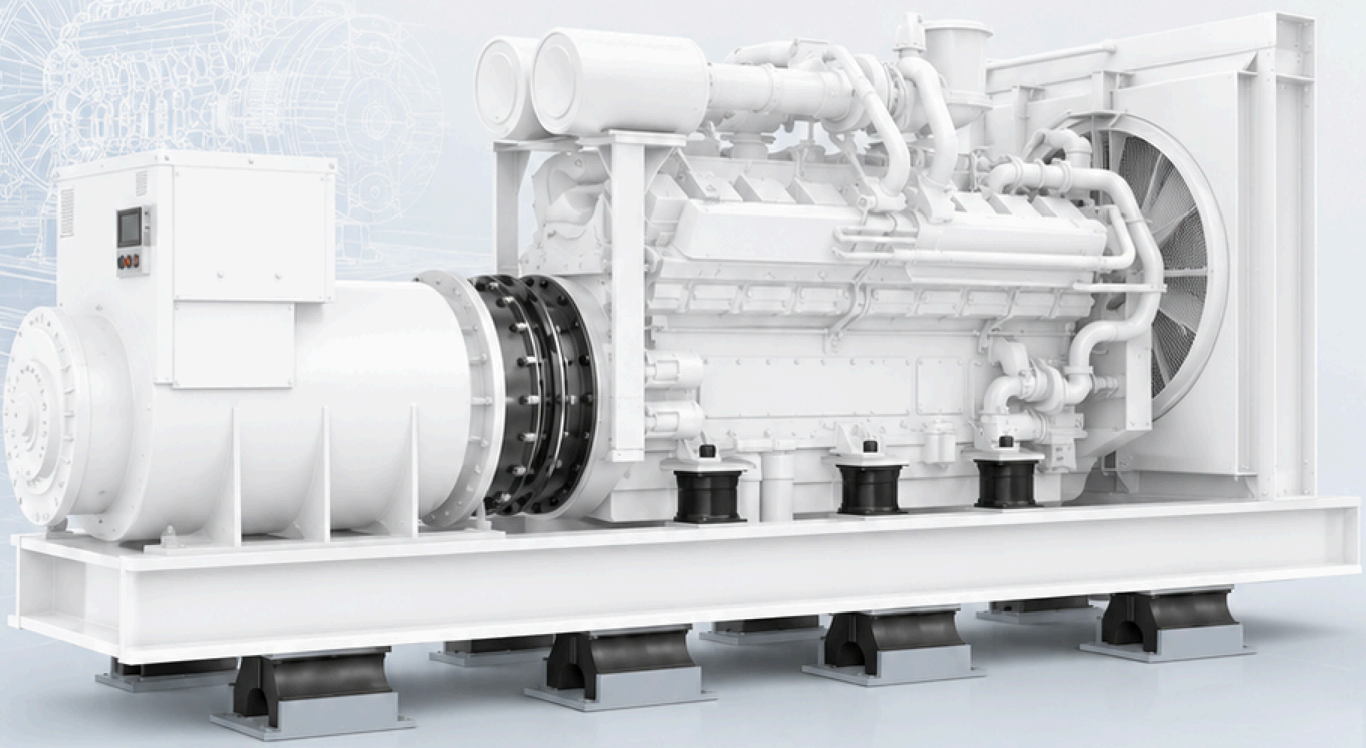
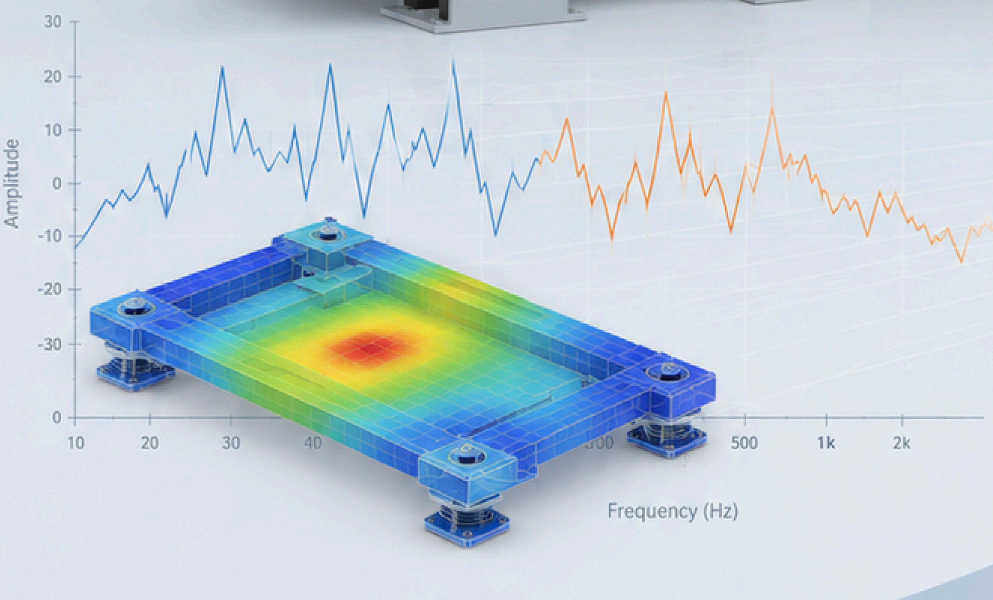
WHITE PAPER

VIBRATION ANALYSIS for a DOUBLE MOUNTED DIESEL GENERATOR





1 Introduction

"If you want to find the secrets of the universe,
think in terms of energy, frequency, and vibration."
Nikola Tesla

Vibration is one of the most fundamental phenomena observed in nature. From the oscillation of a pendulum and the propagation of seismic waves through the Earth to the vibration of a stretched violin string, oscillatory motion governs countless physical processes. Long before the advent of modern engineering, civilizations unknowingly exploited the principles of vibration in the creation of musical instruments. The pleasant tones produced by stringed instruments, drums, bells, and wind instruments are direct consequences of controlled vibrations. Ancient Greek philosophers, particularly Pythagoras (6th century BCE), demonstrated that the pitch of a musical note depends on the length and tension of a vibrating string, establishing one of the earliest scientific relationships between vibration and frequency. These observations laid the foundation for the scientific study of acoustics and wave motion.

The Renaissance and the Scientific Revolution marked a significant advancement in the understanding of vibration. Galileo Galilei observed the oscillatory behavior of pendulums, while Isaac Newton's laws of motion provided the mathematical framework necessary to describe dynamic systems. During the eighteenth and nineteenth centuries, pioneering scientists such as Daniel Bernoulli, Leonhard Euler, Joseph-Louis Lagrange, and Jean le Rond d'Alembert formulated analytical methods for studying vibrating strings, beams, and elastic bodies. Their work established the concepts of natural frequencies, mode shapes, and resonance—principles that remain central to vibration analysis today [Rao, 2017].

Initially, vibration research was primarily associated with acoustics and musical applications. However, the Industrial Revolution transformed the importance of vibration from an academic curiosity into a major engineering concern. The introduction of steam engines, rotating machinery, railways, and later internal combustion engines created mechanical systems operating at increasingly higher speeds and power levels. Engineers soon realized that excessive vibration could lead not only to uncomfortable noise but also to fatigue failure, structural damage, loss of precision, and catastrophic machine breakdown. Consequently, vibration analysis evolved into an essential discipline in mechanical engineering, influencing machine design, structural integrity, reliability, and maintenance practices [Hartog, 1985].

One of the most significant discoveries in vibration engineering was the phenomenon of resonance. When the frequency of an external excitation approaches a system's natural frequency, even relatively small forces can generate disproportionately large oscillations. Historical engineering failures—including bridge collapses, turbine failures, excessive rotor vibrations, and aircraft structural instabilities—demonstrated the potentially severe consequences of neglecting dynamic behavior. These events highlighted the necessity of accurately predicting natural frequencies and dynamic responses during the design stage rather than relying solely on static strength calculations [Thomson and Dahleh, 1998].

The twentieth century witnessed remarkable progress in vibration theory, computational mechanics, and experimental techniques. Classical analytical solutions were complemented by matrix methods, modal analysis, finite element methods (FEM), and digital signal processing. With increasing computational capabilities, engineers gained the ability to model highly complex structures, predict their dynamic characteristics, and evaluate multiple design alternatives before physical prototypes were manufactured. Modern vibration engineering therefore integrates theoretical analysis, numerical simulation, and experimental validation to ensure structural safety, durability, and acceptable noise performance.

In today's engineering landscape, vibration analysis plays a critical role across numerous industries, including automotive, aerospace, marine, rail transportation, power generation, civil infrastructure, consumer electronics, and renewable energy systems. In heavy machinery such as diesel generator sets, reciprocating engine forces, rotating unbalance, combustion-induced excitations, and electromagnetic forces generate dynamic loads that are transmitted through engine mounts to the supporting structure. If these vibrations coincide with the natural frequencies of structural components or mounting systems, resonance may amplify structural responses, resulting in elevated stress levels, increased Noise levels (SBN), accelerated fatigue damage, reduced service life, and deterioration of operational performance.

To mitigate these challenges, modern engineering relies extensively on Finite Element Analysis (FEA) for vibration assessment. FEA enables engineers to determine natural frequencies, mode shapes, harmonic responses, stress distributions, and dynamic load paths under realistic operating conditions. By identifying critical resonances and vibration transmission mechanisms during the design phase, engineers can optimize structural stiffness, modify mass distribution, redesign mounting systems, and improve damping characteristics without the need for multiple physical prototypes. Such predictive capabilities significantly reduce development time, cost, and the risk of field failures while improving overall product reliability.

2 Intro to 2-DOF system

The present white paper builds upon these fundamental principles to investigate the vibration characteristics of the studied mechanical assembly using finite element analysis. A general arrangement of a 2-DOF system is shown in Figure 1, this lays the foundation for multiple mounts in series. A good understanding of such a system is given by [Cheng et al., 2022]

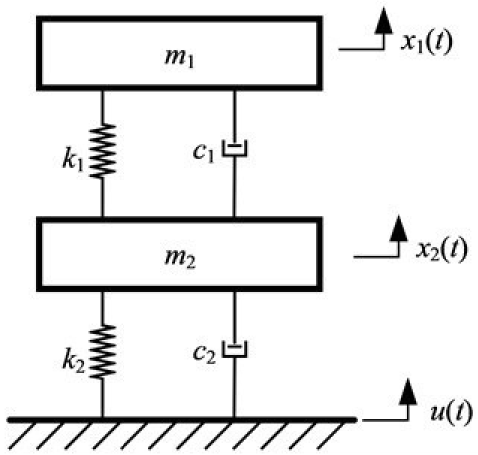


Figure 1: Double Layer Vibration isolation

Conventional mounting systems often fail to adequately isolate low-frequency engine vibrations, in such cases a two stage isolation really attentuates the isloation especially in lower frequencies [Hutchinson Aerospace, 2020].

In further sections we will see the General arrangement and the sequence of design and analysis activities to arrive at an acceptable design for the Baseframe of a diesel engine. Diesel engines are noisy both in terms of air audible noise and vibration noise. Hence a diesel engine structural vibration reduction has been chosen for deliberations.

2.1 General arrangement of DG assembly

A double mounted DG consists of two isolation stages:

• Primary engine mounts

• Secondary Baseframe Mounts

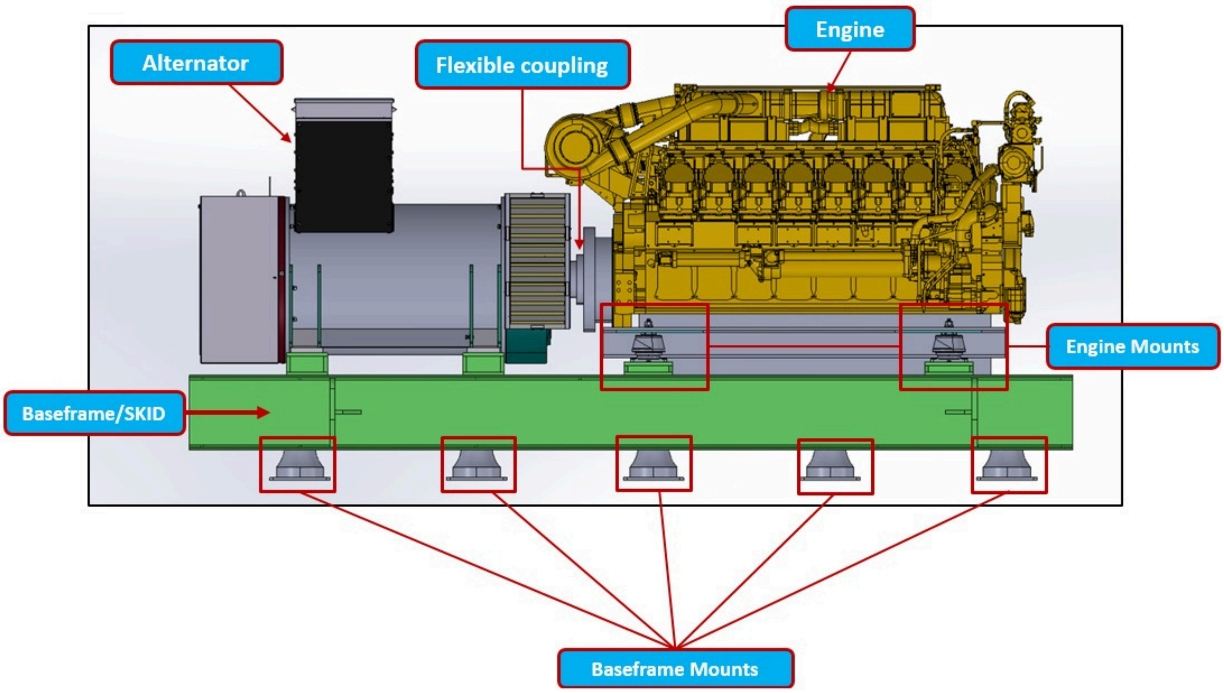


Figure 2: Genenral arrangement for DG assembly

As shown in Figure 2, the V-16 diesel Engine is coupled with flexible coupling having very high dynamic torsional stiffness to the Alternator. The engine is mounted on 4 shock mounts on to the Baseframe and the alternator is rigidly mounted on the Baseframe. For the foundation level isolation 10 in number, high deflection mounts are used. The calculation for mount selection and their specifications will be discussed in further section.

3 Problem Statement

The harmonic vibration of diesel engine can have serious implications when it interfers with the natural frequencies of the system, also the transmitted velocities and accelerations cannot excced certain limits of Structural Borne Noise (SBN), to improve habitability and such similar requirements. In ships it increases underwater noise which affects marine life and interferes with sensors.

The limiting curve for the SBN at top of engine mount is given in below Figure 3 to acheive silent operation.

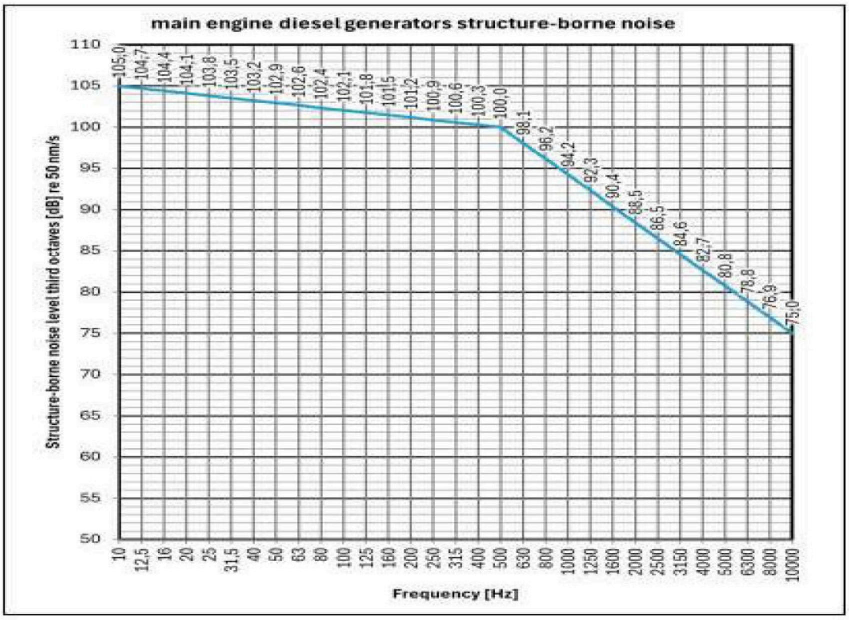


Figure 3: Genenral arrangement for DG assembly

The primary objective of the vibration analysis was to assess the effectiveness of the vibration isolation system over the entire operating frequency spectrum in terms of SBN performance.The first step is always to look for natural frequencies and checking for resonance; this is addressed in more detail at section 5 of this document.

The Baseframe design is the first step, where structural rigidity is the essence while high rigidity contributes faithful transmission of SBN and low rigidity may induce secondary vibrations, thus impedance of the Baseframe plays a very important role.

4 Baseframe Design

The preliminary Baseframe design steps are listed below

• **Section modulus selection:** Based on the weights of the engine, Alternator and the coupling, with a Factor of safety of 1.5 times, we can arrive at the gravity load that the Baseframe has to bear. This load can be distributed to two frames as a simply supported beam. Figure 4 below shows the scheme of this calculation.

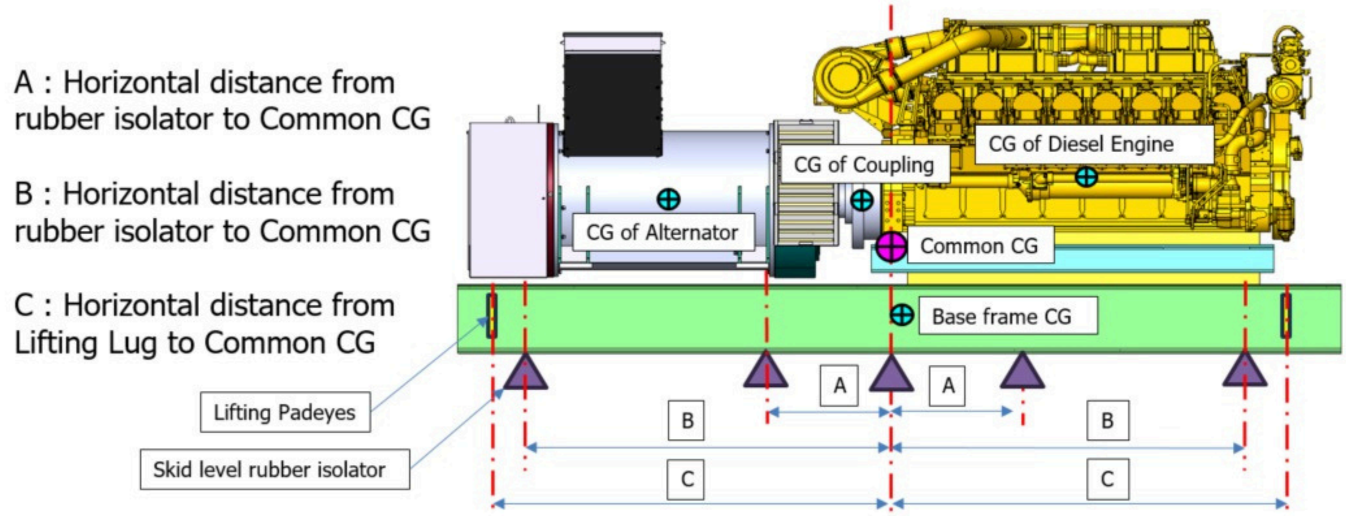
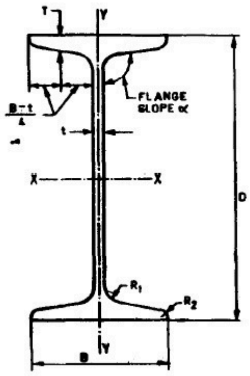


Figure 4: Isolators positioning

From here we calculate section modulus with the yield strength of the material, and once section modulus is arrived, we can look for the size of beam available from [IS 808, 1989] standard given in Figure 5



Designation	Mass M	Sectional Area, a	Dimensions							Sectional Properties					
			D	B	t	T	Flange Slope, *Max* α, deg	R_1	R_2	I_x	I_y	r_x	r_y	Z_x	Z_y
	kg/m	cm²	mm	mm	mm	mm		mm	mm	cm⁴	cm⁴	cm	cm	cm³	cm³
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
MB 100	8·9	11·4	100	50	4·7	7·0	98·0	9·0	4·5	183	12·9	4·00	1·05	36·6	5·16
MB 125	13·3	17·0	125	70	5·0	8·0	98·0	9·0	4·5	445	38·5	5·16	1·51	71·2	11·0
MB 150	15·0	19·1	150	75	5·0	8·0	98·0	9·0	4·5	718	46·8	6·13	1·57	95·7	12·5
MB 175	19·6	25·0	175	85	5·8	9·0	98·0	10·0	5·0	1 260	76·7	7·13	1·76	144	18·0
MB 200	24·2	30·8	200	100	5·7	10·0	98·0	11·0	5·5	2 120	137	8·29	2·11	212	27·4
MB 225	31·1	39·7	225	110	6·5	11·8	98·0	12·0	6·0	3 440	218	9·31	2·34	306	39·7
MB 250	37·3	47·5	250	125	6·9	12·5	98·0	13·0	6·5	5 130	335	10·4	2·65	410	53·5
MB 300	46·0	58·6	300	140	7·7	13·1	98·0	14·0	7·0	8 990	486	12·4	2·86	599	69·5
MB 350	52·4	66·7	350	140	8·1	14·2	98·0	14·0	7·0	13 600	538	14·3	2·84	779	76·8
MB 400	61·5	78·4	400	140	8·9	16·0	98·0	14·0	7·0	20 500	622	16·2	2·82	1 020	88·9
MB 450	72·4	92·2	450	150	9·4	17·4	98·0	15·0	7·5	30 400	834	18·2	3·01	1 350	111
MB 500	86·9	111	500	180	10·2	17·2	98·0	17·0	8·5	45 200	1 370	20·2	3·52	1 810	152
MB 550	104	132	550	190	11·2	19·3	98·0	18·0	9·0	64 900	1 830	22·2	3·73	2 360	193
MB 600	123	156	600	210	12·0	20·3	98·0	20·0	10·0	91 800	2 650	24·2	4·12	3 060	252

Figure 5: Standard Hot rolled sections

• **Mount stiffness selection** The calculation for proper mount selection is very important, initial calculation starts from the weight and the static deflection that the mount will experience and the maximum permissible limits in mount deflection. For example if the engine is weighing 10 tonnes then we can have 4 mounts each having 4000 N/mm as stiffness, so that we would have around 6-7 mm deflection against a maximum permissible static deflection of 10mm.

In other words static deflection should be about 60-70% of full deflection to accommo- date dynamic conditions and age related deterioration. This needs to be verified in the static analysis as shown in Figure 6.

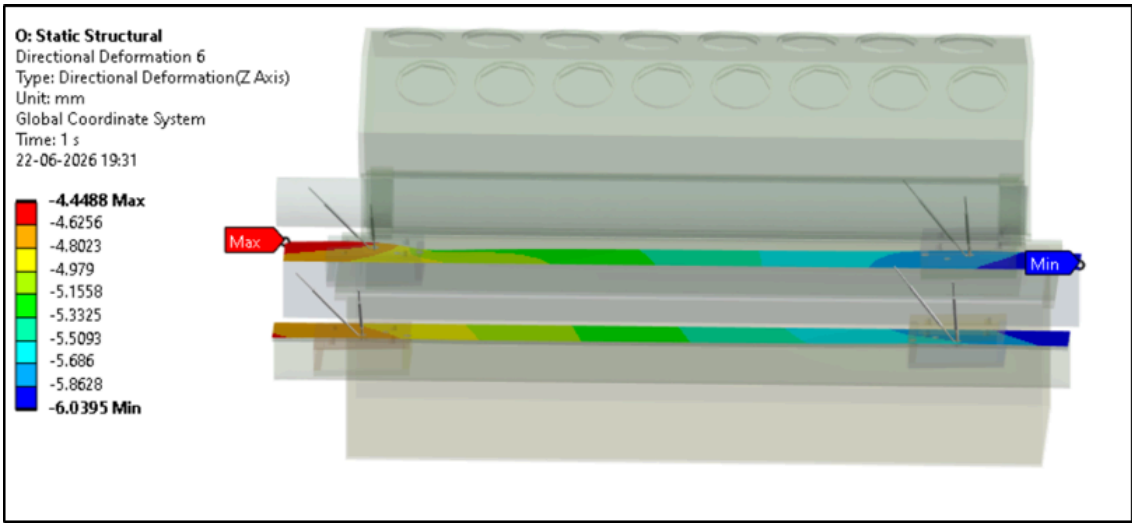


Figure 6: Static Deflections

Once the selection of Beams and mounts is done, we can go ahead with the modal analysis which will give us the preliminary check for the natural frequencies and resonance.

5 Modal Analysis

The governing equation in case of modal analysis:

$$M\ddot{x} + C\dot{x} + Kx = F(t)$$

where

M = Mass Matrix

C = Damping Matrix

K = Stiffness Matrix

Assumptions:

- As it is free undamped vibration, hence the damping and force both are zero.

The natural frequencies are obtained from

$$\det(K - \omega^2 M) = 0$$

The modal analysis results with mode shapes are given below, as we expect that the assembly will have 6 rotations and 6 translations movements. Understanding the resonance frequencies with the modes is very important. A sample output is represented at Figure 7, which demonstrates the efficacy of the mount and the Baseframe design.

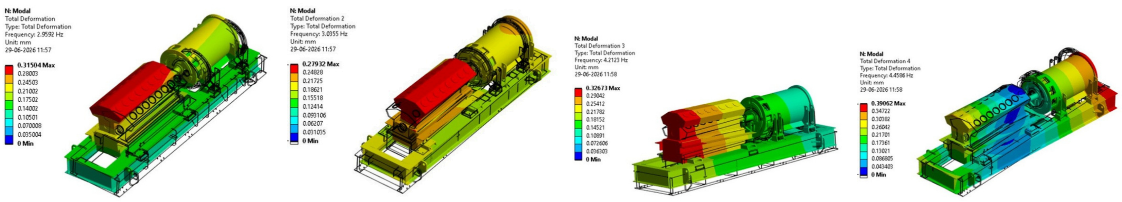


Figure 7: Mode shapes 1-4

Table 1: Natural frequencies obtained from modal analysis

Mode	Frequency (Hz)
1	2.9592
2	3.0355
3	4.2123
4	4.4586
5	5.6492
6	7.4951
7	10.318
8	13.114
9	13.552
10	16.867
11	20.325
12	21.362
13	35.555
14	40.812

The natural frequencies of the assembly should not interfere with the harmonics (0.5x25Hz, 1x25Hz, ...) of the engine rotational speed which could result in resonance condition, also the mode shapes has to be observed carefully, for pitching, rolling, yawing, other translation modes and any mounted body for its independent oscillations.

6 Harmonic analysis

The governing equation for harmonic analysis is given below:

$$m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = F(t)$$

$$m_2\ddot{x}_2 + c_1(\dot{x}_2 - \dot{x}_1) + c_2\dot{x}_2 + k_1(x_2 - x_1) + k_2x_2 = 0$$

For performing harmonic analysis first calculate the loads coming from engine gas pressures on the shaft, and then converted them to moments. Any engine OEM usually provide tangential pressure in terms of Fourier series for different crank angle and harmonic order, where the order is varying from 0.5 to 12 and the crank angle is varying based on the firing order of the engine from 0 to 2π, Given in Eq 1. Once moments are derived from these tangential pressures, then application of the same with phase shift and position on shaft was applied based on the engine firing sequence.

$$\text{Tangential Pressure} = (R + jI) \times e^{j\cdot\text{Order:CA}} + \text{Complex Conjugate} \tag{1}$$

The analysis is carried out with the FULL method and with a sweep frequency of 0-500 Hz with a spacing of 1 Hz frequencies. The Full method equation is given below in Eq.2

$$\underbrace{\left(-\Omega^2[M] + i\Omega[C] + [K]\right)}_{[K_c]} \underbrace{(\{u_1\} + i\{u_2\})}_{\{u_c\}} = \underbrace{(\{F_1\} + i\{F_2\})}_{\{F_c\}} \tag{2}$$

$$[K_c]\{u_c\} = \{F_c\} \tag{3}$$

7 Structure Borne Noise (SBN)

Evaluation of the SBN is critical for understanding the quantum of the vibration en- ergy that passed onto the the foundation, we used the formulation which are mentioned below:

1. **Energetic average method:** In this method we take measurements of velocities squares at each mounting locations and then take a mean, and compare it with Vo (50 nm/sec) in log scale. The governing equation is given in Eq.4

$$L_v = 10\log\left(\frac{1}{n}\sum_{i=1}^{n}\frac{V_i^2}{V_0^2}\right) \tag{4}$$

8 Harmonic Results

The SBN levels are taken for four positions and the results for the same are shown in Figure 8,

1. At top of engine mount
2. At bottom of engine mount
3. At top of Baseframe mounts
4. At bottom of Baseframe mounts

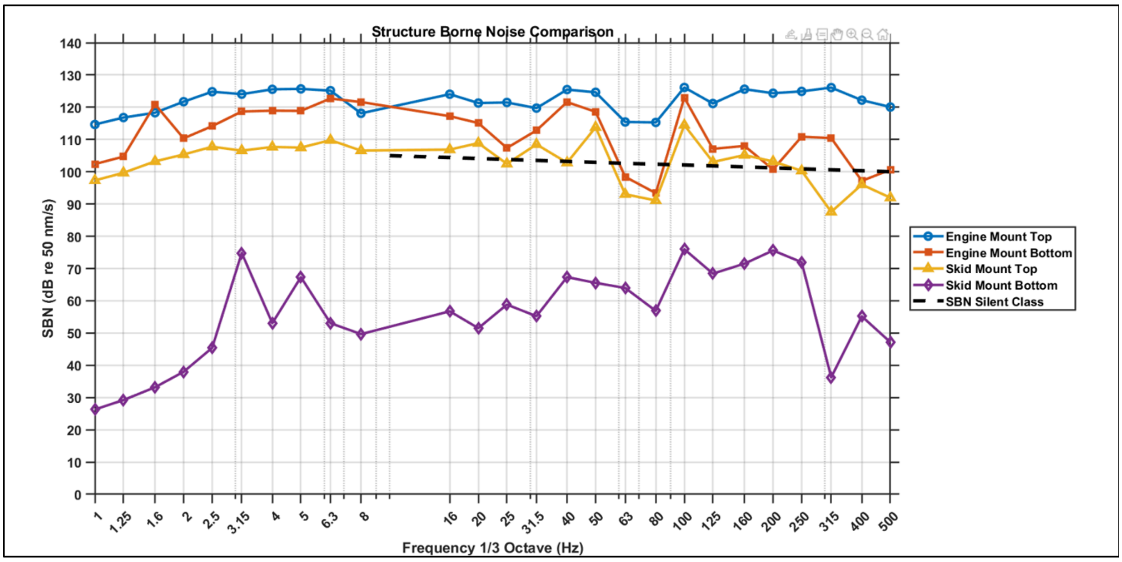


Figure 8: SBN level comparison

9 Conclusion

The inferences that can be made based on the FEA analysis of the DG assembly

• The natural frequencies assessed is in table 1. The forcing frequency based on engine speed including its harmonics (0.5x25 Hz, 1x25 Hz....), are away from natural frequencies and hence chances of resonance during operation is ruled out.

• The static deflection at the engine level AV mount (1st set of isolating mount) is 6 mm and the deflection at the second set of AV mount located at base frame level is 13.6 mm, both sets of AV mounts meet the static deflection criteria specified by the mount supplier and falls in the bracket of less than 80% of full permitted deflection.

• SBN levels Given in Figure 8 shows that the bottom levels of SBN are around 75 dB re 50nm/s, which is considered to be within limits for the intended marine application.

• Softwares like Solidworks and ANSYS Workbench were used for estimation. The vibration levels when converted as power can be used for ABN estimation.

• Additionally the structure in static and harmonic conditions was assessed for stress and deformations and found it be within acceptable limits.

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