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Resonance Mitigation through Frequency-Constrained Topology Optimization: An Alternative Structural Design Strategy

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Abstract: A sub frame is an intermediate structure between vehicle chassis and various functional subsystems. Unlike conventional approach of designing a structure free from external vibration excitations, topology optimisation using Solid Isotropic Material with Penalisation (SIMP) model is used to redistribute the material on the sub frame through constraining mass and eigen frequencies within desired limit to achieve resonance free structure. A comparative study is carried out to evaluate the distribution of material on the sub frame of road transporter vehicle (RTV) to develop a high stiff sub frame with minimum weight considering dynamic properties. The sub frame is subjected to steady state forced vibrations and subsequently its functional performance is studied and improvised through frequency-constrained topology optimization (Dynamic Topology Optimisation). Manufacturing aspects of sub frame is considered while performing topology optimisation in Finite Element Analysis tool.
Keywords: Road transporter vehicle, Sub frame, Dynamic topology optimisation, Eigen frequency, SIMP.

I. INTRODUCTION

A comparative study is conducted to investigate the effect of material redistribution in avoiding resonance induced by onboard rotating machinery, using topology optimisation tool. this study is carried out on the sub frame of solo truck chassis (HMV 12X12) mounted with a hydraulic and electrical power packs along with superstructure which holds a payload of 13 ton. The solo truck (HMV 12X12) along with subsystems (power packs and superstructure) and sub frame is called Road transporter vehicle (RTV). The RTV is used to transport the payload and to position it vertically during operation. The focus is on the sub frame of a RTV, which is optimised to achieve an improved strength-to-weight ratio while minimising the risk of vibration-induced failure under steady-state forced vibrations using SIMP with dynamic constraint. The subframe is an intermediate structure that couples superstructure, hydraulic and electrical power packs and electronic components to the vehicle chassis, serving as a platform for mounting these subsystems. this study investigates the severity of prolonged vibrations generated by rotating machineries mounted on the sub frame and explores their mitigation by an alternative design approach using topology optimisation, enabling a more reliable and vibration-resistant sub frame.

Shape and topology optimisation techniques have found extensive applications in modern product design and structural engineering. The theoretical foundation of topology optimisation was first introduced by Bendsøe and Sigmund [1], [2] who's work established the fundamental principles that underpin optimisation methods widely adopted in commercial finite element analysis (FEA) tools. Patel and Rokade [3] applied topology optimisation to the design of a tilt beam, developing a feasible and manufacturable configuration. Their study employed the SIMP (Solid Isotropic Material with Penalisation) approach to redistribute material, thereby achieving an optimal balance between structural strength and weight reduction. The SIMP method, originally developed by Bendsøe [1], employs a fictitious material model where element densities serve as design variables. These densities are penalised through a power-law interpolation (penalty factor) to approximate discrete material distribution. The influence of the penalty factor on convergence was extensively studied by Rietz [4], who highlighted its critical role in obtaining physically meaningful topologies. VolkanKandemir et al. [5] further evaluated the performance of the SIMP method, particularly for cases where the penalty factor was set to one, demonstrating its implications on structural response.

The dynamic characteristics of a structure, such as Eigen frequencies and mode shapes, are directly influenced by material distribution. As the topology evolves, these characteristics change accordingly. In dynamic topology optimisation (DTO), frequency-related constraints are introduced to shift natural frequencies and mode shapes to desired values or ensure they remain within prescribed ranges.

Hongling Ye et al. [6] investigated the topology optimisation of three-dimensional continuum structures under frequency constraints, employing a DSQP model to solve the dynamic optimisation problem. Maeda et al. [7] proposed a method to maximise selected Eigen frequencies and corresponding mode shapes of vibrating structures using DTO with the SIMP approach, demonstrating its effectiveness in enhancing dynamic performance. Apart from SIMP-based approaches, another widely used method is Evolutionary Structural Optimisation (ESO). First introduced by Xie and Steven [8], ESO is based on the principle that an optimal structure characterised by maximum stiffness and minimum weigh can be achieved by progressively removing inefficient material from the design domain. Resonance may occur, potentially causing severe structural damage. Therefore, incorporating frequency constraints in topology optimisation is essential for ensuring vibration-resistant designs. Prolonged exposure to resonance frequencies often results in fatigue failure due to cyclic stresses. Jong Wook Lee et al. [9] proposed a novel topology optimisation method for assessing dynamic fatigue failure in the frequency domain under random excitation forces, highlighting the importance of accounting for resonance-induced stresses. While external excitations may be harmonic or transient in nature, their effects during resonance are particularly catastrophic. Shubham Saurabh et al. [10] presented a robust topology optimisation (RTO) framework for structures under transient loading. Using a density-based SIMP approach, their study minimised dynamic response while satisfying volume constraints, thereby improving structural resilience. Xuqi Zhao et al. [11] proposed an optimization model that takes the structural static response of the structure as a weighted part of the objective function. It made the optimized configuration more applicable for engineering design and introduced an efficient method for calculating frequency responses over a frequency interval.

Although significant research has been conducted on topology optimisation, very few studies have specifically addressed steady-state forced vibration as the primary source of excitation which to be removed from structure through SIMP model. Furthermore, shifting of resonance through topology optimisation with frequency as a constraint using FE analysis for designing a mechanical structure remains largely unexplored. this paper addresses this gap by investigating topology optimisation with mass and frequency constraints, aiming to shift resonance frequencies into a desired bandwidth using the SIMP model ensuring improved vibration resistance for a particular frequency while considering practical design and manufacturing aspects.

II. MATHEMATICAL FORMULATION

A. *Topology optimisation*

In structural optimisation, the general problem is to determine the optimal material distribution within a prescribed design domain such that a given performance parameters are optimised while satisfying physical and resource constraints. Mathematically, a topology optimisation problem can be expressed as:

$$\min_{x} \; f(x)$$

Subject to

$$g_i(x) \leq 0, \qquad i = 1,2,3,\ldots\ldots m$$

$$h_j(x) = 0, \qquad j = 1,2,3,\ldots\ldots p$$

$$x_{min} \leq x_k \leq x_{max}, \qquad k = 1,2,3,\ldots\ldots n$$

Where f(x) is objective function (e.g., compliance minimisation), x [x1, x2,…, xn] is vector of design variables, n is number of design variables, $g_i(x)$ and $h_j(x)$ are functions subjected to inequality and equality constraints respectively. This study employs the Solid Isotropic Material with Penalisation (SIMP) method. The design variable is the element-wise material density, ρ_e which takes values between 0 (void) and 1 (solid). The primary objective is to minimise compliance (equivalently, maximise stiffness) subject to mass and eigen frequency constraints. The finite elemental topology optimisation problem is written as:

Where C(ρ) is compliance of the structure, F is global load vector, u is displacement vector, K(ρ) is element stiffness of structure, K_e^0 is stiffness for solid material, ρ_e is elemental density, V_e is volume of element e, V* is prescribed maximum material volume, N is total number of finite elements, p is stiffness penalisation factor (typically p=3) $\rho_{min} \ll 1$ is small lower bound to prevent singularity. The penalisation factor p drives the material interpolation towards discrete 0/1 solutions, ensuring a physically meaningful topology. Constraint (1.1) ensures static linear analysis whereas (1.2) shows stiffness as a function of density to create void/solid elements in the system. Constraint (1.3) is the mass conservation at finite element level.

$$\min_{\rho} C(\rho) = F^T u \tag{1}$$

Subject to

$$K(\rho)\, u = F \tag{1.1}$$

$$K(\rho) = \sum_{e=1}^{N} \rho_e^p K_e^0 \tag{1.2}$$

$$\sum_{e=1}^{N} \rho_e V_e \le V^* \tag{1.3}$$

$$0 < \rho_{min} \le \rho_e \le 1 \tag{1.4}$$

B. Eigen frequency constraint

To ensure smooth dynamic performance and stability, eigen frequency constraints are incorporated in SIMP model. The governing eigenvalue problem is

$$K(\rho)\varphi = \lambda M(\rho)\varphi, \quad \lambda = \omega^2 \tag{2}$$

$$M(\rho) = \sum_{e=1}^{N} \rho_e^q M_e^0 \tag{3}$$

Where φ is eigen mode vector, λ is eigenvalue, corresponding to squared natural frequency, ω is natural angular frequency, $M(\rho)$ is element mass of structure, M_e^0 is mass for solid material, q is mass penalisation factor usually taken as 1. Eq. (2) serves as a basic for finding undamped natural frequency of system. Equation 3 shows mass as a function of density to create void/solid elements in the system. By constraining selected eigen frequencies within prescribed limits, undesirable resonances are avoided. In this study eigen frequency constraint serve as an additional constraint to minimum compliance objective (1). SIMP Model with objective function (1) is employed with respective constraints and Eq. (2) & (3) is added as an additional constraint to study the distribution of material on the Sub frame to have desired resonant frequencies removed.

C. Modal and Frequency Response Analysis

Excessive vibrations can cause fatigue failure of structural components. In applications such as sub frames of road transporter vehicle, vibrations may originate from both environmental loads (road excitations) and onboard rotating machinery (e.g. power packs and hydraulic Pumps). The vibration from onboard rotating machineries is predominantly sinusoidal and induce steady-state forced vibrations in the structure. The general equation of motion governing dynamic response is given as

$$M\ddot{u} + C\dot{u} + Ku = F(t) \tag{4}$$

Where M is mass, C is damping coefficient, K is stiffness of system and F(t) is time-dependent external force. $u, \dot{u}, \ddot{u}$ are displacement, velocity, and acceleration of system respectively. In a system when $F(t)$=0, it forms a free vibration analysis and the Eq. (4) becomes an Eigen value problem (2) whose Eigen value is proportional to square of undamped natural frequency and respective Eigen vectors are the mode shape of the system. Modal analysis is carried out to find undamped natural frequency and mode shape of the system. Using mode superposition method, these mode shapes and natural frequencies are evaluated for harmonic or frequency response analysis. For this, the Eq. (4) becomes

$$M\ddot{u}(t) + C\dot{u}(t) + Ku(t) = Fe^{i\omega t} \tag{5}$$

$C = \alpha M + \beta K$, Where α and β are two real scalars and independent of ω; F is the amplitude of the sinusoidal load and ω its excitation frequency. Assume that $u(t) = Ue^{i\omega t}$ is solution of Eq. (5). Substituting it into Eq. (5). yields

$$(K + i\omega C - \omega^2 M)U(\omega) = F \tag{6}$$

The velocity and acceleration amplitudes of dynamic responses are given as $\dot{U}(\omega) = i\omega U(\omega)$, $\ddot{U}(\omega) = -\omega^2 U(\omega)$ respectively. Substituting this, the final equation for response becomes

$$U(\omega) = \sum_{j=1}^{n} c_j(\omega)\varphi_j, c_j(\omega) = \frac{\varphi_j^T F}{\lambda_j - \omega^2 + i\alpha\omega + i\beta\omega\lambda_j} \tag{7}$$

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Where λ_j is the j^{th} Eigen value, $\lambda_j - \omega_j^2$, the square of the j^{th} natural frequency in rad/s; φ_j is the associated Eigen mode vector. This formulation [11] captures the frequency dependent response $U(\omega)$ under rotating machinery-induced vibrations on the system.

III. GEOMETRIC FORMULATION

For carrying out frequency constrained topology optimization, a RTV consisting of 12X12 high mobility vehicle (HMV) and superstructure with subsystems mounted on the sub frame is considered. this sub frame holds various subsystems required to transport and operate sensitive payload. the general layout of RTV is shown in figure 1. Initially the sub frame is assumed to be a rectangular plate of 13310 x 3010 x 150 mm and 47262 Kg mass. The dimension of sub frame and CG location of different subsystems conforms to defence industry guidelines and are marked on the sub frame as shown in figure 2. The width of chassis of 12X12 HMV is 90 mm and hence two patches of each 90 mm width is drawn on the sub frame to mark the fixed region between chassis and sub frame. Table 1 presents the approximate masses of all subsystems. Load Shell 2 is an electrical power pack incorporating a diesel generator operating at 1800 RPM (30 Hz). Load Shell 3 comprises a hydraulic power pack equipped with a motor-driven pump running at 1800 RPM to circulate hydraulic fluid to the relevant subsystems. Other subsystems are static in nature.

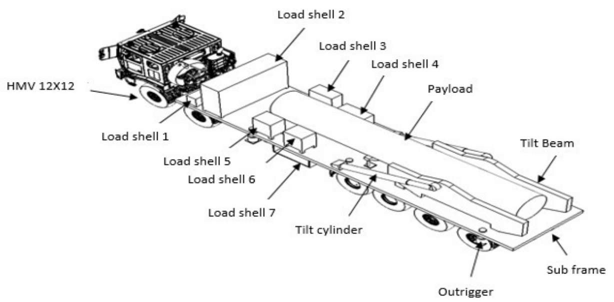

Fig. 1 General layout of road transporter vehicle (RTV)

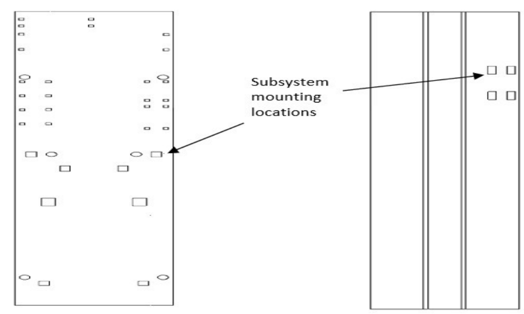

Fig. 2 Mounting locations of subsystems on the sub frame

The hydraulic pump powers six outriggers mounted on the sub-frame (two pairs at the front and rear, and one pair at the mid-section). Load Shells 1, 5, and 6 house electronic components that serve as interfaces between the payload and the RTV, while Load Shell 7 contains the user interface device and is positioned beneath the sub-frame for improved ergonomics. The payload is supported by a tilt beam, hinged at the rear and simply supported at the mid-section of the sub-frame. The tilt beam is actuated by a centrally mounted tilt cylinder. Together, the tilt beam and tilt cylinder constitute the superstructure. During operation, electrical power from Load Shell 2 and hydraulic power from Load Shell 3 actuate the outriggers, which extend to establish ground contact and transfer the RTV loads to the ground. Simultaneously, the tilt cylinder elongates to raise tilt beam along with payload from the horizontal position to the vertical position (0° to 90°). The main focus is on the hinged location of superstructure as vibration from on board rotating machineries during operation will transfer to sensitive payload via these hinges.

Table 1. Subsystems mounted on sub frame and their approximate masses

Subsystem	Mass (Kg)
Load shell 1	500
Load shell 2	1000
Load shell 3	800
Load shell 4	500
Load shell 5	500
Load shell 6	500
Load shell 7	300
Superstructure with payload	20000
Outriggers	each 250

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IV. FINITE ELEMENT FORMULATION

ANSYS workbench is used to carry out topology optimisation of sub frame. Initially the sub frame is modelled as a rectangular plate of structural steel of conventionally used grade S690QL with density 7850 Kg/m^3 and yield strength of 690 MPa. As this study only requires load distribution of various subsystem over the sub frame, subsystems are modelled as point masses at their

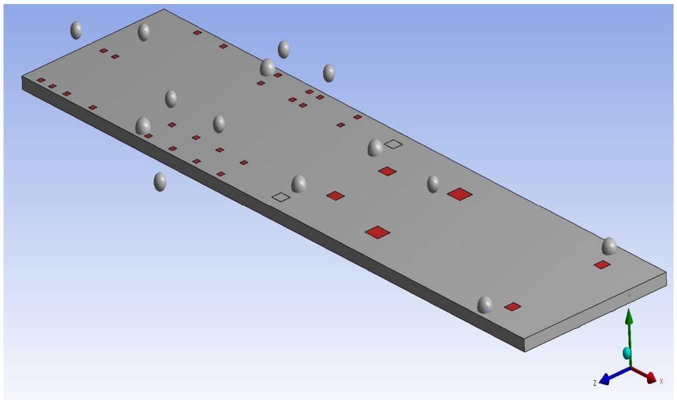


Fig. 3 Subsystems as a point mass at their respective mounting location on the sub frame

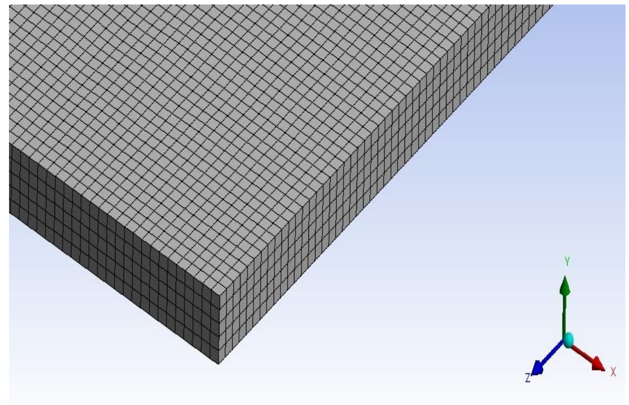


Fig. 4 Meshed model of sub frame

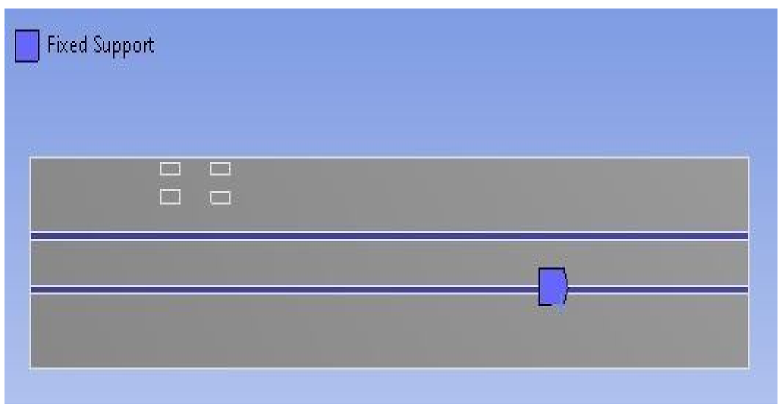


Fig. 5 Fixed location on the sub frame

Respective location in FEA tool. The sub frame is discretise using linear 3D Hexahedral (Hex 20) element with an optimum mesh size of 20 mm. a total of 270523 nodes with 233909 elements are formed. The sub frame is fixed at two patches making all 3 degrees of freedom of their respective nodes to zero. Since the primary objective of this study is to determine the optimal material distribution within the sub-frame, linear static stress analysis is considered sufficient, as static loading is the predominant operating condition. Dynamic loads encountered during RTV movement and other transient loading conditions are excluded from the scope of this study. A 1g load is applied to the sub frame in transverse (Y) direction to find linear static stresses. Stress measurement serves as a prerequisite to topology optimisation. The flow chart shown in figure 6 gives brief detail of process required in topology optimisation of sub frame. The objective is to minimise compliance subject to mass retention. The mass retention percentage is assumed based on practical industrial experience and can be any number between 0 to 100. For this study, 20% (min) to 23% (max) of original mass of sub frame is considered. The optimised design obtained under mention constraint is then converted into a feasible design keeping manufacturing aspects. Modal analysis of this new design of sub frame is carried out to find mode shapes and undamped natural frequencies. if any of the natural frequency of sub frame is closer to the frequency of harmonic excitations generated from load shell 2 and 3, resonance will occur. Excessive vibrational due to resonance propagates throughout the sub frame including the hinged support of tilt beam holding payload. Hence this natural frequency exciting hinged support needs to be displaced apart effectively to ensure the safety of payload.

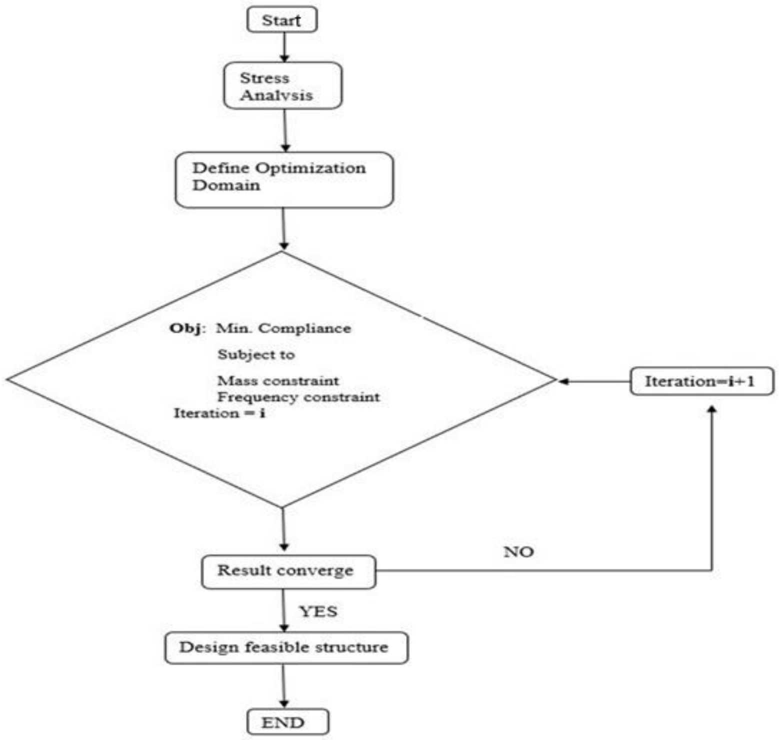


Fig. 6 Flow chart for Topology Optimisation

Based on the results of modal analysis, a dynamic topology optimisation i.e. topology optimisation with eigen frequency as an additional constraint is employed in FE analysis. this is done to redistribute the material on the sub frame such that critical mode shape and frequency falling within the range of operating condition of load shell 2 and 3 is removed. the final feasible design obtained after this approach is verified using harmonic analysis with same boundary condition and is taken forward for manufacturing.

V. RESULTS AND DISCUSSION

A. Topology Optimisation of Sub frame without frequency constraint

The Rectangular sub frame along with various subsystem is subjected to 1g vertical load. this is to verify the static load capacity of assumed rectangular sub frame mounted on the 12X12 chassis. the von misses stress measured is 5.52 MPa which is far below the yield strength of S690QL. it is evident from the stress distribution shown in figure 7 that a good amount of volume of sub frame is subjected to very less or negligible stress. this volume can be removed without compromising the overall strength of sub frame. the topology optimisation is carried out with objective of minimising compliance (reciprocal of Stiffness) by constraining mass from 20% min to 23 % max of overall mass of sub frame. the design region as well as excluded region (the location of subsystems and fixed region) is shown in figure 8. the material distribution given by topology optimisation tool is shown in figure 10. the result converges to objective function value after 23 number of iterations under given constraints. this design is not feasible from manufacturing aspects and hence a feasible design is developed based on the topology generated by optimisation tool to meet fabrication constraint as shown in figure 11.

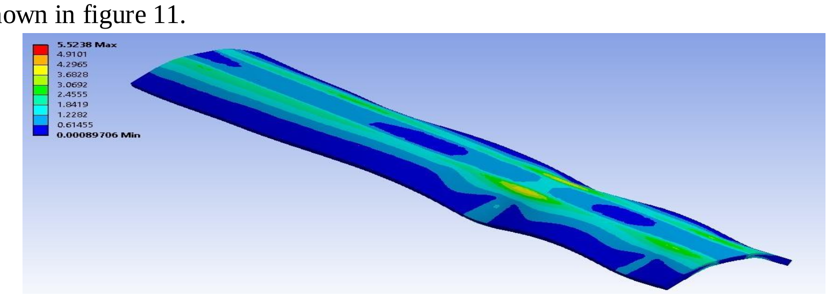


Fig. 7 Von misses stress level on the rectangular sub frame

The mass of newly obtained design is 5847 Kg. the longitudinal members are made of square box section with cross section of 200 x 200 x 20 mm and transverse members are made using tapered C channel with thickness of 20 mm following industries manufacturing standards.

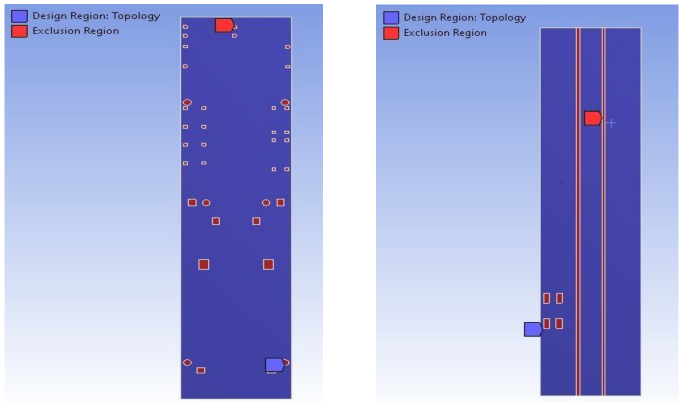


Fig. 8 Design region and excluded region for topology optimisation

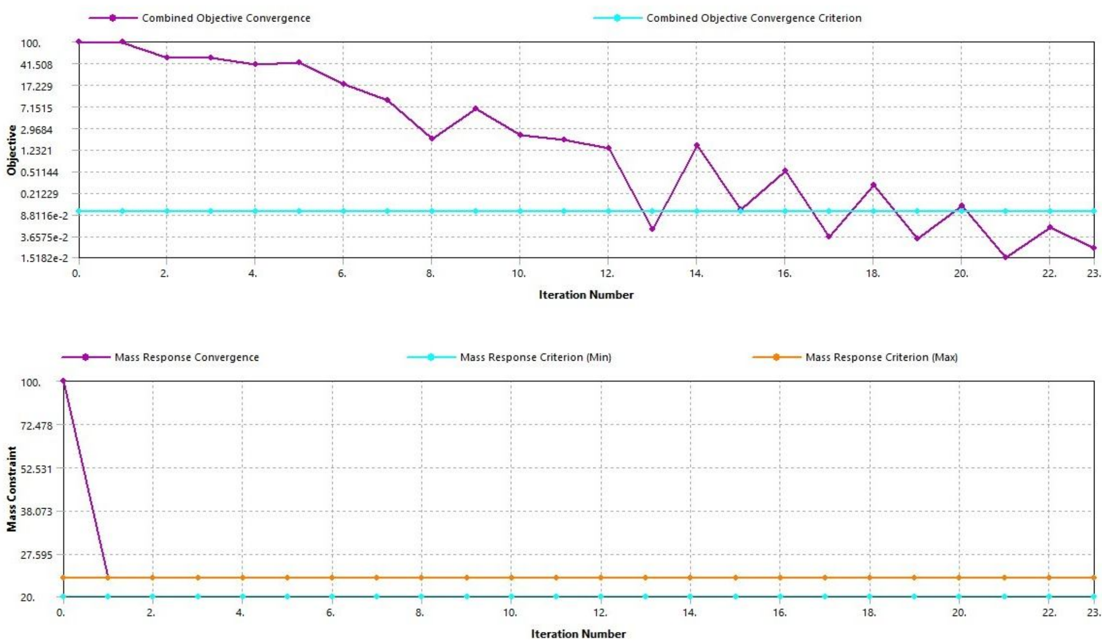


Fig. 9 Convergence graph for topology optimisation

The feasible design shown in figure 11 is validated by mounting various subsystems on it and subjected to 1g load in vertical direction. As shown in figure 14, the von misses stress on the feasible design is 102.12 MPa which is below yield strength of S690QL. Hence this design having mass 5847 Kg is considered for further analysis.

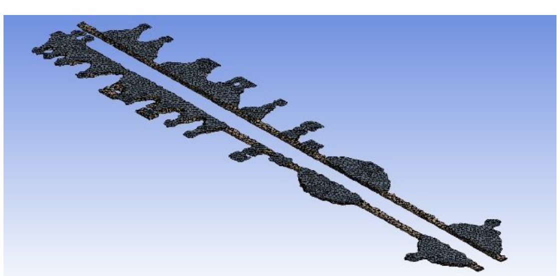


Fig. 10 Topology optimised sub frame

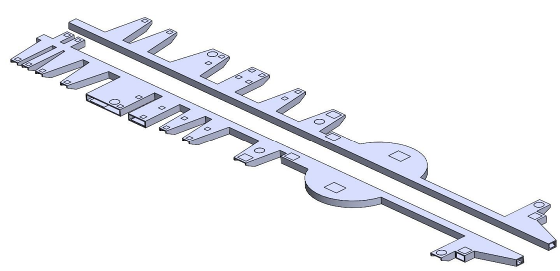


Fig. 11 Feasible design of sub frame

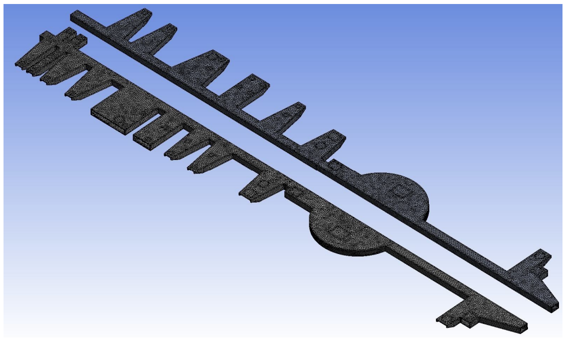


Fig. 12 Meshed model of sub frame

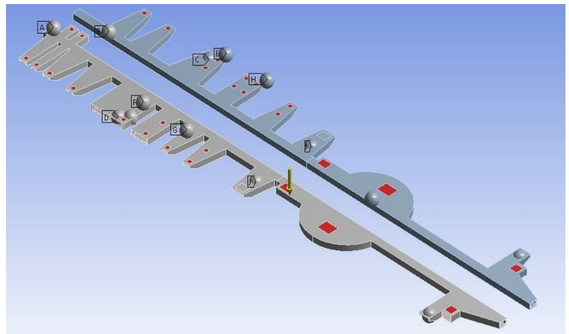


Fig. 13 Subsystems mounted on Feasible designed sub frame

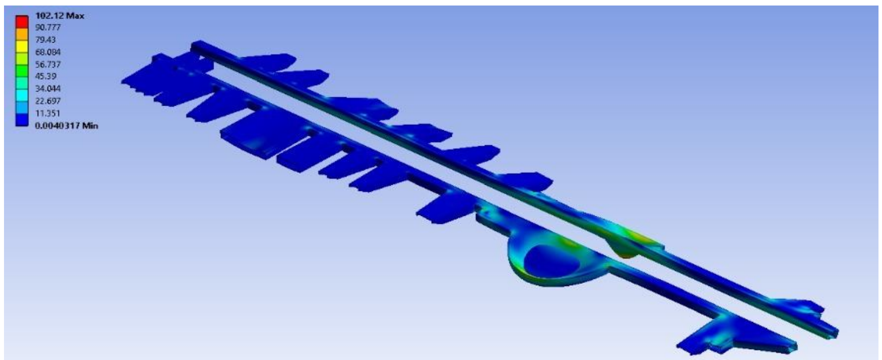


Fig. 14 Von misses stress level on feasible designed sub frame

B. Modal analysis

Modal analysis of feasible design with various subsystem mounted on it is carried out to find the mode shapes and undamped natural frequencies under same boundary condition. from figure 15a to 15k, it is seen that 9^{th} mode is vibrating hinged location of tilt beam with 30.57 Hz frequency. this bending mode is critical as machineries are rotating with 30 Hz which is close to the frequency of 9^{th} mode of vibration of sub frame and will excite tilt beam affecting payload functionality during operation.

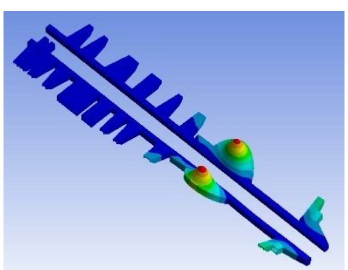


Fig. 15(a) 1^{st} mode with 12.55 Hz

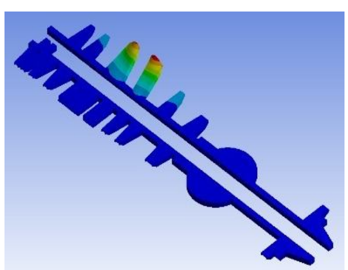


Fig. 15(b) 2^{nd} mode with 15.37 Hz

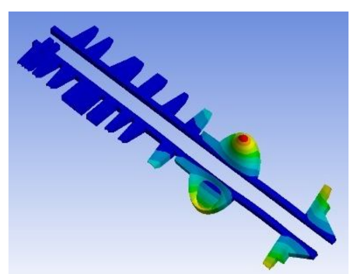


Fig. 15(c) 3^{rd} mode with 18.04 Hz

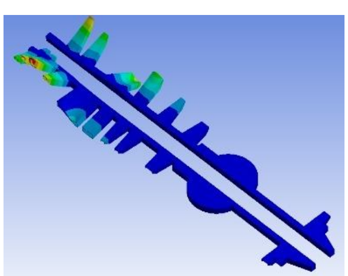


Fig. 15(d) 4^{th} mode with 18.46 Hz

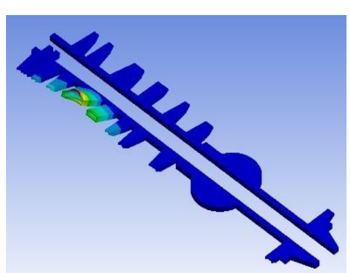


Fig. 15(e) 5^{th} mode with 22.34 Hz

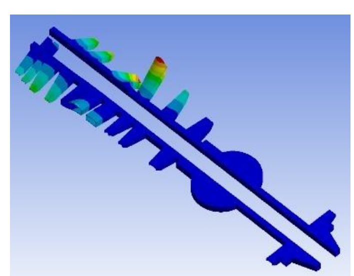


Fig. 15(f) 6^{th} mode with 23.50 Hz

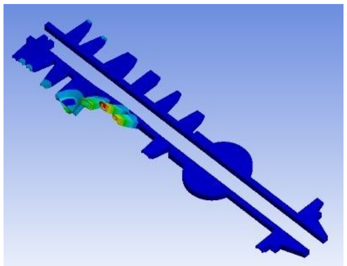


Fig. 15(g) 7th mode with 27.96 Hz

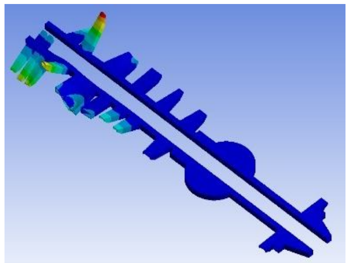


Fig. 15(h) 8th mode with 28.73 Hz

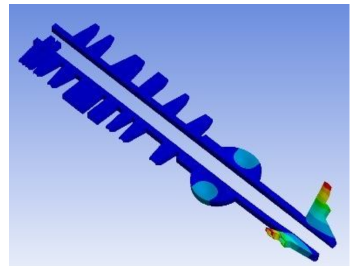


Fig. 15(i) 9th mode with 30.57 Hz

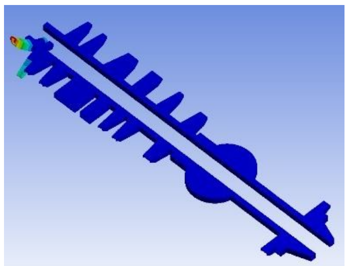


Fig. 15(j) 10th mode with 31.48 Hz

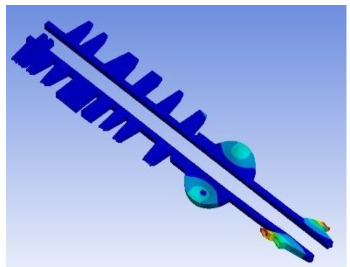


Fig. 15(k) 11th mode with 32.85 Hz

Fig. 15 (a-k) 1st to 11th mode of vibration with corresponding natural frequencies.

C. Topology optimisation of sub frame with frequency constraint

Modal analysis shows that the hinged area of tilt beam of the sub frame is executing bending mode with 30.57 Hz frequency and is close to excitation frequency of 30 Hz. hence the rear area of sub frame holding tilt beam is only considered for frequency constraint optimisation or dynamic topology optimisation. this approach is acceptable as reduction in mass from this area will result in change in desired mode shape. This is shown in figure 16. The objective is minimising compliance of selected region of sub frame Subjected to mass constraint of 50 % (min) to 99 % (max) of overall mass of sub frame. The mass percentage is kept high as already enough amount of volume is reduced and further reducing mass may result in weak structure.

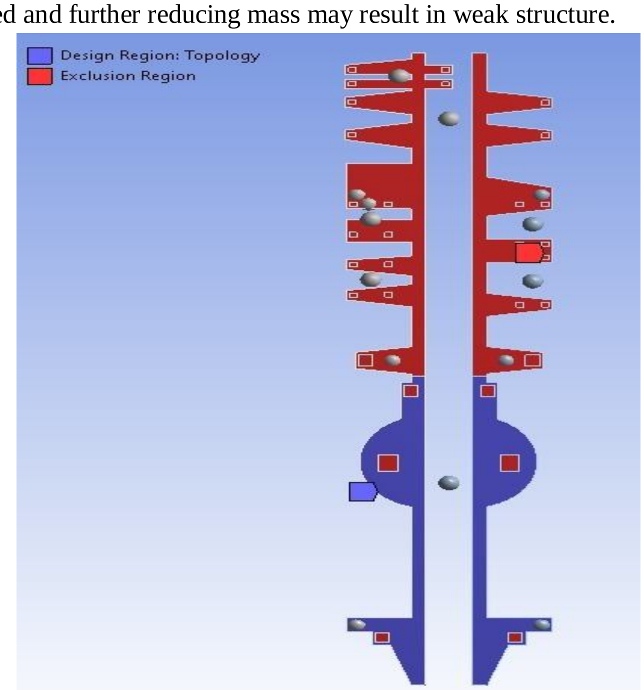


Fig. 16 Included and excluded Region

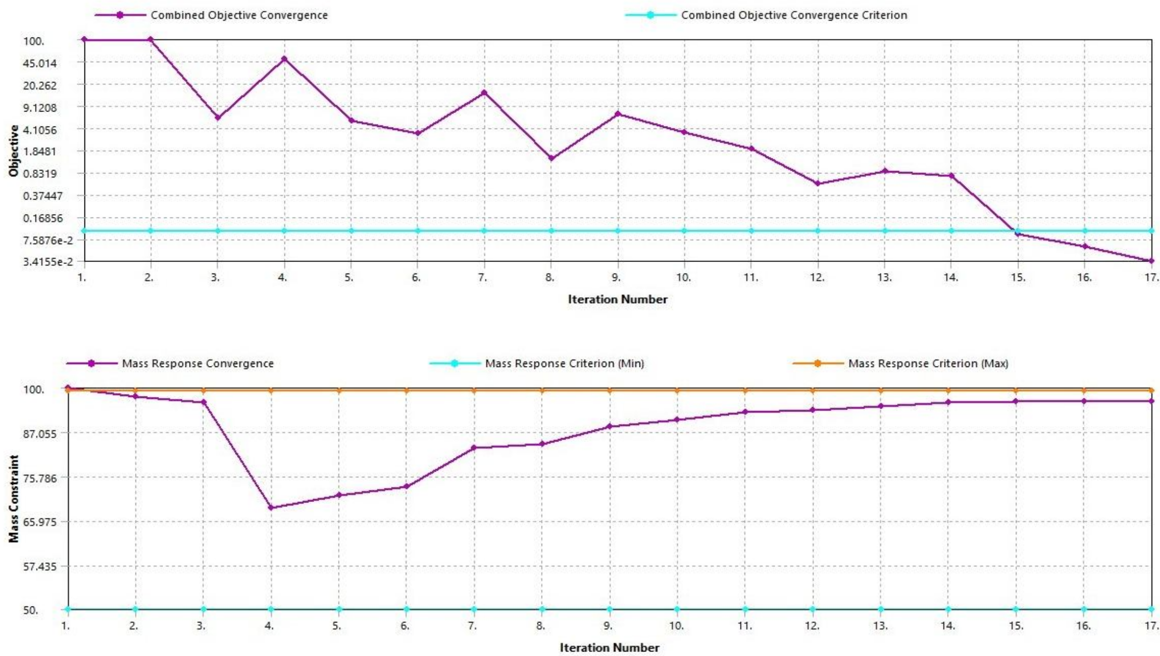
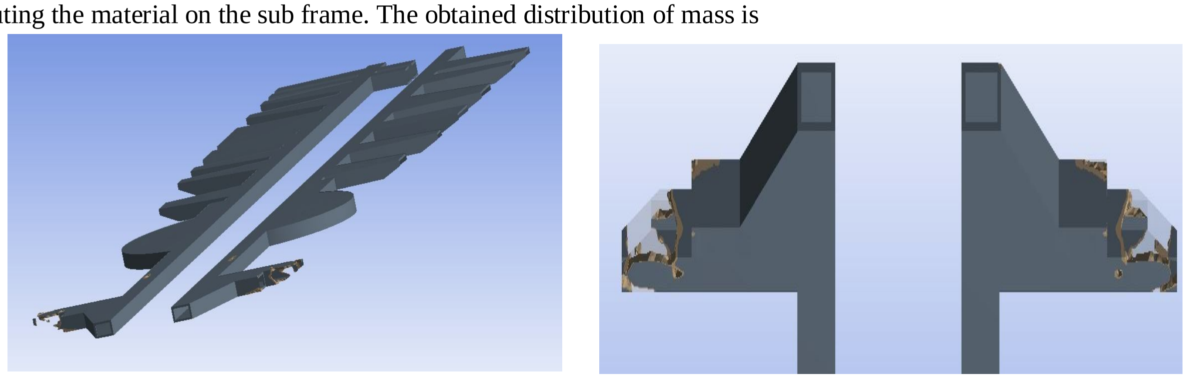


Fig. 17 Convergence graph for dynamic topology optimisation

The desired frequency range is kept between 20 Hz to 25 Hz for 9th mode of vibration away from 30 Hz of on board vibrations. this will shift the mode shape and frequency of 9th mode of vibration from 30.57 Hz to somewhere between 20 Hz to 25 Hz by redistributing the material on the sub frame. The obtained distribution of mass is



(a) (b)

Fig. 18 (a) and (b) showing optimised mass distribution of sub frame

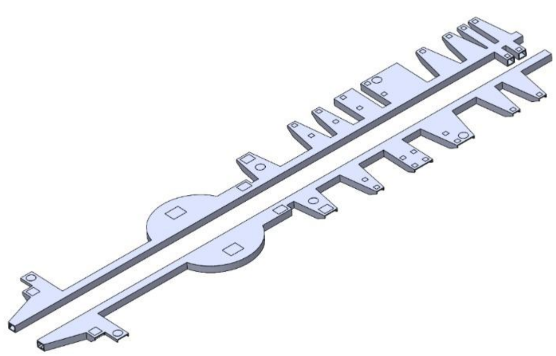


Fig. 19 Feasible design after dynamic topology optimisation

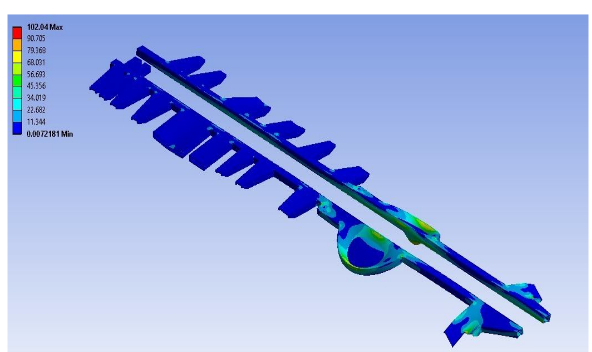


Fig. 20 Stress level of new feasible design

shown in figure 18 (a-b). The objective is achieved after 17th iteration as shown in figure 17. the topology optimisation approach suggests removal of material from transverse part of rear outrigger location and hence accordingly a new feasible design (figure 19) is modelled and validated. By including different region of sub frame other than shown in figure 16 in design optimisation and different value of mass retention, one can have different topology of subframe still satisfying objective function and constraints. This new feasible design is subjected to 1g load and stress level is evaluated. the von misses stress found is same as earlier i.e. 102 MPa as shown in figure 20. The mass of new feasible design is 5792 Kg.

D. Modal analysis

Modal analysis of new feasible design of sub frame with same boundary condition as previous is carried out to determine mode shapes and undamped natural frequencies. It shows that the 9th mode of vibration (figure 15i) is shifted to 5th mode of vibration (figure 21e) and the critical frequency is removed. the natural frequency of that mode is change from 30.57 Hz to 21.94 Hz away from 30 Hz of excitation frequency and within the desired frequency range. there is a considerable change in frequency due to redistribution of material at rear outrigger location obtained through dynamic topology optimisation.

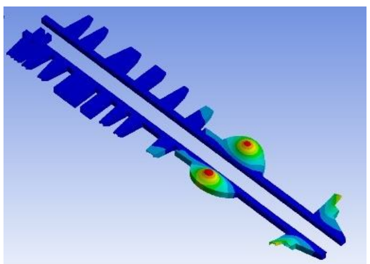


Fig. 21(a) 1st mode with 12.39 Hz

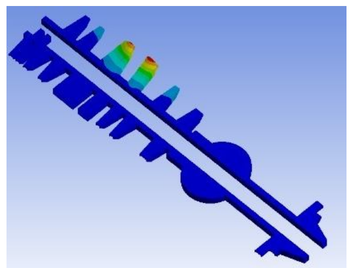


Fig. 21(b) 2nd mode with 15.36 Hz

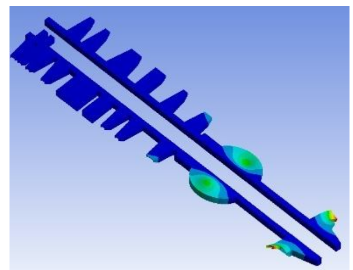


Fig. 21(c) 3rd mode with 17.51 Hz

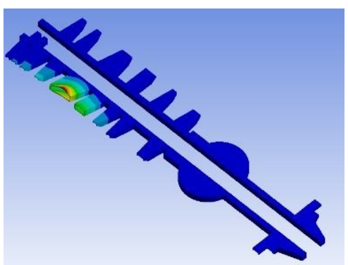


Fig. 21(d) 4th mode with 18.45 Hz

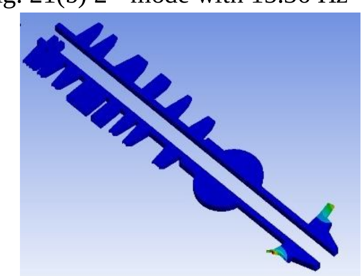


Fig. 21(e) 5th mode with 21.94 Hz

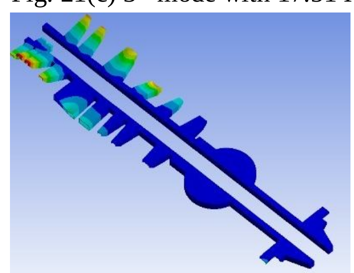


Fig. 21(f) 6th mode with 22.12 Hz

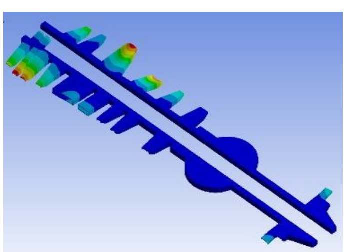


Fig. 21(g) 7th mode with 22.99 Hz

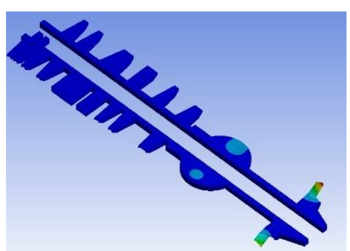


Fig. 21(h) 8th mode with 23.23 Hz.

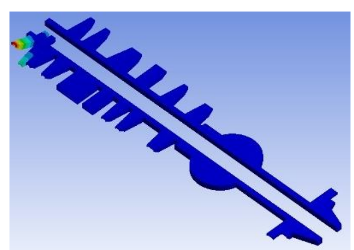


Fig. 21(i) 9th mode with 25.15 Hz

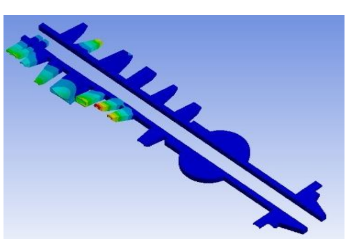


Fig. 21(j) 10th mode with 27.18 Hz

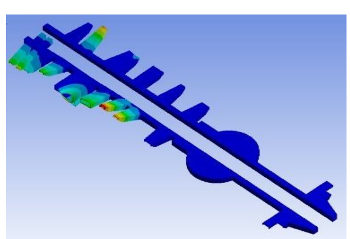


Fig. 21(k) 11th mode with 28.21 Hz

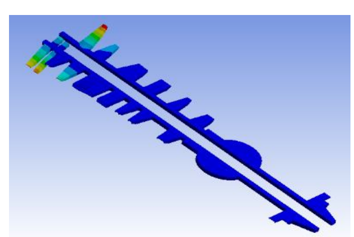


Fig. 21(l) 12th mode with 32.14 Hz

Fig. 21 (a-l) 1st to 11th mode of vibration with corresponding natural frequencies

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E. Harmonic Analysis

The dynamic characteristics of developed feasible design of sub frame with and without frequency constraint in topology optimisation is evaluated by giving sinusoidal excitation generated by load shell 2 and 3 mounted on the Sub frame to hinged location of tilt beam in transverse (Y direction). It is assumed that load shell 2 and 3 are rotating with maximum amplitude of 1000 N at 1800 rpm. to measure accurate response of the sub frame, first 30 modes of vibration from modal analysis is evaluated in harmonic analysis making ratio of effective mass to total mass around 90 in harmonic load direction (Y). this is necessary since mode superposition method is used to evaluate frequency response of the sub frame with and without frequency constraint. A harmonic load of 1000 N with frequency sweeping from 25 Hz to 35 Hz is input to hinge area of tilt beam on sub frame as shown in figure 22. Damping of 2% is given to the system.

The response is measured at the hinged location of tilt beam on the sub frame. A sharp peak at 30.5 Hz (figure 23) is seen on the sub frame due to harmonic vibrations from load shell 2 and 3 before applying frequency constraint in topology optimisation. On the other hand, there is no peak after topology optimisation with frequency constraint indicating removal of resonant frequency from the sub frame. It is clear from the graph that the newly obtained sub frame is dynamically stable against excitation from on board vibrations. the performance of payload will not be affected due to functioning of load shell 2 and 3 under this dynamically optimised sub frame and is best suited for RTV. Figure 24 shows various subsystems mounted on dynamically optimised sub frame fitted to HMV 12X12. The obtained design of sub frame is comparatively in good agreement with industries guidelines and more suitable than initial rectangular sub frame.

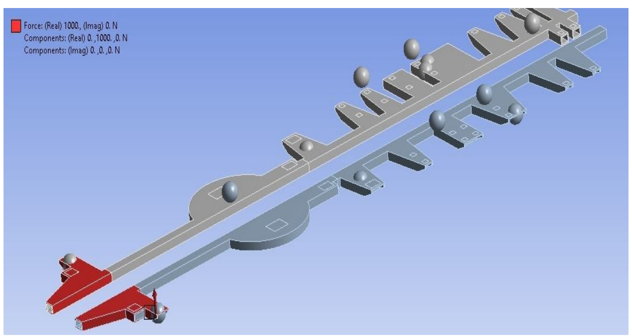

Fig. 22 Input region of harmonic load

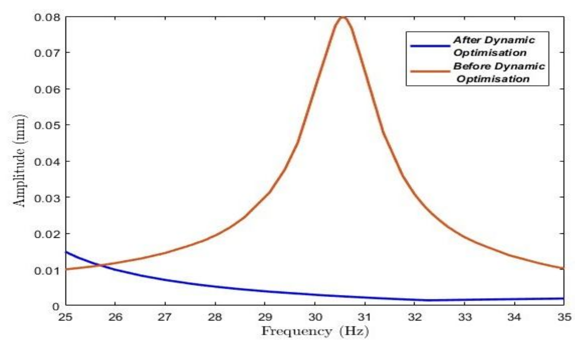

Fig. 23 Responses at hinged

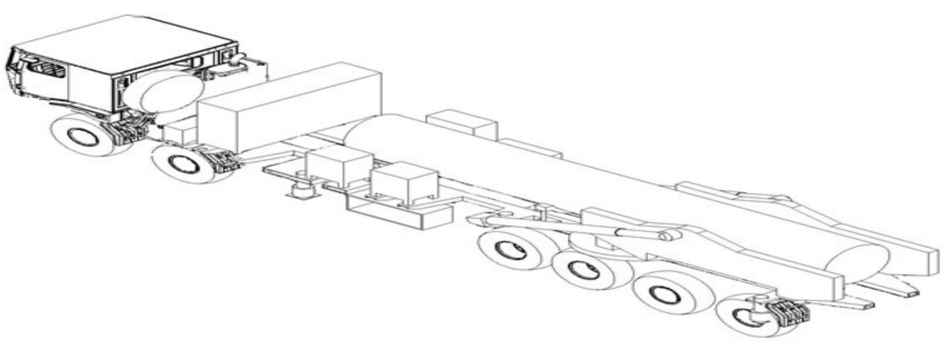

Fig. 24 RTV mounted with dynamically optimised sub frame

VI. CONCLUSIONS

A comparative study was conducted to investigate the design limitations of mechanical structures subjected to vibration loads. An alternative approach is proposed to achieve a resonance-free structure by redistributing material while considering mass and dynamic characteristics as constraints in topology optimization. In this study, the SIMP method with frequency constraints was employed to develop a sub-frame with a high strength-to-weight ratio and free from critical modes and resonant frequencies. once the optimal material distribution was obtained, a manufactural design considering practical fabrication constraints was developed and validated through linear static stress analysis. The initial rectangular sub-frame, with a mass of 47,262 kg, exhibited a maximum stress of 5.52 MPa under 1g static loading with all subsystems mounted on it. In contrast, the dynamically optimized sub-frame, under identical boundary conditions, showed a maximum stress of 102.34 MPa against a material yield strength of 690 MPa, with a significantly reduced mass of 5,792 kg. This corresponds to an 87.74% reduction in sub-frame mass with only a moderate increase in stress. Modal and harmonic analyses were performed to evaluate the real-time dynamic effects induced by on board rotating machinery. Although the final stress levels, mass, and shape of the optimized sub-frame may vary depending on the initial dimensions and mass retention criteria, the proposed methodology demonstrates a novel design procedure beyond conventional topology optimization practices.

VII. ACKNOWLEDGMENT

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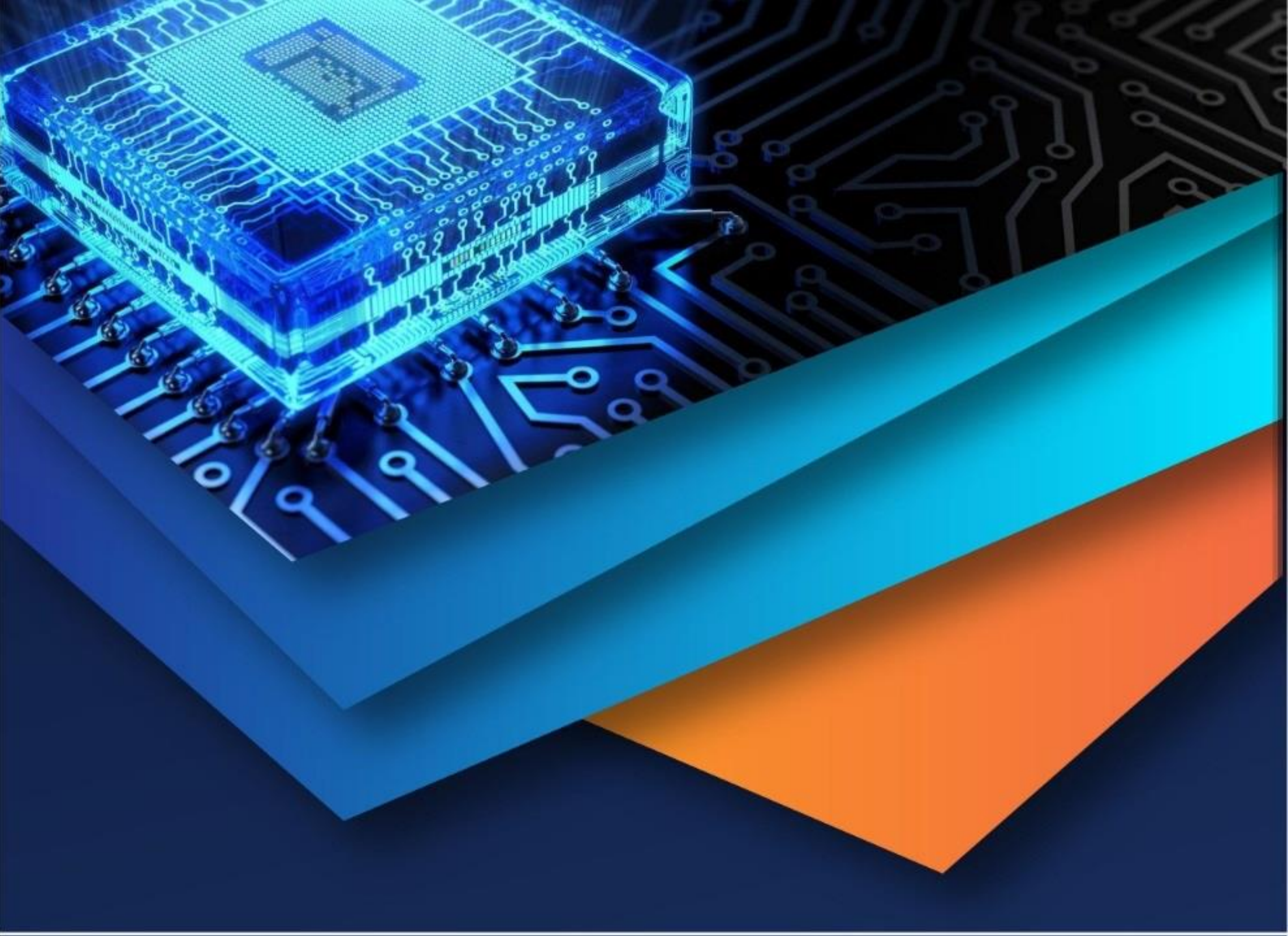
REFERENCES

[1] Bendsoe, M.P, “Optimal shape design as a material distribution problem,” Struct. Optim., 1989, 1(4), 193– 202.
[2] Bendsoe, M.P. & Sigmund, O, “Material interpolation schemes in topology optimization,” Arch. Appl. Mech., 1999, 69 (9-10), 635–54.
[3] Nilesh Patel, and Vinay kumar Rokade, “Topology Optimisation and Fabrication Aspects for Light Weight Design of an Articulating Beam of Article Launching System,” Defence Science Journal, Vol. 68, No. 1,26-32.
[4] Rietz, A, “Sufficiency of a finite exponent in SIMP (power law) methods,” Struct. Multidiscip. Optim.,2001, 219(2), 159–63.
[5] Volkan Kandemir, OguzDogan and UlasYaman, “Topology optimisation of 2.5 parts using SIMP method with a variable thickness approach,” Procedia Manufacturing 17 (2018) 29–36.
[6] Hongling Ye, Ning Cheng, Yunkang Sui, and Jun Tie, “Three-Dimensional Dynamic Topology Optimization withFrequency Constraints Using Composite Exponential Function and ICM Method,” Hindawi Publishing Corporation Mathematical Problems in Engineering Volume 2015, Article ID 491084.
[7] Y. Maeda1, S. Nishiwaki1, K. Izui1, M. Yoshimura1, K. Matsui and K. Terada, “Structural topology optimization of vibrating structures with specified eigen frequencies and eigen mode shapes,” International journal for numerical methods in engineering, Int. J. Numer. Meth. Engng 2006; 67:597–628.
[8] Y. M. Xiet and G. P. Steven, “Evolutionary structural optimization for dynamic problems,” Compufers & Sm,crures Vol. 58, No. 6. pp. 1067.-1073, 1996.
[9] Jong Wook Lee, Gil Ho Yoon and Seung Hyun Jeong, “Topology optimization considering fatigue life in the frequency domain,” Computers and Mathematics with Applications 70 (2015) 1852–1877.
[10] Shubham Saurabh, Abhinav Gupta, Rajib Chowdhury and RavindraDuddu, “Robust topology optimization for uncertainty in load positions of transient dynamic loading,” Reliability Engineering and System Safety 264 (2025) 111440.
[11] Xuqi Zhao et al., “Topology optimization of proportionally damped structures under harmonic excitations: Analysis of velocity and acceleration responses,” Engineering Structures 258 (2022) 114140.



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