

# Modal Analysis of Ship's Mast Structure using Effective Mass Participation Factor

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## Abstract

**Background/Objectives:** Each structure tends to vibrate at particular frequencies, called resonant or natural frequencies. When a structure is excited by dynamic load with frequency coinciding one of its natural frequencies the structure experiences stresses and large displacements. In this paper effective mass participation factor criterion is used to solve the vibration problem in the ship mast. **Methods/Statistical Analysis:** The effective mass participation factor provides a measure of the energy contained within each resonant mode. Vibration problem originated when one of the antenna at top of mast was replaced by a new antenna with greater mass at same location. The overall mast structure started vibrating because of the resonance of natural frequencies of the mast structure with natural frequencies of rotary equipment. **Findings:** It caused interruption in sensitivity of equipment installed on the mast structure. Instead of fabricating the new mast structure, some alteration has been carried out on the basis of results obtained from modal analysis. **Application/Improvements:** The study is very effective to overcome the vibration problems in ship mast.

**Keywords:** Effective Mass Participation, Modal Analysis, Mode Shape, Ship Mast

## 1. Introduction

It is well known that mechanical structures can resonate, i.e. small forces can result in deformation. Resonance can likewise harm the complete structure. Resonant vibration is essentially brought on by an interaction between the inertial and elastic properties of the materials inside of a structure<sup>1</sup>. Resonance is often a contributing factor to many of the vibration and noise related problems that occur in structures and operational machinery. The resonant frequencies of a structure should be identified and measured to better see structural behavior<sup>2</sup>.

Modal analysis has turned into a far reaching method for finding the modes of vibration of a machine or structure<sup>3</sup>. Modes are inherent properties of a structure, and are determined by the material properties (mass, damping, and stiffness), and boundary conditions of the structure. Each mode is characterized by a natural frequency, modal damping, and a mode shape. Either the material

properties or the boundary conditions of a structure change, its modes of vibration will modify<sup>4</sup>.

## 2. Effective Mass Participation Factor (EMPF) Method

The Effective Mass Participation Factor (EMPF) provides a measure of the energy contained within each resonant mode. It represents the quantity of system mass participating in a particular mode<sup>5</sup>. For a particular structure, with a mass matrix  $[M]$ , normal mode shapes and a ground motion influence coefficient  $r$ , participation of each mode can be obtained as the effective mass participation factor,

$$P_i = \frac{\phi_i^{\rightarrow T} [M] \vec{r}}{(\phi_i^{\rightarrow T} [M] \phi_i^{\rightarrow})^2} \quad (1)$$

A mode with a large effective mass is generally a major contributor to the response of the system. It is possible to

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calculate an EMPF for a particular direction (x, y or z). The sum of the effective masses for all modes in a given response direction must be equal to the total mass of the structure. Priestley et al. confirm that a sum of all EMPF (known as Cumulative Effective Mass Participation Factor, CEMPF) of 80% to 90% in any given response direction can be considered sufficient to capture the dominant dynamic response of the structure<sup>6</sup>.

$$80 \leq 100 \sum_{i=1}^n p_i \leq 90 \tag{2}$$

Where n is the number of modes taken under consideration. If we expect a vibration in the x-direction, we need to keep calculating modes until the sum of all EMPF in the x-direction is about 80-90%. We need consistency in the results to compare the exciting frequency with the sufficient natural frequencies<sup>7</sup>.

### 3. Rules and Standards for Modal Analysis

#### 3.1 BV Rules

As stated by bureau VERITAS rules for naval ships, “each normal mode frequency  $f_{Ni}$  (Hz) of the ship mast should be in compliance with one of the following formulae:

$$0.8f_{E(MIN)} > f_{Ni} > 1.2f_{E(MAX)} \tag{3}$$

where  $f_{E(MIN)}$  (Hz) and  $f_{E(MAX)}$  (Hz) are the minimum and the maximum possible excitations frequencies respectively due to the ship’s motions or the propulsion system at the following speeds;- cruise speed and maximum continuous rate speed. At the point when the dynamic analysis depends on normal modes, their number is, all in all, to be such that the modal effective mass is at the very least 95% of the mass of the system constituted by the mast and its supporting structure. The modal effective mass is defined as.

$$\sum_{i=1}^N \gamma_i^2 \tag{4}$$

Where  $\gamma_i$  is the *i*th modal participation factor and *N* is the number of the considered normal modes”<sup>8</sup>.

#### 3.2 Military Standards

The maximum vibration limits for top of the ship mast are given in Table 1<sup>9,10</sup>.

**Table 1.** Maximum Limit for Top of Ship Mast

Mechanical Test	Top of Mast Limits	Standard
Vibration	Amp ± 1mm to 16 Hz 10m/s <sup>2</sup> 16 to 23 Hz	(MIL-STD 810 G)
	Endurance 30 minute ± 1 mm to 16 Hz	

## 4. Modal Analysis of Mast Structure

### 4.1 Modeling of Structure

Antenna installed on top of the Mast was replaced with a new antenna having 80Kg weight (which is 50 kg heavier than previous). Due to change in mass at top of the mast, the mast structure started to vibrate during cruising of the ship. Also data receiving from antenna was distorted and wrong due to vibrations. This problem was sent for finite element analysis and proposal of a solution to cater vibrations in ship mast. The frequency of equipment installed on the ship at various speed is mentioned in Table 2.

Material used in mast structure was Aluminum. Young’s modulus (E), Poisson ratio ( $\nu$ ) and density ( $\rho$ ) of Aluminum are 71000MPa, 0.3 and 2700 kg/m<sup>3</sup> respectively. Three equipments installed on ship mast with 320 kg, 62kg and 80 kg mass respectively.

### 4.2 Modal Analysis of Mast Structure

Frequencies which amplify the effect of load on a structure are identified by modal analysis. It is also used to find the mode shapes of a structure. The displacement shape of modes provides enough information about loose component(s) or missing constraints. The model will have six rigid body modes if a single component has no constraints. They correspond to the component translating respectively in X, Y and Z and rotating about X, Y and Z. Mode at an inconvenient frequency needs to be moved or rather change the frequency of the mode. The frequency can be amplified or lessened according to requirements of the structure. For a single degree of freedom spring-mass, the natural frequency is,

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \tag{5}$$

Where *m* is the mass and *k* is the spring stiffness. Increasing the natural frequency requires a decrease in *m* or an increase in *k*. Finite element model of existing

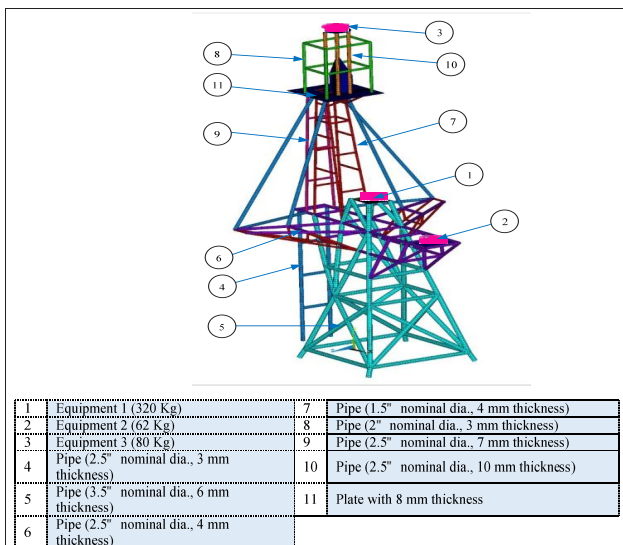
mast structure with initial stiffening and equipment loads applied on the mast structure by using point mass element are shown in Figure 1.

Mast structure was modeled by using beam and shell modal. All the pipe structure was modeled by using the beam-189 element type whereas plates were modeled using shell-181 element. Mass 21 was used for equipment mass. The element edge length of meshing was 50 mm for both pipes and plates. Hence the total number of elements was 1200. Different modes were extracted using modal analysis in ANSYS 15.0. Mode shape of mast structure at different frequencies (only critical) is shown in Figure 2.

The results in Figure 2 shows that almost all the frequencies are producing the resonance according to BV Rules limits. Also the results are not validating the Military Standard Limits for Vibration of Mast because

**Table 2.** Frequency of Equipment installed on the Ship

Sr. #	Ship Speeds	Exciting Frequencies	
01	Cruise speed	Engine	10.03
		Shaft	4.98
		Propeller	24.9
		Mast equipment	< 1
02	Maximum Speed	Engine	18.4
		Shaft	7.96
		Propeller	39.80
		Mast equipment	< 1

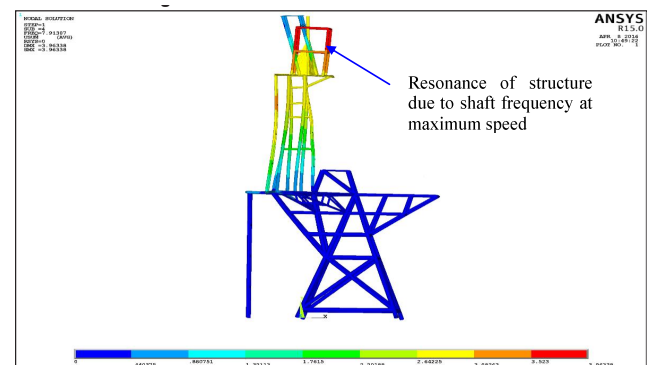


**Figure 1.** Isometric View of Mast Structure.

the displacements are greater than the limits defined in the standards. By plotting all the mode shapes it can be interpreted that the main reason of the resonance is weak stiffness at the top and middle of the mast structure.

At this stage the Effective Mass Participation Factor (EMPF) becomes the critical point as all the modes are not actively participating in the resonance of structure. As a next step the effective mass participation factor of each mode in checked and modes participating having more than 1% EMPF are segregated and also verified the Cumulative Effective Mass Participation Factor (CEMPF) which should be between 80-90 %. Relative displacements are obtained from modal analysis (free vibration analysis). All 40 modes are causing resonance in the structure but after applying effective mass participation factor criteria, following modes are participating more than 1% and also not validating the Military Standard Limits for Vibration hence these are critical.

The results shown in Table 3 that by applying the 1% EMPF criteria the modes producing effective reso-



**Figure 2.** Resonance of Structure Due to Shaft Frequency.

**Table 3.** Modes Having EMPF More Than 1%

Mode	Resonant Frequency (fNi)	Range		CEMPF (%)	EMPF (%)
		0.8 (fNi)	1.2 (fNi)		
3	5.07	4.06	6.08	19.76%	1.16%
4	7.91	6.33	9.50	53.96%	34.20%
8	12.60	9.88	14.82	84.63%	30.63%
9	12.53	10.08	15.12	91.33%	6.70%
19	27.25	21.80	32.70	94.01%	1.58%
28	33.61	26.89	40.33	95.97%	1.34%
38	41.45	33.16	49.74	98.52%	1.59%
40	43.29	34.63	51.95	100.00%	1.34%

nance reduced but still these modes are not satisfying the MIL-STD 810 G standard as the displacement are not within the range. For effective utilization of mass participation factor method on the mast structure to cater resonance the mass and stiffness of the structure has to be changed. As a result mass and stiffness has been increased in the structure shown in Figure 3. These modifications were proposed for fabrication.

### 5. Results

Modified structure was analyzed using ANSYS. Resonance in the structure was observed due to shaft frequency at maximum speed. Similarly engine frequency at maximum speed caused resonance in the structure. But the amplitude was within limits of  $\pm 1\text{mm}$ ; hence fulfill the requirements of military standard MIL-STD 810-G.

Results shown in Table 4 are within the limitations of the MIL-STD 810-G standard for vibration and resonance of the mast top. Same procedure for EMPF was adopted for modified mast structure. Improvement in the behavior of structure is shown in Figure 4. Only two modes were participating more than 1% but the resonance amplitude was within limits of military standards.

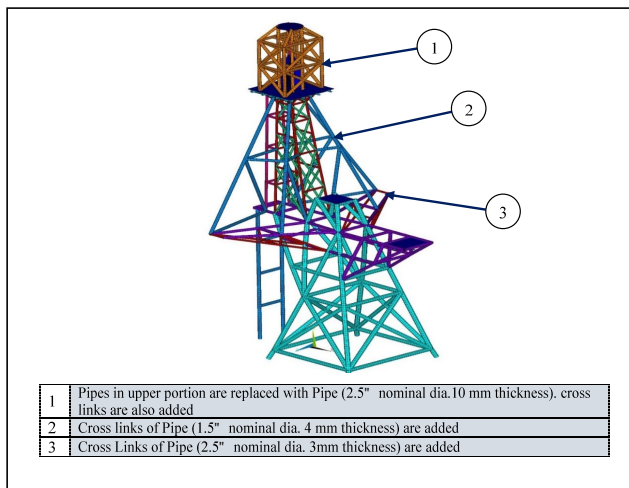


Figure 3. Modified Ship Mast Structure.

Table 4. Bureau VERITAS Criteria for Mode analysis

Mode	Resonant Frequency ( $f_{Ni}$ )	Range		CEMPF (%)	\EMPF (%)
		0.8 ( $f_{Ni}$ )	1.2 ( $f_{Ni}$ )		
2	6.70	5.36	8.04	50.44%	50.48%
5	17.12	13.69	20.54	80.92%	30.50%

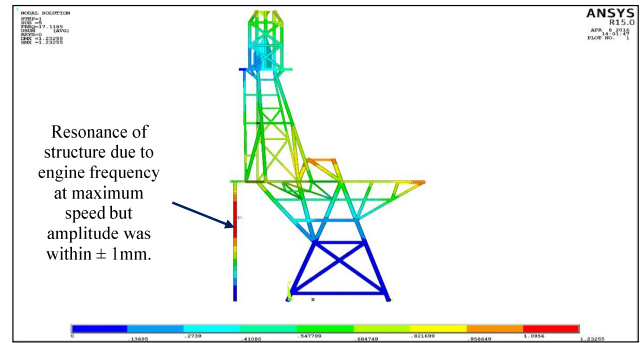


Figure 4. Resonance of structure due to engine frequency.

Table 5. Experimental results using tri-axis spectral vibration recorder

Frequency (Hz)	Amplitude (mm)	
	Cruise Speed	Maximum Speed
6	0.35	0.40
7	0.51	0.55
8	0.38	0.85
9	0.31	0.50
10	0.25	0.30
11	0.21	0.36
12	0.17	0.23
13	0.15	0.27
14	0.07	0.43
15	0.05	0.50
16	0.01	0.25

Experimental data given in Table 5 after the modification in the structure was obtained by using tri-axis spectral vibration recorder OMPCP 2.07. This data validated the numerical results.

### 6. Conclusion

Modal analysis of Ship mast with new antenna having greater mass was carried out in the ANSYS software ver. 15.0 using mass participation factor method. Resonance of the structure was identified and then modification in the existing structure was recommended for fabrication. Fabricated structure was observed using tri-axis spectral vibration recorder. Displacement values were within limit of  $\pm 1\text{mm}$  and hence within the limitations of the MIL-STD 810 G. Modal analysis gives us the rela-

tive displacements of any structure under free vibration. In future ship mast vibration spectrum will be analyzed using forced vibration analysis to get actual displacements and will be compared with experimental data.

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