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EFFECT OF TUNED ABSORBERS LOCATION ON BEAM STRUCTURE VIBRATION BY FINITE ELEMENT ANALYSIS

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ABSTRACT

Beam structures are common parts in many structural applications. Due to its slenderness, beams are susceptible to lateral vibration motion. Continuous structure such as beam has finite number of vibration modes that make it difficult to control. In this paper, the concept of tuned vibration absorbers applied to a beam structure was analysed by finite element analysis code, ANSYS APDL. The tuned vibration absorbers were attached to the fixed-fixed end beam with four different conditions according to its location of attachment. First, modal analysis has been carried out to determine the natural frequencies and natural mode shapes of the studied beam structure. Then, the effect of tuned absorbers locations on beam vibration characteristics have been studied by harmonic analysis. The vibration characteristics of the beam were discussed with respect to the corresponding tuned absorbers and beam natural modes. It was found that the absorber location affect the beam vibration amplitude greatly. High percentage of vibration amplitude reduction is achieved when the beam amplitude node is avoided. Beam vibration amplitudes were reduced by 99.9% and 99.8% at its 1st and 2nd mode respectively when the absorbers were placed near the fixed end.

Keywords: vibration absorber, beam, finite element.

INTRODUCTION

In engineering field, most vibrations are undesirable because it will give negative effects such as increased in stress, energy losses, induce fatigue, low efficiency and others. Generally, excessive vibration in a system will cause disturbance, discomfort, damage and destruction. The undesirable vibrations are to be reduced to prevent damage on machines or structures. These undesirable vibrations can be reduced by mean of tuned vibration absorber.

A tuned vibration absorber is a relatively small spring-mass oscillator that suppresses the response of a relatively large, primary spring-mass oscillator at a particular frequency. Usually, the mass of the dynamic vibration absorber is a few percent of the mass of the primary mass, but the motion of the dynamic vibration absorber is much greater than the expected motion of the primary mass. The natural frequency of the dynamic vibration absorbers is tuned to be the same as the frequency of excitation. The dynamic vibration absorbers are work effectively when the excitation frequency is close to the natural frequency of the primary mass (Rao, 2011).

A classical DVA consists only a single pair of an auxiliary mass-spring system. This classical DVA only useful for a single degree of freedom system (Zainulabidin and Jaini, 2012), (Zainulabidin and Jaini, 2013) hence limiting its application. Design of vibration absorbers can be tracked back to year 1909. First vibration absorber proposed by Den Hartog consists of a second mass-spring device attached to the main device, also modelled as a mass-spring system, which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device. This classical problem of vibration has a well-known solution. If damping is added to the absorber, the vibration amplitude of the main mass cannot be made zero at the forcing frequency but the sensitivity of the system to variations in the forcing frequency decreases (Den Hartog, 1956) . Bonsel *et al.* used damped and undamped DVA to suppress first resonance of a piecewise linear system (Bonsel *et al.*, 2004). Wu studied the inertia effect of helical spring of the absorber on suppressing the dynamic responses of a beam subjected to a moving load (Wu, 2006). Vibration absorber for beam structures with three different types of end condition has been studied by Foda and Albassam. The end conditions are simply support, free and clamped end (Foda and Albassam, 2006). Wong *et al*. studied a DVA combining translational and rotational type absorber for beam vibration under point or distributed harmonic excitation (Wong at al., 2007). Kojima and Saito utilized non-linear dynamic vibration absorber to confine the forced vibration of a beam has been studied (Kojima and Saito, 1983).

In this paper, a new control strategy has been tested by finite element simulation in order to absorb vibration of a beam structure. Only the first two natural modes were studied and the beam structure is made to vibrate in transverse direction only. The vibration absorbing device is attached at 4 different conditions to optimally reduce the vibration level a fixed-fixed end beam. This special DVA can be used to control the vibration level of a building built in earthquake prone area, to control the vibration level of a bridge exposed to high speed or turbulence wind and to control airplanes wing flutter.

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METHODOLOGY

The studied systems

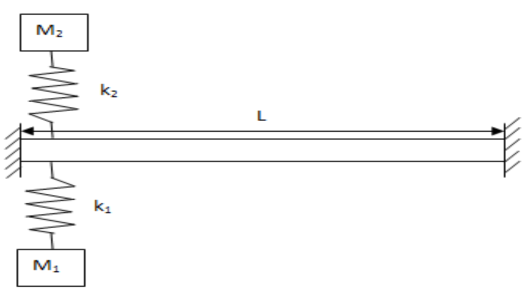
The total length, L of the beam is 0.8 m and it is fixed at both of its end. The beam modulus of elasticity, density and mass moment are 69 GPa, 2800 kgm^{-3} and 1.67×10^{-11} kgm^2 respectively. Both masses, M_1 and M_2 are 0.1 kg in weight. The spring stiffness k_1 and k_2 are adjusted so that the value of $\sqrt{k_1/M_1}$ and $\sqrt{k_2/M_2}$ are equal to the beam 1st and 2nd natural frequency respectively.

Four absorbers arrangement conditions were studied. These conditions are purposely arranged to study the effect of tuned absorbers location on vibration behaviour of the beam structure.

These conditions are:

i) First condition - Both DVAs at side
ii) Second condition - First DVA at side, second DVA at beam centre.
iii) Third condition - Both DVAs at beam centre
iv) Fourth - First DVA at centre, second DVA at side.

The tuned absorber at side is arranged so that it will be placed in between fixed-end and antinode in the first and second beam mode. It is also intended to minimise external weight effect on the beam structure. The tuned absorber at the centre is arranged so that it will be placed on the antinode (maximum magnitude) of the first beam mode and on the node (zero magnitude) of the second beam mode.



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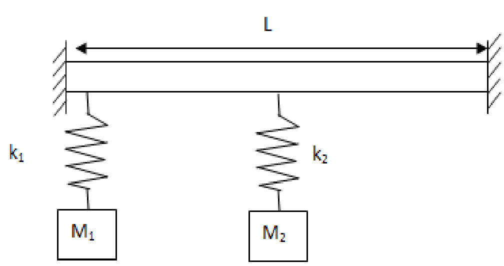

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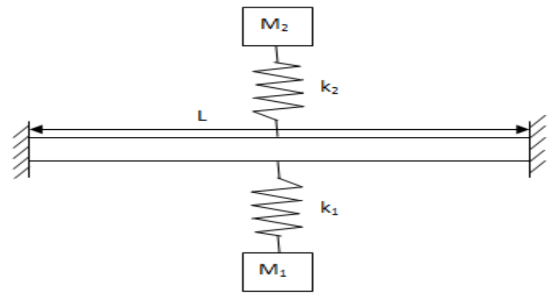

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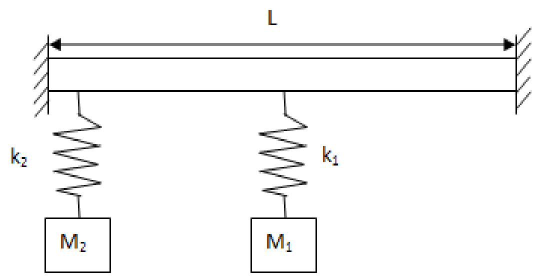

(d)

Figure-1. (a)-(d) Schematic of the 1st, 2nd, 3rd and 4th conditions respectively.

Finite element model

The finite element model consist of 3 main components; the beam structure, the first absorber which was tuned the beam 1st natural frequency and the second absorber which was tuned to the beam 2nd natural frequency.

The beam structure was modelled by BEAM3 element. BEAM3 is a 2D elastic, uniaxial element with tension, compression, and bending capabilities. The element has three degrees of freedom at each node; translations in the nodal x and y directions and rotation about the nodal z-axis (ANSYS 14.0, 2011). The beam was divided into twenty equal size elements. There are 21 nodes on the beam and the distance between each node is 0.04m.

The two tuned absorbers consist of 2 parts which are the spring and the auxiliary mass. The spring part was modelled by COMBIN14 element. COMBIN14 is a spring-damper element. It has longitudinal or torsional capability in 1D, 2D or 3D applications. The longitudinal spring-damper option is a uniaxial tension-compression element with up to three degrees of freedom at each node; translations in the nodal x, y, and z directions. No bending or torsional motion is considered (ANSYS 14.0, 2011). In this study, the longitudinal capability was used and the damping capability had been removed from the element. COMBIN14 element has no mass. Masses can be added by using the appropriate mass element. The auxiliary mass was modelled by MASS21 element. It is a point element having up to six degrees of freedom; translations in the

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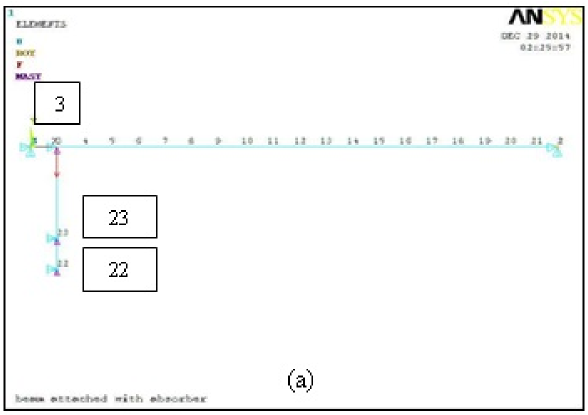
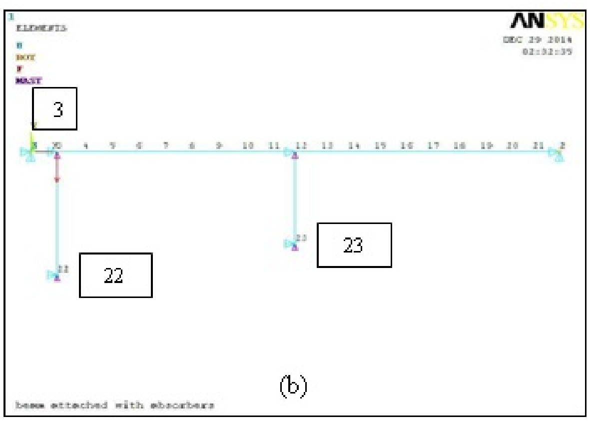
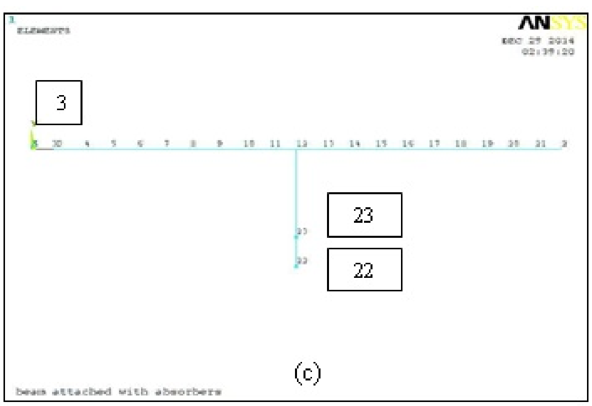
nodal x, y, and z directions and rotations about the nodal x, y, and z axes (ANSYS 14.0, 2011). The auxiliary masses were constrained such that it will only move in vertical direction.

The displacement values of the beam, the absorber tuned to the 1st beam mode and the absorber tuned to the 2nd beam mode were taken at point 3, 22 and 23 in Figure-2 respectively. The beam displacements were taken at point 3 because at the 1st and 2nd beam mode, it is located in between node and antinode of beam vibration amplitude.







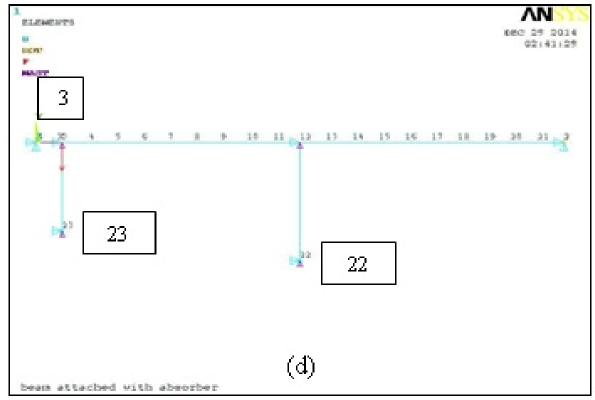


Figure-2. (a)-(d): Finite element model of the 1st, 2nd, 3rd and 4th conditions respectively.

RESULTS & DISCUSSION

Modal analysis

Modal analysis was carried out in order to the determine the natural frequencies of the beam structure without and with influences of auxiliary/absorber masses. The natural frequencies of the auxiliary masses will be tuned to that of beam without mass. The natural frequencies of the beam structure with mass are useful in explaining the vibration mode of the beam in Harmonic analysis in the next sub-section. The natural frequencies of the beam structure with different auxiliary mass combinations are tabulated in Table-1.

Modal analysis was also carried out to identify the beam natural mode of vibration without and with influence of auxiliary masses. Figure-3 (a)-(b) show the natural vibration mode of the studied beam structure. It was found that the auxiliary masses has significant effect on the beam first two natural vibration modes. The natural vibration mode of beam structure can be described based on its vibration amplitude. The maximum and zero amplitude is depicted as antinode and node respectively.

Table-1. Natural frequencies of beam with different auxiliary mass combinations.

Beam Descriptions	Nat. Frequency (Hz)	
	1st mode	2nd mode
Wihout mass	15.948	43.961
1st Condition	15.344	16.184
2nd Condition	8.6481	15.596
3rd Condition	6.4315	20.412
4th Condition	7.8848	30.432

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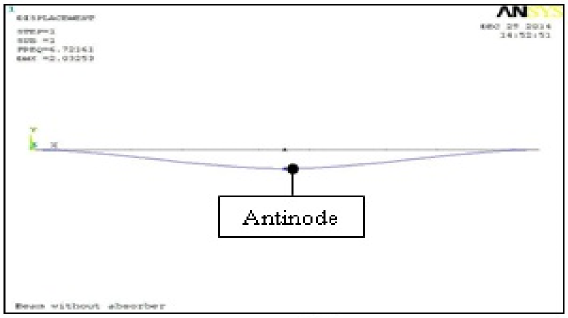

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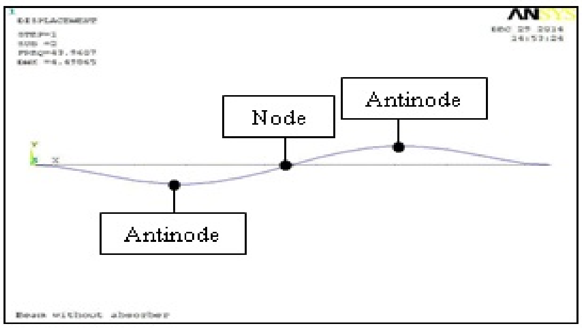


(b)

Figure-3. (a)-(b): Beam vibration mode of the 1st and 2nd mode respectively.

Harmonic analysis

Harmonic analysis was carried out to study the effect of vibration absorbers location on beam vibration characteristics. It is interesting to find out which condition will produce optimum vibration absorption. It is also intriguing to understand the characteristics of tuned absorber masses at each condition. There are three criterias used in describing the characteristics of the beam-absorber system. The critirea are as follow:

i) If the beam amplitude is reduced and the tuned absorber shows significant peak amplitude at the same frequency, it indicates 'vibration absorption' by the absorber.

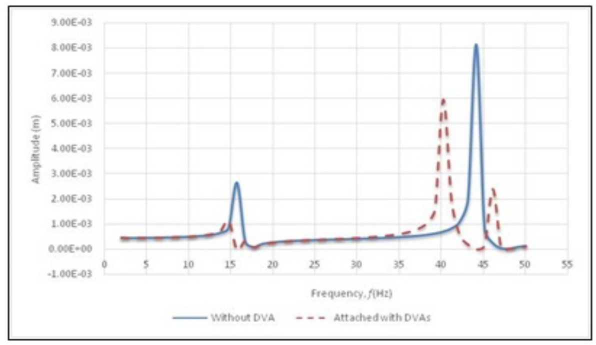
ii) If the beam amplitude is reduced but the tuned absorber does not shows any peak amplitude at the same frequency, it indicates 'no vibration absorption' by the absorber.

iii) If the tuned absorber shows significant amplitude at any frequency but no significant peak amplitude by the beam, it indicates the absorber vibrates at its own natural mode.

The results of harmonic analysis for the 1st, 2nd, 3rd and 4th conditions are presented in Figure-4 (a)-(c), Figure-5 (a)-(c), Figure-6 (a)-(c) and Figure-7 (a)-(c) respectively.

Results of the 1st condition

Figure-4 (a)-(c) show the harmonic analysis results for the 1st condition. In the 1st condition, both absorbers were placed at side of the beam which is neither on node or antinode of the beam amplitude. The vibration amplitude at the 1st and 2nd mode of beam vibrations are greatly reduced by both absorbers. Peak amplitude at ~16 Hz in Figure-4 indicates the vibration were 'absorbed' by the tuned absorber. However, the beam amplitude for the 2nd mode has not been 'absorbed' by the tuned absorber. The tuned absorber only vibrates with the beam at the two new peaks.



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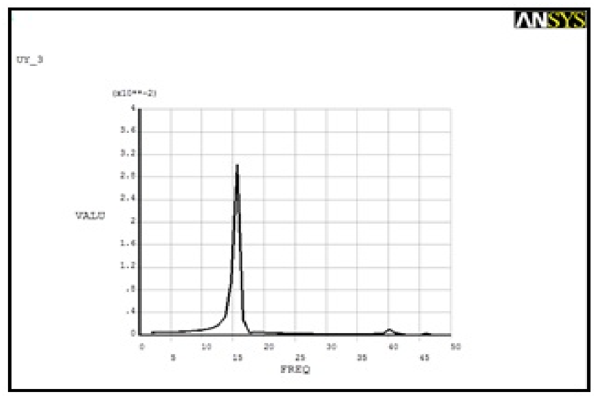

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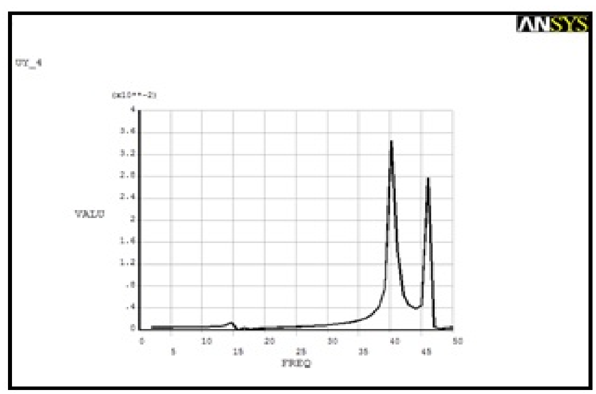

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Figure-4. Harmonic analysis results for the 1st condition **(a)** Vibration response of the beam before and after tuned absorbers attachment; **(b)** Vibration reponse of the absorber tuned to the 1st beam mode; **(c)** Vibration response of the absorber tuned to the 2nd beam mode.

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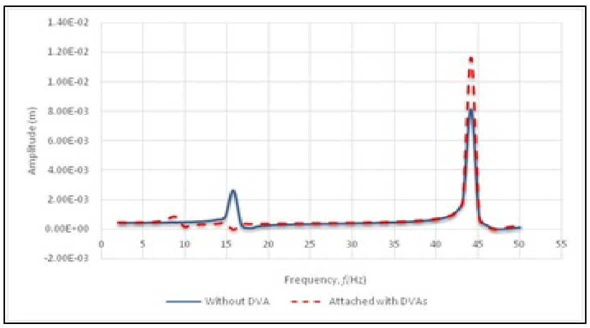
Results of the 2nd condition

Figure-5 (a)-(c) show the harmonic analysis results for the 2nd condition. In the 2nd condition, the absorber tuned to the beam 1st mode is placed at side while the absorber tuned to the beam 2nd mode is placed in the middle of the beam. The absorber tuned to the beam 1st mode successfully reduced the beam vibration amplitude. This characteristic is in good agreement with the absorber amplitude in Figure-5(b). However, the absorber tuned to the beam 2nd mode did not reduce the beam vibration amplitude. The absorber mass only vibrate at frequency of 8.6 Hz which corresponds to the 1st natural frequency of beam with 2nd condition presented Table-1.



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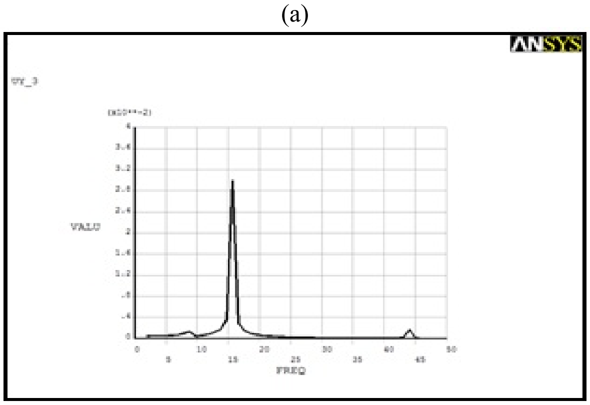

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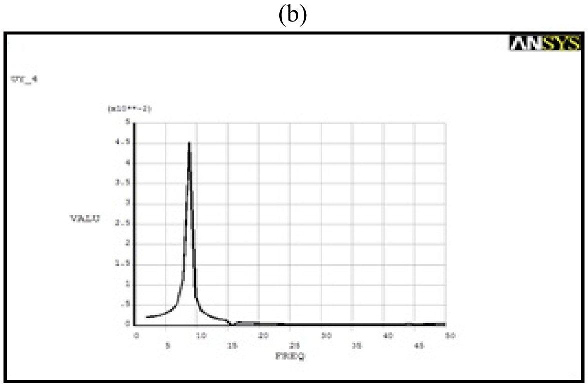

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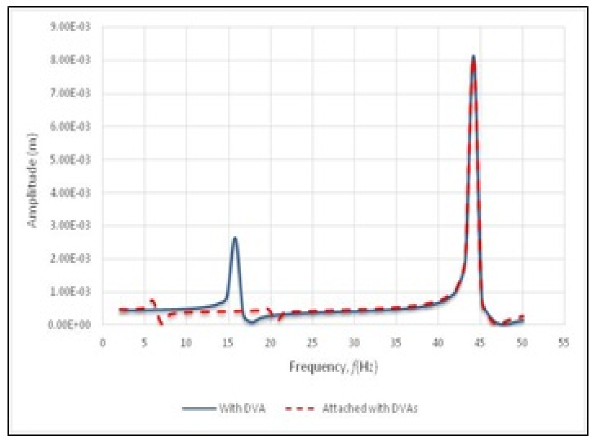
Figure-5. Harmonic analysis results for the 2nd condition **(a)** Vibration response of the beam before and after tuned absorbers attachment**; (b)** Vibration reponse of the absorber tuned to the 1st beam mode; **(c)** Vibration response of the absorber tuned to the 2nd beam mode.

Results of the 3rd condition

Figure-6 (a)-(c) show the harmonic analysis results for the 3rd condition. In the 3rd condition, both absorbers are placed at the middle of the beam. The beam 1st mode amplitude has successfully been reduced. However, this reduction has not been reduced by the absorber tuned to the beam 1st mode. There is no peak amplitude at ~15 Hz in Figure-6 (b) that would indicate the absorber has 'absorbed' the beam vibration. The beam amplitude at the 2nd mode fail to be reduced. The absorber tuned to the 2nd beam mode is placed on the amplitude node, hence its role as vibration absorber at the 2nd beam mode is nulified. Both absorbers only vibrate at the 1st (6.4 Hz) and 2nd (20.4 Hz) natural mode of beam with 3rd condition as presented in Table-1.



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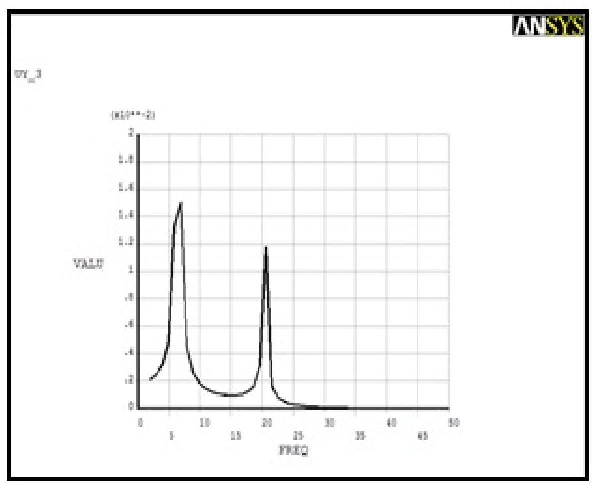

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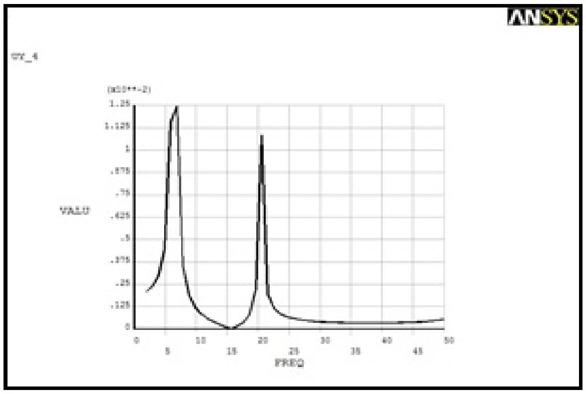
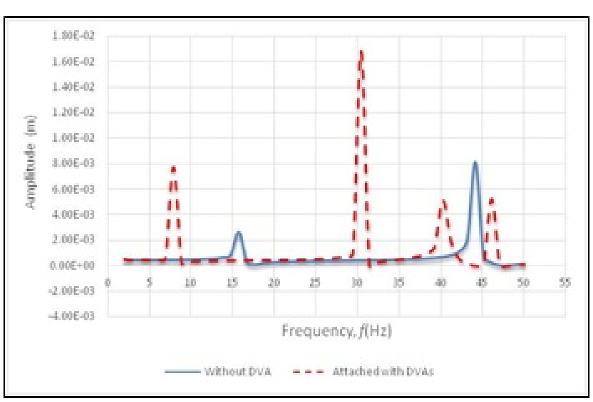


(c)

Figure-6. Harmonic analysis results for the 3rd condition **(a)** Vibration response of the beam before and after tuned absorbers attachment; **(b)** Vibration reponse of the absorber tuned to the 1st beam mode; **(c)** Vibration response of the absorber tuned to the 2nd beam mode.

Results of the 4th condition

Figure-7 (a)-(c) show the harmonic analysis results for the 4th condition. In the 4th condition, the absorber tuned to the 1st beam mode is placed at the middle and the absorber tuned to the 2nd beam mode is placed at side. The beam 1st mode amplitude has successfully been reduced. However, similar to the case in the 3rd condition, this reduction has not been reduced by the absorber tuned to the beam 1st mode. There is no peak amplitude at ~15 Hz in Figure-7(b) that would indicate the absorber has 'absorbed' the beam vibration. The tuned absorber only vibrate at the 1st (7.9 Hz) and 2nd (30.4 Hz) natural mode of beam with 4th condition as presented in Table-1. Similarly, the beam amplitude has been successfully be reduced for the 2nd mode. However, the beam amplitude for the 2nd mode has not been 'absorbed' by the tuned absorber. The tuned absorber only vibrates with the beam at the two new peaks. Peak amplitude can be observed at frequency ~40 and ~46 Hz in Figure-7(c). The tuned absorber also vibrate at the 1st (7.9 Hz) and 2nd (30.4 Hz) natural mode of beam with 4th condition as presented in Table-1.



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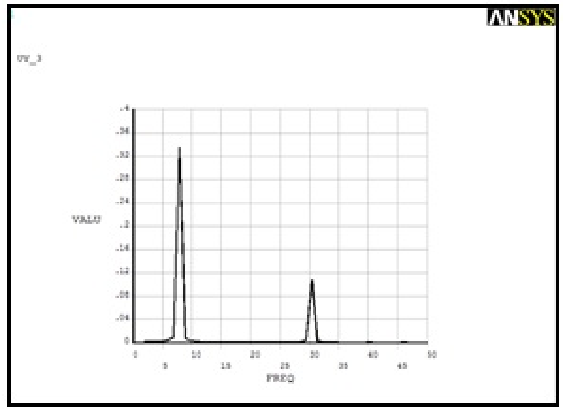

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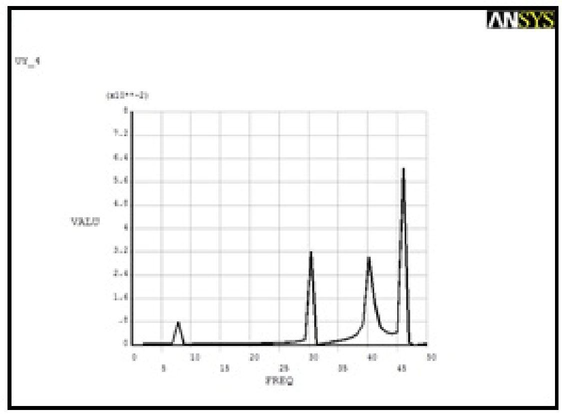

(c)

Figure-7. Harmonic analysis results for the 4th condition **(a)** Vibration response of the beam before and after tuned absorbers attachment; **(b)** Vibration reponse of the absorber tuned to the 1st beam mode; **(c)** Vibration response of the absorber tuned to the 2nd beam mode.

General performance analysis

Table-2 shows the comparison of absorber performance for all four conditions. The 1st condition produced the best performance among all the four conditions followed by the 4th condition. In the 1st and 4th conditions, neither of the tuned absorber was placed on the amplitude node. In the 2nd and 3rd condition, the absorber tuned to the beam 2nd mode was placed on the node of the beam 2nd mode. Since at node the vibration amplitude is zero, its role as an absorber was nullified, hence resulting in poor absorption performance.

The observations of the 1st beam mode amplitude reduction in the 3rd and 4th conditions imply that, the reduction that occur for the 1st mode in both conditions are not the work of the tuned absorbers. Even though, the absorbers in both cases were tuned to the 1st natural mode of the beam and were placed on the antinode of the beam amplitude, the absorbers did not absorb the vibration.

Every structure has its own natural vibration mode. When auxiliary masses were added to the structure, the structure and the auxiliary masses become a new

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multi-structure. This new multi-structure has new natural vibration modes. When the natural frequency of the auxiliary mass coincide with the natural frequency of the original structure inside the multi-structure, the initial amplitude of the original structure seems to be 'absorbed' by the auxiliary mass. At other frequency values, the structure and the auxiliary mass vibrate as multi-structure with its own natural modes. At certain frequency the amplitude of the new multi-structure might be less than the original structure due indirect influence of the auxiliary masses i.e. change in total structural mass.

Table-2. Percentage of amplitude reduction.

Description	First Mode		Second Mode	
	Amplitude (m)	Reduction (%)	Amplitude (m)	Reduction (%)
Without DVA	$2.67x10^{-03}$	ref	$8.16x10^{-03}$	ref
1st Condition	$3.21x10^{-06}$	99.88	$1.40x10^{-05}$	99.83
2nd Condition	$3.19x10^{-06}$	99.88	$1.17x10^{-02}$	-43.12
3rd Condition	$4.28x10^{-04}$	83.96	$8.10x10^{-03}$	0.76
4th Condition	$4.35x10^{-04}$	83.69	$1.40x10^{-05}$	99.83

CONCLUSIONS

The effect of tuned vibration absorbers location on the beam structure has been studied by finite element analysis. The results show that the absorber location affect the beam vibration amplitude greatly. High percentage of vibration amplitude reduction is achieved when the node is avoided. At node, the role of the auxiliary mass absorber seems to be nullified.

Addition of auxiliary mass on structure will change the natural frequencies and vibration response of the structure. When the natural frequency of the auxiliary mass coincide with the natural frequency of the structure, the initial amplitude of the structure seems to be 'absorbed' by the auxiliary mass. At other frequency values, the structure and the auxiliary mass vibrate as one single multi-structure.

Knowledge from this study is vital in designing a vibration suppression to control the vibration level of a building built in earthquake prone area, to control the vibration level of a bridge exposed to high speed or turbulence wind and to control airplanes wing flutter.

In the future work, the researcher should try to analyse the vibration characteristics of the beam-absorber with more number of modes i.e. 3 or 4 modes.

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