Dynamic Analysis of A Fixed Ended Beam with Focus on Vibration Neutralization in Ship Hull Structure

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ABSTRACT

Ship hull vibration has always been a subject of great interest to the naval architects because of its adverse effects both upon the ship's structure and upon the comfort of passengers and crews. Vibration can take place in beams or structures which can be approximated as beams like the deck beams, hull girders, decks, hatch covers, shafts etc of a ship. This paper investigates firstly the transverse modes of vibration characteristics of a fixed ended beam. Then direct frequency analysis of the beam is performed where a point load is applied at the centre of the beam at different frequencies to find out the amplitude of any point on that beam at those frequencies. The NASTRAN finite element software has been used for this purpose. After computing the natural frequencies of the beam using NASTRAN the results are compared with the theoretical values. Then using the modal analysis and direct frequency analysis, an effective approach to study the vibration neutralization of the beam is presented. The study has a great impact on ships as well as offshore structures as these are composed of beams which are frequently under periodic excitation from motor or engine.

Keywords: Beam, Vibration, Natural Frequency, Modal Analysis, Direct Frequency Response Analysis.

INTRODUCTION

Vibration has always been a very important subject for naval architects and structural engineers as its presence can seriously affect the comfort of passengers on ships and the integrity of structures like ships, bridges, offshore structures, airplanes, cars etc. It is well known that structures can resonate, that is, small forces can result in significant deformation, and possibly, damage can be induced in the structure.

"Ship hull vibration is an old but a new problem", is often said by many naval architects and marine engineers. That is because the vibration of the hull structures caused serious problems in old times and that it still now brings new kinds of problems. In addition, it is also frequently seen that when some area vibrates heavily, vibration can be decreased by providing reinforcements to that part. Nevertheless, these reinforcements are likely to cause vibration in another area. As a result, although additional weight is added but the vibration of structure is not removed completely. This justifies the use of vibration neutralizers which do not transfer vibration in any other structural members.

Beams are basic structural members of a ship and their vibration analysis is thus very important. Vibration of deck beams of a ship takes place when the forcing disturbances come from the shafting or propellers (Todd, 1961). Okumoto et al. (2009) mentioned that the unbalanced forces of engines are so large that they can produce hull girder vibration. They also showed that the deck of a pure car carrier and a hatch cover of a bulk carrier can be effectively modeled as a beam to calculate their natural frequencies as these members can be subjected to vibrations.

Many researchers have worked on transverse vibration of beams and as this is a subject of practical engineering interest, has been the objective of many recent theoretical investigations. A review of the past works is presented below.

The bending linear vibration of an elastically restrained beam carrying concentrated masses located within the beam span was analyzed by Hamdan and Jubran (1991) and in the analysis, the base beam equation of motion is solved to obtain mode shape functions which satisfy all the geometric and natural boundary conditions at the beam ends. These functions are used in conjunction with Galerkin's method to obtain the free and the forced response.

Khalil and Islam (1999a) presented a detailed mathematical analysis of the principle of vibration neutralizers. It is shown that the neutralizer exercises its effect in neutralizing vibrations when its own natural frequency matches with that of the exciting force. This paper is taken as a guide to the present study in designing the characteristics of vibration neutralizer to effectively avoid resonance vibration of a beam.

Khalil and Islam (1999b) demonstrated the possibility of reducing vibrations in the deck beams of a ship by using vibration neutralizers. It was proved in this paper that if a simply supported deck beam of uniformly distributed mass is replaced by a spring-mass system of the same natural frequency, then the mass of the latter system will be nearly equal to half of the total mass of the deck beam, and the spring constant will be equal to the stiffness coefficient of the beam.

Rossit and Laura (2001) presented the exact solution of free vibrations of a cantilever beam with a spring–mass system attached to the free end using the Bernoulli–Euler theory of beam vibrations. Natural frequencies are obtained for a wide range of the intervening physical parameters. The problem is of interest in naval and ocean engineering systems since in order to avoid dangerous resonance conditions the designer must be able to predict natural frequencies of the overall mechanical system: structure–motor and its elastic mounting.

Zhou and Ji (2006) studied the dynamic characteristics of a beam with continuously distributed spring-mass which may represent a structure occupied by a crowd of people. Dividing the coupled system into several segments and considering the distributed spring-mass and the beam in each segment being uniform, the equations of motion of the segment are established. The transfer matrix method is applied to derive the eigenvalue equation of the coupled system. It is interesting to note from the governing equations that the vibration mode shape of the uniformly distributed spring-mass is proportional to that of the beam at the attached regions and can be discontinuous if the natural frequencies of the spring-masses in two adjacent segments are different. Parametric studies demonstrate that the natural frequencies of the coupled system appear in groups.

There are many other researchers who have made important contributions in the field of dynamic analysis of beams. They are Lau (1984), Schafer and Holzach (1985), Kojima et al. (1986), Liu and Huang (1988), Nagaya and Ishikawa (1995) etc. The target of these studies is to understand the behavior of beams so that proper remedial measure can be taken to control the vibration that may result.

MODAL ANALYSIS FOR DETERMINING NATURAL FREQUENCIES

Natural frequencies and mode shapes of a fixed ended beam for transverse vibration

Let us consider a fixed ended beam as shown in *Figure 1*. Let A be the cross sectional area, E be the modulus of elasticity, ρ be the mass per unit length, ρ' be the density, I be the moment of inertia and L be the length of the beam, then the expressions of the natural frequency of transverse vibration and mode shapes are as follows:

Natural frequency $\omega_i = \frac{\lambda_i^2}{L^2}\left(\frac{EI}{\rho}\right)^{1/2}$ in radian per second (i = 1, 2, 3, 4 and 5) and

Natural frequency $f_i = \frac{\lambda_i^2}{2\pi L^2}\left(\frac{EI}{\rho}\right)^{1/2}$ in Hz (i = 1, 2, 3, 4 and 5) (1)

where

λ_1 = 4.73004074, λ_2 = 7.85320462, λ_3 = 10.99560790, λ_4 = 14.13716550 and λ_5 = 17.27875970.

Mode shapes are given by

$$\cosh\frac{\lambda_i x}{L} - \cos\frac{\lambda_i x}{L} - \beta_i\left(\sinh\frac{\lambda_i x}{L} - \sin\frac{\lambda_i x}{L}\right) \qquad (2)$$

where $\beta_i = \frac{\cosh\lambda_i - \cos\lambda_i}{\sinh\lambda_i - \sin\lambda_i}$, i = 1, 2, 3, 4 and 5.

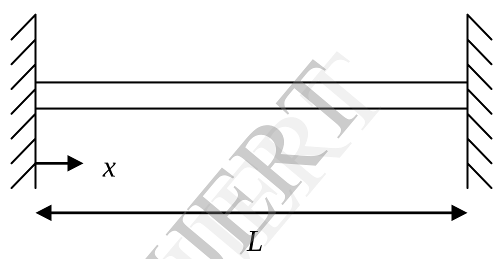


Figure 1. Fixed ended beam

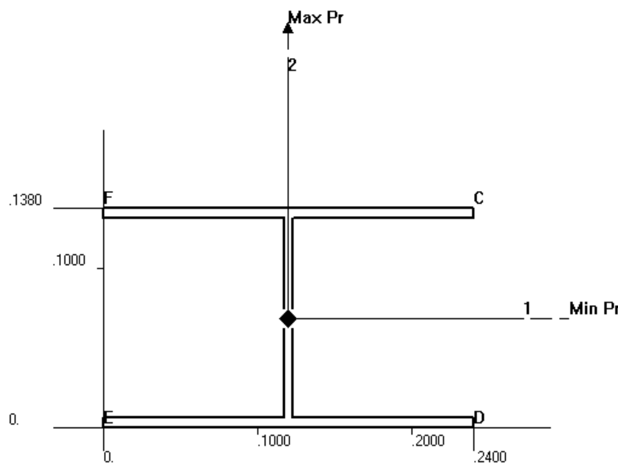
One may find the detailed derivation of the above expressions in Rao (2000).

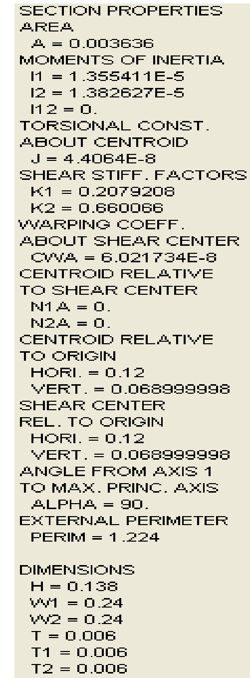
Modeling the beam

For analysis (both theoretical and finite element), the dimensions and characteristics of the beam are selected as follows:

E = 216 GPa, A = 0.003636 m^2, ρ' = 7800 kg/m^3 and L = 1 m

The beam is of I section with flanges 240 mm and web 138 mm width and overall thickness 6 mm as in *Figure 2(a)*. It is modeled in finite element pre-processing software MSC PATRAN to calculate the properties. The results are shown in *Figure 2(b)*.





(a) Shape of the cross-section (b) Properties

Figure 2. Beam property analysis by PATRAN

Free undamped frequency (modal) analysis

For finite element analysis, two models of beam (1 dimensional, 2 node beam element) is made. One has 50 elements and 51 nodes and the other one has 10 elements and 11 nodes. *Figure 3* shows the 10 element model.

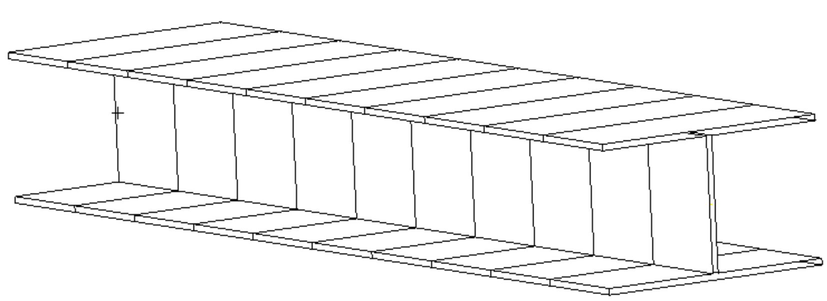


Figure 3. Finite element model of a beam having 10 elements and 11 nodes.

Results of natural frequencies found out by hand calculation using equation (1) and by finite element modal analysis (free vibration without damping) by professional software MSC NASTRAN are presented in *Table 1* for comparison. It is to be noted that finite element analysis produces flexural modes along with other modes like twisting, in-plane and mixed. We have only considered the flexural modes.

Table 1: Comparison of the natural frequencies of the fixed ended beam for transverse vibration

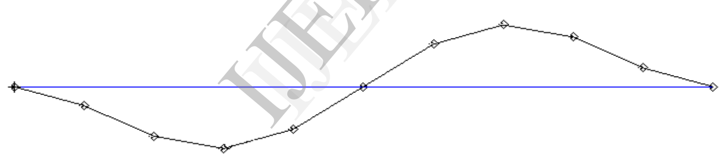
Normal mode	Hand calculation		Frequency in Hz for Model with 10 elements	Frequency in Hz for Model with 50 elements
	Frequency (Radian/sec)	Frequency (Hz)		
1	7188.287	1144.0524	1144.100	1144.100
2	19815.109	3153.6751	3154.500	3153.700
3	38845.526	6182.4627	6188.500	6182.500
4	64212.216	10219.7002	10247.000	10220.000

From *Table 1*, it is seen that the analysis of finite element model having 50 elements produce results (natural frequency in Hz) closer to those obtained by hand calculation.

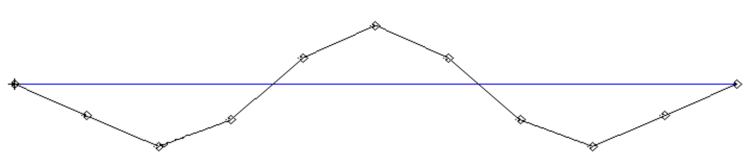
Mode shapes

Figure 4 shows, the mode shapes of the beam model having 10 elements.

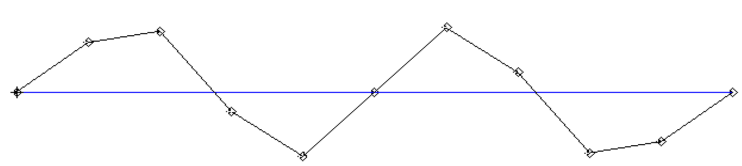
(a) First mode



(b) Second mode



(c) Third mode



(d) Fourth mode

Figure 4. First four flexural mode shapes of the beam model having 10 elements

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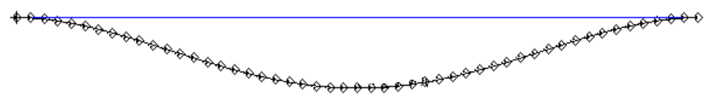
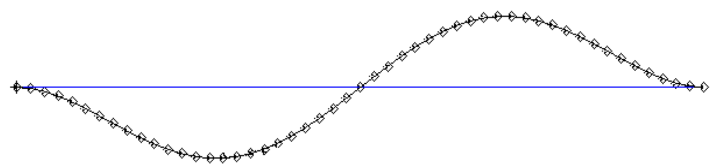
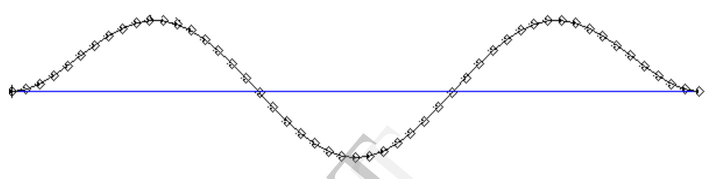
It is seen from the figure above that due to less number of nodes, the shapes are not properly achieved, especially mode 4. *Figure 5* shows the mode shapes of the beam model having 50 elements. It is clear from *Figure 5* that mode shapes are smoother. As modal frequencies are closer to theoretical values as well as the mode shapes are smoother, the 50 element model will be used for further analysis.



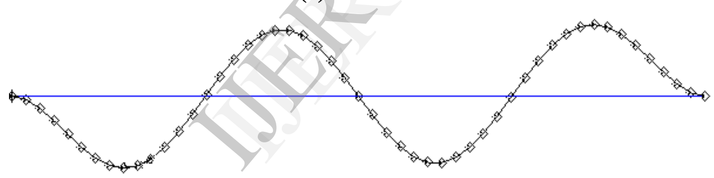
(a) First mode



(b) Second mode



(c) Third mode



(d) Fourth mode

Figure 5. First four flexural mode shapes of the beam model having 50 elements

DIRECT FREQUENCY RESPONSE ANALYSIS

Without damping

A 50 N harmonic load is applied at node 26 (mid point of the beam). The frequency range is from 0 to 10000 Hz at 100 Hz interval. The amplitudes of the nodes on the beam at all the frequencies in the applied range will be found out by this analysis. No structural damping is used. The response curves for nodes 7, 13 and 26 are shown in *Figures 6-8*.

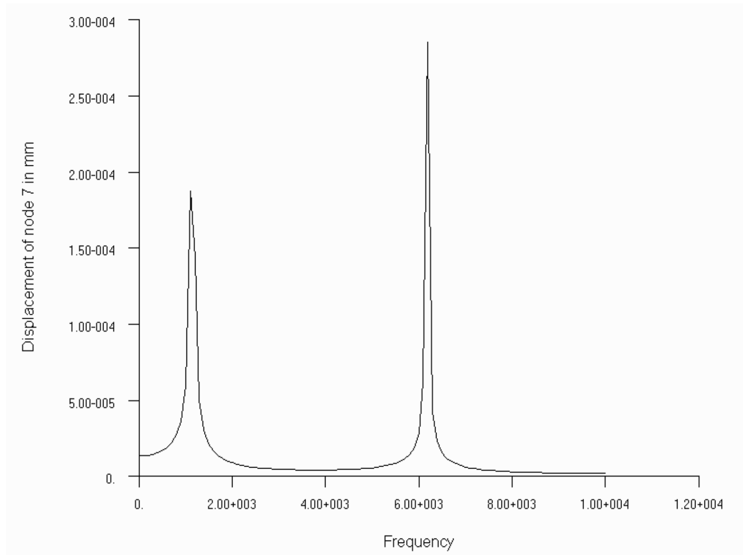


Figure 6. Amplitude of node 7 in mm

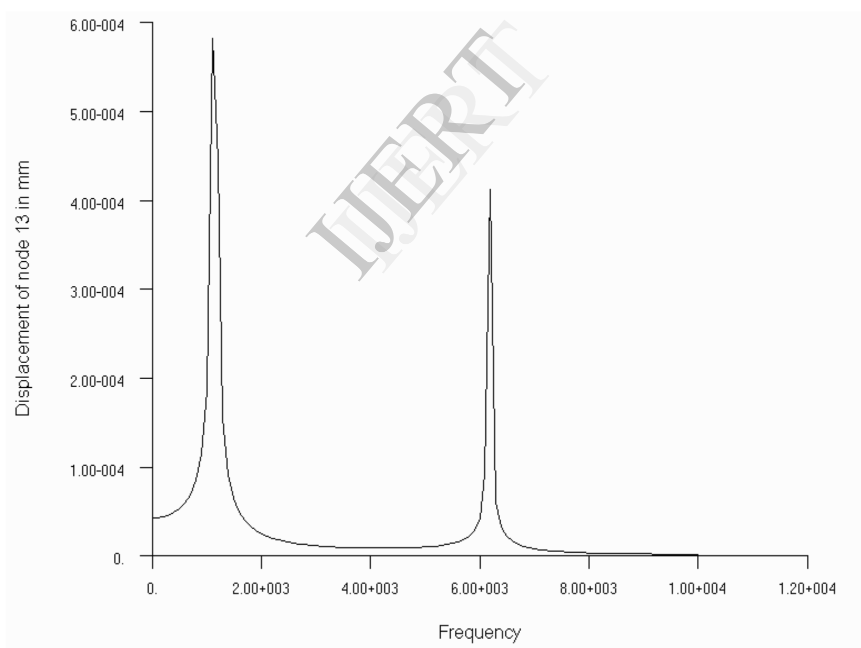


Figure 7. Amplitude of node 13 in mm

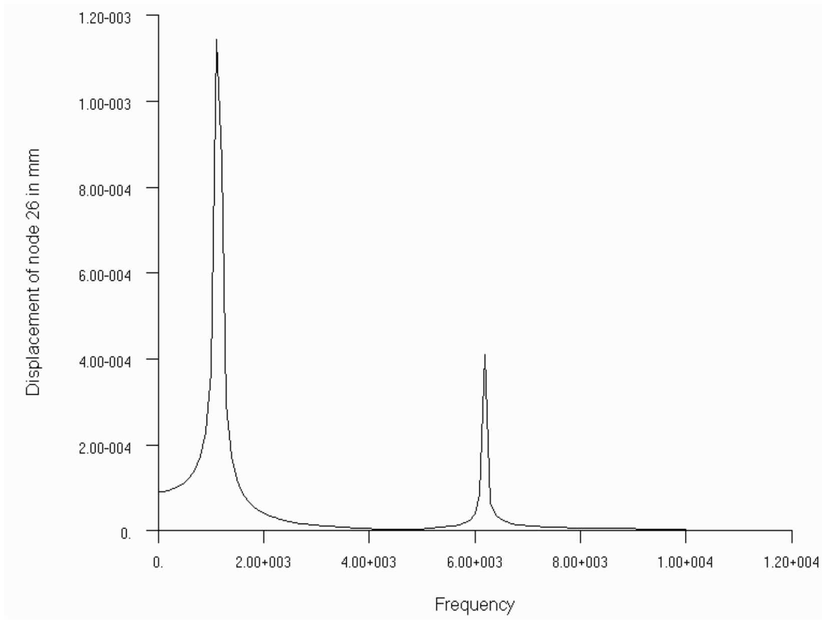


Figure 8. Amplitude of node 26 in mm

Though a structure has many natural frequencies, it is seen from *Figures 6-8* that for the beam in consideration, only two frequencies (first and third) are of importance. In other frequencies the amplitudes are very low or zero for the three reference nodes. From this analysis the important frequencies at which structural damage can take place can effectively screened out from all other frequencies. A 50 Hz interval was also done and similar response to this was found.

With damping

The same analysis is carried out considering the structural damping coefficient 0.01. Here damping is applied just to see the effect on the response curve. The response curves (structural damping coefficient. 0.01) for nodes7, 13 and 26 are shown below:

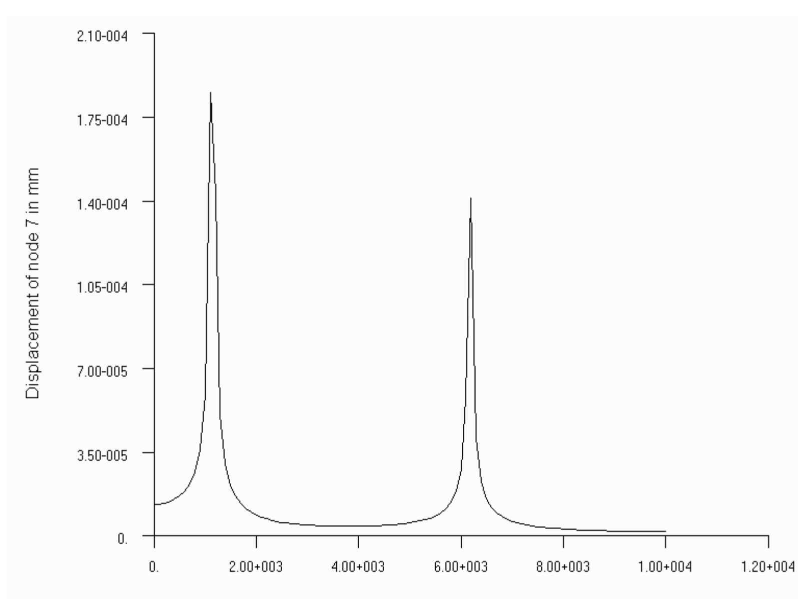


Figure 9. Amplitude of node 7 in mm

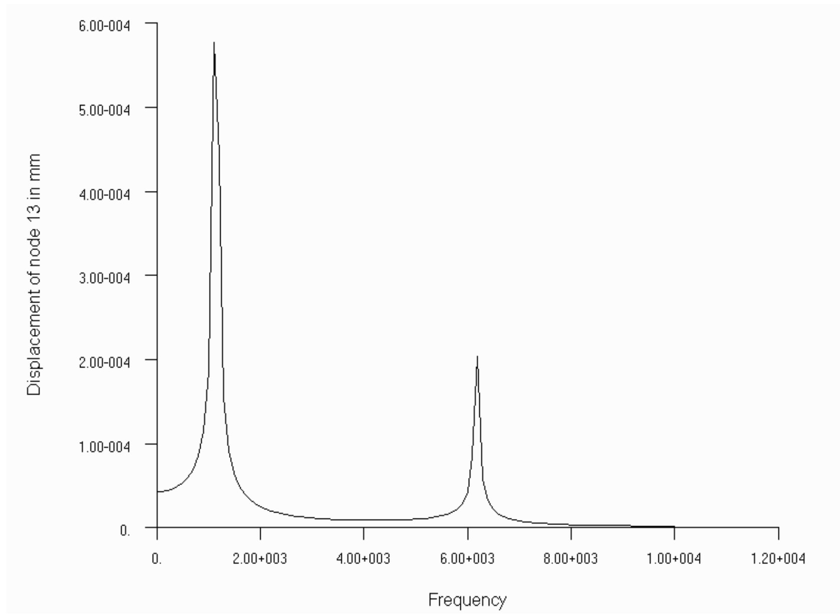


Figure 10. Amplitude of node 13 in mm

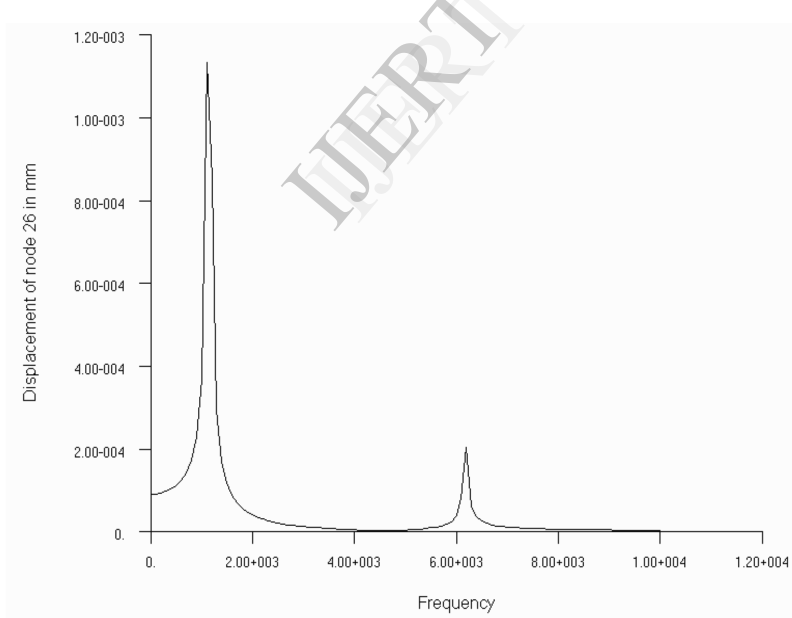


Figure 11. Amplitude of node 26 in mm

Comparing *Figures 6-8* and *9-11*, it is seen that the amplitudes have reduced value when structural damping is considered. It is seen that structural damping causes reduction of amplitude at third resonance frequency more than that at first resonance frequency.

VIBRATION NEUTRALIZATION OF THE BEAM

Case-1: An additional beam is attached to the original

From the figures above it is seen that the first natural frequency is 7188.287 radian/sec (1144.1 Hz). Now Khalil and Islam (October, 1999) showed that if a spring mass system is under harmonic excitation of frequency equal to that of the spring mass system, then the system will be in resonant vibration. If a vibration neutralizer (another spring mass system) is attached such that it has frequency equal to the harmonic excitation, the vibration of the main mass will be transferred to the vibration neutralizer leaving the main mass at rest.

Now we will investigate the effect of adding an additional beam of the same frequency (different cross sectional area and moment of inertia) to the original beam. If a 50 mm wide and 150 mm high bar is selected to attach, the length needed for the same frequency is 842.15 mm. For convenience, 800 mm length is considered to add as shown in *Figure 12*.

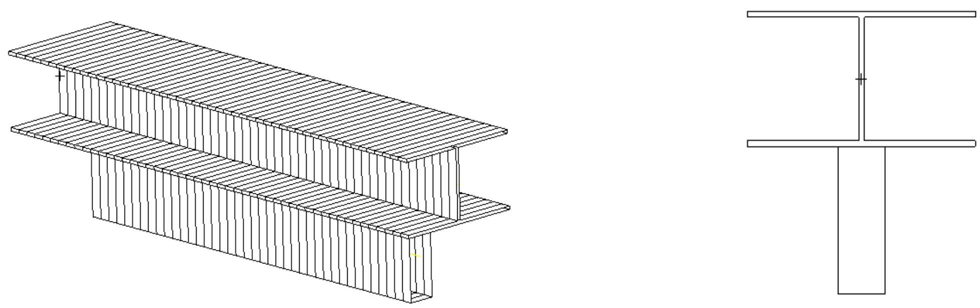
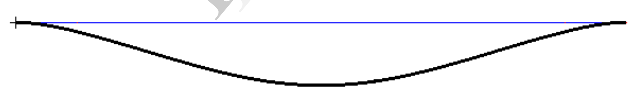
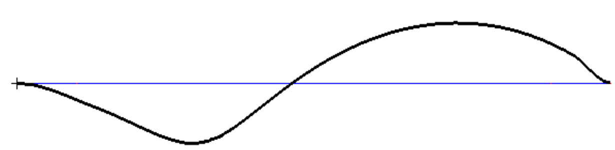


Figure 12. Finite element model of a beam when an additional beam is attached

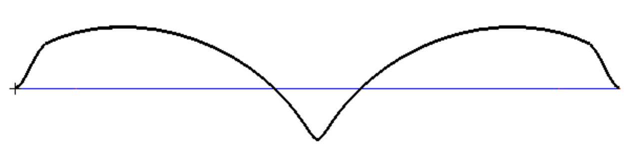
The first three frequencies of transverse mode are 1175.34, 3487.42 and 6370.5 Hz respectively. We see that the primary frequency does not change much even if a large and heavy beam of such is attached. The mode shapes are as follows:



(a) First mode



(b) Second mode



(c) Third mode

Figure 13. First three flexural mode shapes of the beam model when an additional beam is attached

Case 2: A spring-mass system is attached to change the dynamic behavior

In next model, from the principle of vibration neutralizer, a spring mass system is selected such that its frequency becomes equal to that of the beam (and also to the forcing frequency). The mass is taken as 3 kg (zero dimensional lumped mass) and the stiffness of the spring (1 dimensional) 155014.4 N/m.

Modal analysis

The result is very satisfactory in the sense that the first mode is now transferred to 884.61 Hz. The other transverse modes are 1472.8 Hz, 3153.7 Hz, 6205.5 Hz etc. So, the effectiveness of vibration neutralizer is seen using the spring mass system as vibration neutralizer.

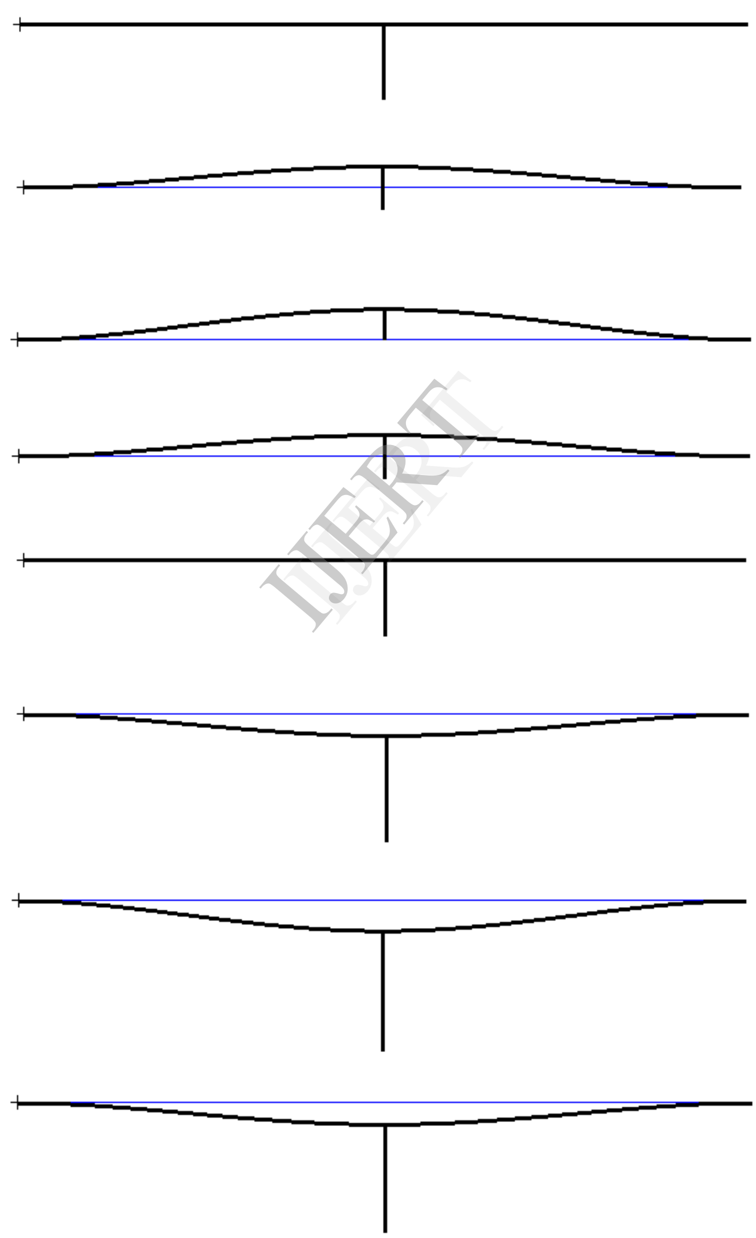


Figure 14. The steps from the animation for first mode shape

The reason for presenting *Figure 14* is to show that the vibration neutralizer is actually having motion as the beam vibrates in first mode.

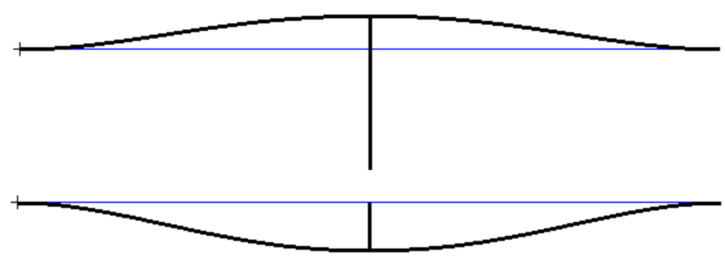


Figure 15. The steps 2 and 5 from the animation for second mode shape

Figure 15 shows that the mode shape in step 2 is similar to first one but deflection of spring and beam both seems higher compared to the first mode. In step 5 the position of mass is on top of the beam. *Figure 16* shows the mode shapes at mode 3 and 4 respectively.

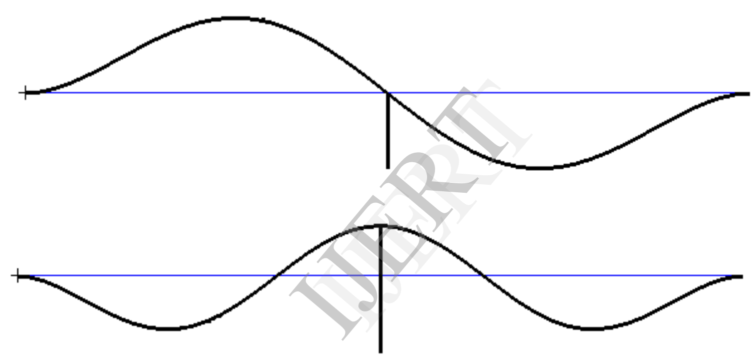


Figure 16. Mode 3 and 4

Direct frequency response analysis

A 50 N harmonic load is applied at node 26 (mid point of the beam). The frequency range is from 0 to 10000 Hz at 100 Hz interval. The amplitudes of the nodes on the beam at all the frequencies in the applied range will be found out by this analysis. No structural damping is used. The response curves for nodes 7, 13 and 26 are shown in *Figures 17-19*. The reason for this analysis is to verify the modal analysis results. It is seen that each other validates.

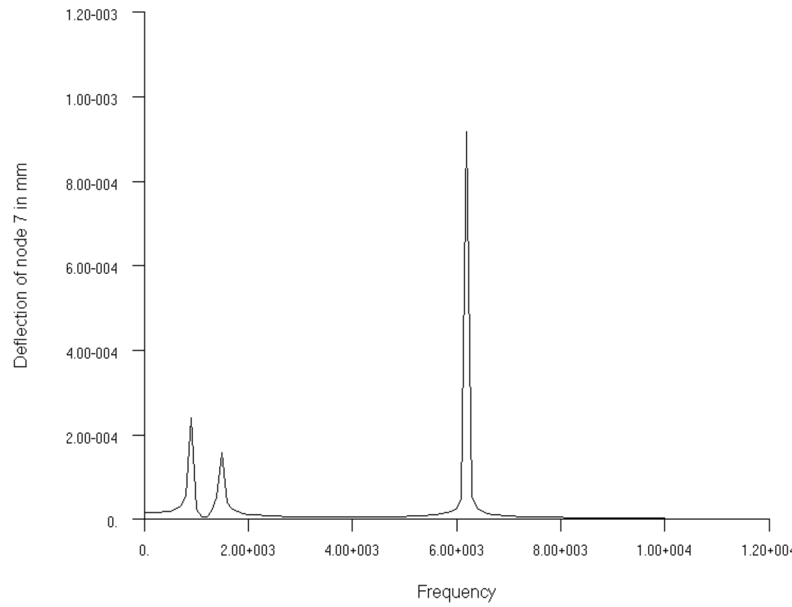


Figure 17. Amplitude of node 7 in mm

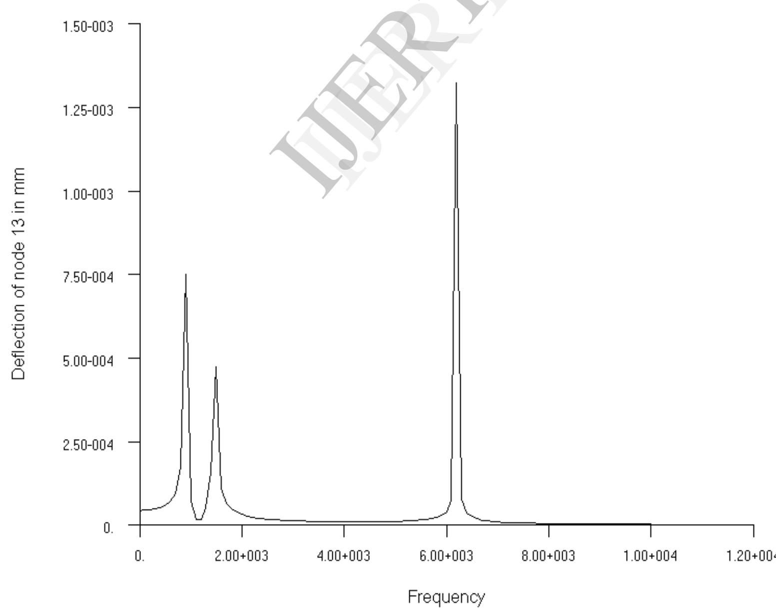


Figure 18. Amplitude of node 13 in mm

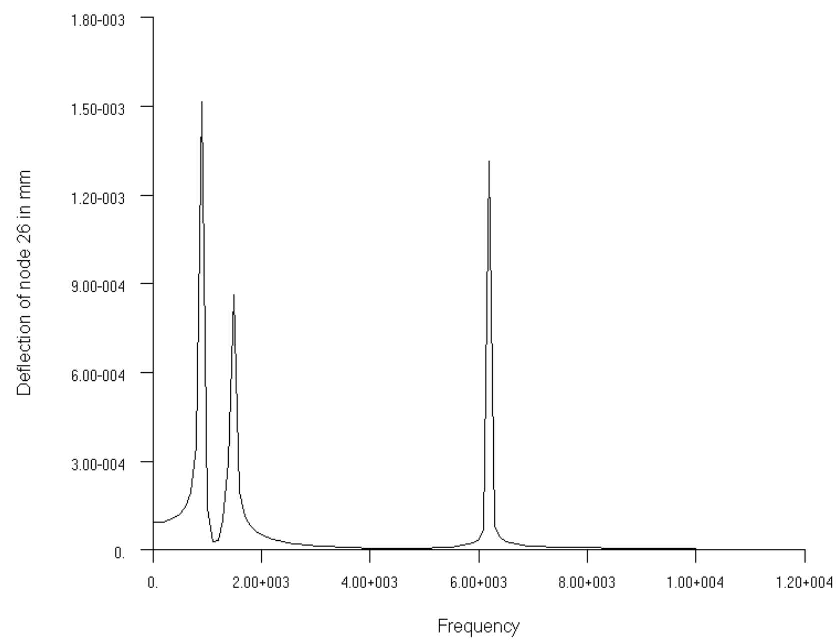


Figure 19. Amplitude of node 26 in mm

Figures 17-19 clearly show that the primary resonance is completely eliminated at 1144.1 Hz but two frequencies (884.61 Hz and 1472.8 Hz) at which resonance will take place is found instead as a result of using vibration neutralizer (spring-mass system). If the operational frequency of exciting force remains same (1144.1 Hz), the other resonance frequencies have no effect on the structure.

CONCLUSIONS

A detailed finite element analysis on the basic dynamic behavior of a fixed ended beam is presented first. The results are then compared with hand calculation using theoretical equations. The closeness of comparison validates the choice of elements and procedure of analysis. Two types of analysis were selected: modal analysis and direct frequency response analysis. By modal analysis the natural frequencies and mode shapes are found. In direct frequency response analysis, an exciting point load at different frequencies of a range of 0 to 10,000 Hz at an interval of 50 and 100 Hz respectively is applied at the mid point of the beam. Three reference points are selected at 1/2, 1/4 and 1/8 length of the beam to see the amplitudes in mm of these points. In this way the effective resonance frequencies are screened out from the rest. The effect of structural damping is also investigated.

In the next analysis, the target was to remove the resonance of the beam from 1144.1 Hz. First an additional beam is attached to the original one having same frequency but different cross sectional area, moment of inertia and length. The result found is not at all effective.

Then a vibration neutralizer consisting of a spring and a mass is attached to the beam at the mid point. The spring and mass was selected such that the frequency of this matches with the frequency of the beam. The modal analysis and the direct frequency response analysis were again carried out. The results again showed good agreement. The first resonant frequency was totally moved from 1144.1 Hz to 884.61 Hz and 1472.8 Hz. Thus the principle of vibration neutralizer was validated.

Though the analysis was done for a beam, the same procedure can be applied to other structures like ship deck (stiffened panel), engine room of a ship etc. which may be subjected to resonant vibration.

One of the leading commercial finite element software named MSC NASTRAN and MSC PATRAN were used to model and analysis of the beam. MSC PATRAN is pre as well as post processor software for modeling and result viewing whereas MSC NASTRAN is a software for the analysis of the mathematical model.

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