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EFFECT OF FLAP POSITION ON TRANSVERSE VIBRATION OF RECTANGULAR PLATE WITH CUTOUT

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Abstract

The present work deals with the analysis of transverse vibration of a clamped rectangular plate with cutout using vibratory flap subjected to harmonic excitation. Vibratory flap is an auxiliary plate, which can vibrate independently when it is attached on the main plate as a cantilever plate. The vibratory flap acts as a dynamic vibration absorber which reduces the vibration of the plate at tuning and higher frequencies. This study attempts to discover the effect of flap position on transverse vibration of rectangular plate with cut-out. Analysis has been carried out for different combinations of flap position and cutout locations through finite element analysis using ANSYS software. From the analysis it was found that the flap position plays a significant role in vibration control of plate. The vibration of the plate was reduced by 83% and 75% respectively in first and second frequencies in the presence of cutout and flap when compared with the uncut plate. This shows improved dynamic response of the plate in the presence of flap.

Keywords: Transverse vibration, Rectangular plate, Vibratory flap, Flap position, Cutout location.

1. Introduction

Cutouts are inevitable in structures. Cutouts are widely used in plates as ventilation openings to facilitate heat dissipation, to meet the demands for reduced weight, passage of ducts, conduits, cable and to provide access to different components. The designer must or should know the effect of these perturbations upon the dynamic characteristics of the structural element. Because of the presence of cut- outs, the structure will weaken dynamically due to an external excitation and experience amplification in its response as compared to those of the case without a cutout. Presence of cutout and its location has a significant effect on the dynamic behaviour of the plate. The undesirable vibrations may cause sudden failures due to resonance in the presence of cutouts. It is, therefore, important to predict the natural frequencies and mode shapes due to geometrical alternations in the design.

A similar concept has been used in the analysis of plate with cutout by using an additional plate called flap, which can vibrate independently, when it is attached as a cantilever plate on the main plate. The size and shape (square) of the flap is taken as same as that of cutout of the plate. The optimally placed vibratory flap reduces the vibration of the plate with cutout at tuning and higher frequencies. Automotive panels are considered to be such cases where flaps were incorporated inside a panel and it would not interfere with design functions and aesthetics.

Several investigators have examined the use of dynamic vibration absorber technique for vibration control of plate with or without cutouts and a few on vibration control of plate-by-plate type dynamic vibration absorber. The effect of mass ratio and location of the point mass analysed [1] through the method of superposition to obtain an analytical solution for free and forced vibrations of square and rectangular cantilever plates carrying point masses.

Amabili et al. [2] described experimentally the linear vibration of rectangular plate carrying concentrated masses. Jacquot [3] presented a Numerical method to predict the effectiveness of application of damped dynamic vibration absorber to suppress stationary random vibration of rectangular simply supported plate. Arpaci and Savci [4] developed the concept of use of a cantilever beam in suppressing excessive resonance amplitudes of rectangular cantilever plate by proper use of tuned cantilever beam dampers.

Ozguven, and Çandir [5] employed a procedure to determine the optimum parameters of two dynamic vibration absorbers for the first two resonance of a cantilever beam. Curadelli et al. [6] reported the numerical and experimental study to determine the efficiency of the attached masses of the vibration control system. A simple mass spring damper vibration absorber is employed [7] to suppress the nonlinear vibrations of the forced nonlinear oscillator for the primary resonance conditions and shown the effectiveness of the dynamic vibration absorber for suppressing primary resonance vibrations.

McMillan and Keane [8] considered the vibration of a simply supported rectangular thin plate carrying a number of concentrated masses and shown that the addition of the masses has the effect of reducing the Eigen values within certain bounds. Avalos and Laura [9] discussed analytically the transverse vibration of simply supported rectangular plate with two rectangular holes. In the plate the cut-outs were chosen to be the same aspect ratio as original plate

Ranjan and Gosh [10] worked on forced vibration response of a rectangular thin plate with single discrete mass and patch act as dynamic vibration absorbers over certain frequency range has been determined using finite element method. Aida et al. [11] proposed a numerical technique for controlling the several predominant modes of vibration of plate using a new plate-type dynamic vibration absorber connecting through springs and dampers using the optimum tuning method. Kerlin [12] investigated theoretically the vibration response of clamped circular plate with plate-like dynamic vibration absorber to suppress effectively the first resonance and provides significant attenuation simultaneously at more than one frequency.

Ulz and Semercigil [13] introduced the use of cutouts to improve the performance of the plate by creating the flaps as incision and also they have reported the use of auxiliary flap placed at an assumed inclination of 30º at a location of one fourth of its total length and obtained attenuations about 70% between the attachment of absorber and the nearest boundary. Mahadevaswamy and Suresh investigated the effect of position and tilt angle of the flap on vibration control of rectangular plate using auxiliary flap by the finite element analysis [14] and by experiments [15] and reported that better attenuation of vibration could be obtained at the tilt angle of 45°.

The objective of this investigation is to predict the effect of flap position on vibration control of a clamped rectangular isotropic plate with cutout when subjected to harmonic excitation using finite element method. In the analysis a rectangular plate with square cut-out and flap is used for different combinations of flap position and cutout location is considered. The size of the square flap is same as that of cutout on the plate to maintain constant mass among the uncut plate and cutout plate with flap. In the analysis the two different configurations used are a plate with cutout at a distance 25% and 50% of the length of plate with flap placed different position.

2. Finite Element Analysis

The analysis has been done on the clamped rectangular plate with cutout and flap placed at different locations. All the four edges of the rectangular plate are completely fixed in all degrees of freedom (UX, UY, UZ and ROTX, ROTY ROTZ) as shown in Fig.1. The size of the square flap is taken same as that of cutout of the plate to maintain constant mass among the uncut plate and plate with cutout and flap.

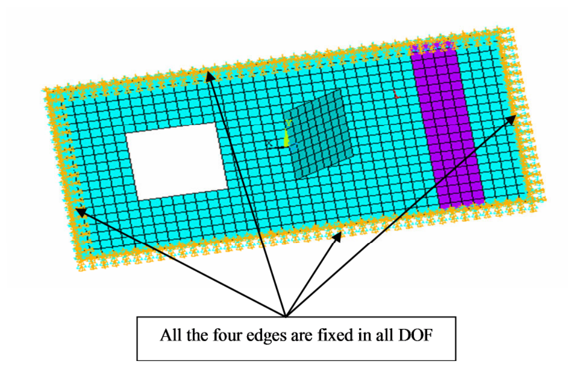


Fig. 1. Boundary Conditions of FE Model of the Plate with Cutout and Flap.

The flap is fixed as a cantilever plate at tilt angle of 45° at different locations of the plate with fixed edge of the flap perpendicular to length of the plate. Justification for using tilt angle 45° is that the vibration attenuation is more tilt angle 45° and it is evident from the numerical (14) and experimental (15) investigation of effect of tilt angle on vibration control of plate reported by the same authors.

The flap is fixed on the bare plate using welded joint as a cantilever plate. In structural applications bolted and riveted joints are used to introduce damping [16], but it reduces stiffness of the structure, by producing debris due to joint slip and cause fretting corrosion. Stiffness is one of the parameter in addition to mass, which alter the natural frequency of a system. This is the reason for selecting welded joint between plate and flap.

The position of the flap is symmetrical with respect to centre line along the length of the plate. The size of the rectangular plate is 400 mm x 200 mm with aspect ratio (length to width) 2 and that of flap is 80mm x 80mm (40% of smaller side of the plate) with uniform thickness 3mm. The plate is modelled and meshed with shell 63 element using ANSYS and total of 800 elements were sufficiently fine enough to achieve good results as per convergence criterion. The plate was meshed separately with solid185 elements and analysis was done. The results showed that the change of displacement differed by less than 2% .This shows that shell 63 and solid 185 both gives the almost same results. Shell63 has been used in the analysis as the element has both bending and membrane capabilities and has six degrees of freedom at each node. The material for both plate and flap is steel having modulus of elasticity 200 GPa, Poisson's ratio 0.3 and mass density of 7800 kg/m^3.

For the purpose of avoiding infinite response at the resonant frequencies, the constant damping ratio of 0.02 is used in harmonic analysis. The present objective is to determine the conditions under which tuning is established between the flap and the main plate to be controlled. Including actual structural damping is meaningful only after establishing the tuning condition to dissipate the harmful energy is transferred to the flap as a result of strong interaction.

Analysis has been done in two cases over the plate for different flap position with cutout. In the first case cutout is made at 25% of the length (SC-Side cutout) and in the second case cutout is at 50% (CC-Centre cutout) of the length of the plate. The cutout at 75% of length of the plate is symmetrical with 25% length of the plate, hence neglected in the analysis. For the first case the flap is attached at five different locations I – V shown in Fig. 2(b) and in the second case the flap is attached at three locations I-III shown in Fig. 2(c). At each location, two positions of the flap inclined at a tilt angle of 45º towards left (Case A) and right (Case B) is considered.

Further Case (1A) refers to plate with side cutout and flap placed at location I and bent towards left. Case (1B) refers to plate with side cutout and flap placed at location I and bent towards right. The remaining cases may be defined in the similar way.

For case, displacement histories are obtained at 9 locations (1-9) along the centre line over the plate. The displacement on either side of centre line is low as compared to displacement along centre line of the plate. The schematic drawing of the plate with and without cutout showing different observation points and flap locations is shown Fig. 2.

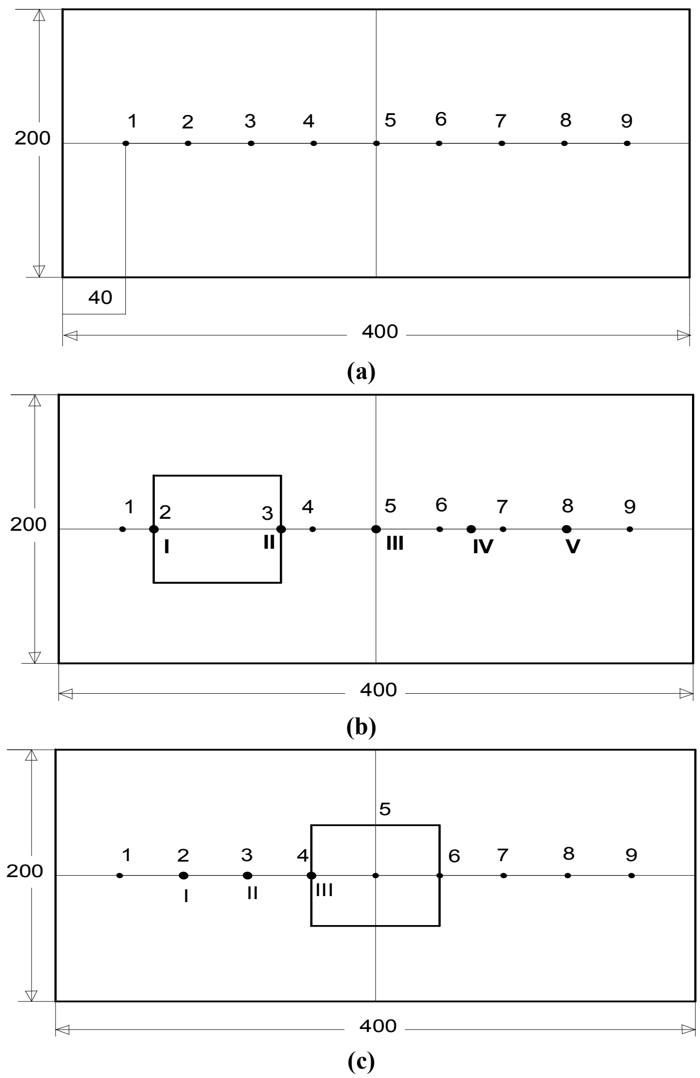


Fig. 2. Schematic Diagram of the Plate Showing (a) Uncut Plate (b) Plate with Side Cutout (c) Plate with Centre Cutout: 1-9 Displacement Observation Points and I-V Flap Locations.

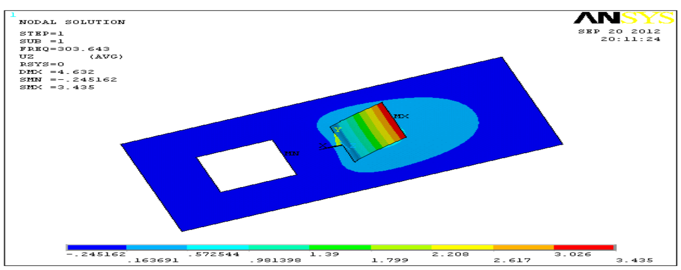
2.1. Modal analysis

The modal analysis of clamped rectangular isotropic plate with cut-out and flap has been performed to get the mode shape and natural frequencies. The mode shape and natural frequencies are obtained for different configuration of the plate. Natural frequencies of the plate at different modes with different configurations are tabulated in Table 1. It is well known that for uncut plate, the first mode is in-phase mode with largest deflection in the mid-span. The second mode is the asymmetric mode where the left and right sides move out-of-phase with largest deflections around a quarter of the total length. Placing the absorber at locations 25% to 75% of length seems to produce the appreciable attenuations, whereas moving closer to the fixed wall seems to reduce this effect.

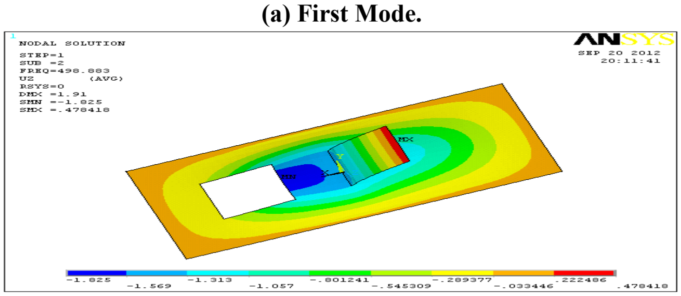
Table 1. Natural Frequency of the Plates with Side Cutout and Flap.

Configuration	Natural frequency (Hz)				
	Mode1	Mode 2	Mode 3	Mode 4	Mode 5
Uncut	449.2	581.2	817.2	1155.7	1168.7
SC	458.0	631.4	933.7	1108.5	1110.7
Case (1A)	312.9	458.1	624.4	789.6	839.6
Case (1B)	263.4	459.8	634.7	740.2	842.6
Case (2A)	195.8	412.2	431.2	580.9	709.4
Case (2B)	230.0	393.3	449.8	557.5	686.2
Case (3A)	301.4	498.5	679.5	826.0	916.0
Case (3B)	303.6	498.9	651.9	932.2	925.6
Case (4A)	304.3	504.6	626.5	834.9	959.1
Case (4B)	309.4	478.5	656.4	836.1	954.1
Case (5A)	304.8	487.9	642.9	841.3	890.6
Case (5B)	327.0	466.4	617.5	853.9	897.8

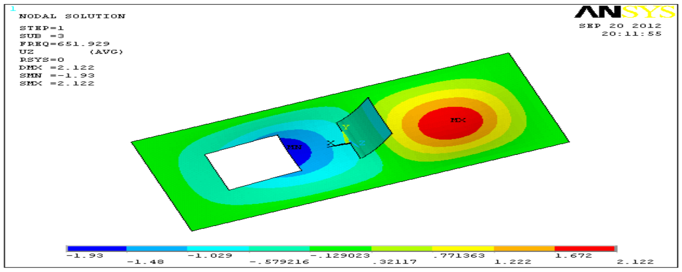
The high modal displacement of the plate with side cutout lies at the centre of the plate and around the cutout. It is necessary to reduce the vibration at these locations to achieve attenuation. The first and second natural frequencies of the uncut plate are 449.2 Hz and 581.2 Hz respectively; these are the target frequencies at which attenuation in vibration was observed. The fundamental frequency of the flap is tuned nearly to the fundamental natural frequency of the uncut plate. Figure 3 shows the different mode shapes of the clamped rectangular plate with cutout and flap for the best combination of the flap and cutout locations.



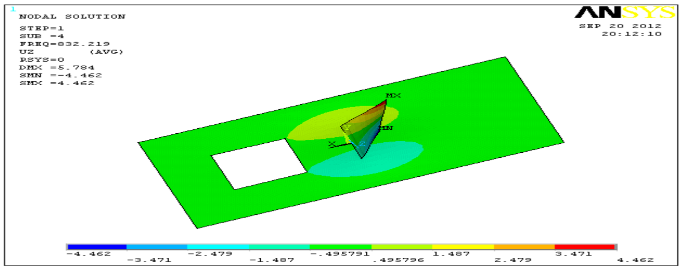
(a) First Mode.



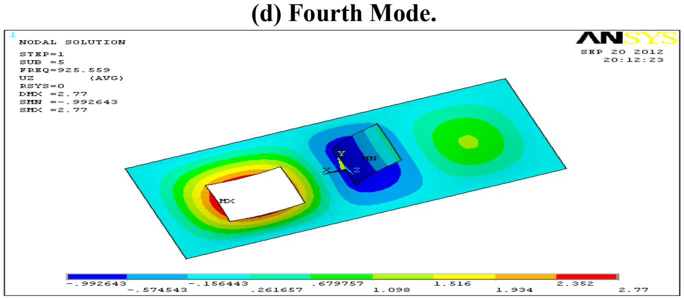
(b) Second Mode.



(c) Third Mode.



(d) Fourth Mode.



(e) Fifth Mode.

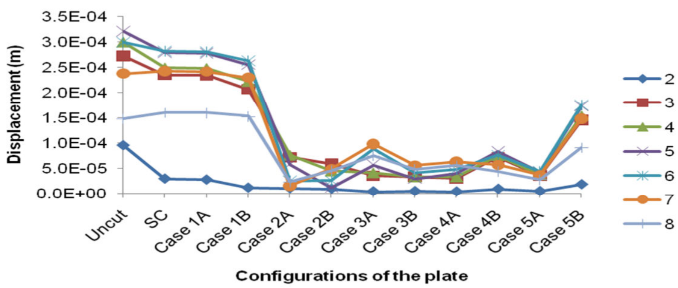
Fig. 3. Different Mode Shapes of the Clamped Rectangular Plate with Side Cutout and Central Flap (Case 3B).

2.2. Harmonic analysis

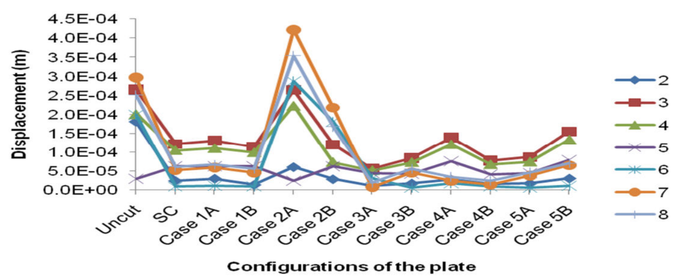
Harmonic analysis has been done to obtain the dynamic response of a plate at several frequencies. It has been carried out for the uncut plate and plates with flap and cutout. The frequency of the flap is tuned nearly to the fundamental frequency of the plate (target frequency). A transverse excitation force of 100 N with the frequency varies from 0 to 1200 Hz is applied at a location 75% of length and 75% of width. The force is applied away from the centre of the plate to avoid nodal lines. The frequency response was obtained from the harmonic analysis of plate with different configurations at different locations of the plate. The response (displacement) was observed at 9 different locations (1-9) along the centre line over the plate and these points are 40 mm (10% of length) apart.

3. Results and Discussion

The frequencies of the plate with cutout and flap at different locations were found from the modal analysis and were used to find the effect of location of flap on first and second frequency of the plate. From the harmonic analysis the frequency response curves were obtained for all the configurations of plate with side and central cutouts. From these curves the displacements over the plate were observed at first and second target frequencies and were used to plot the graph of different configuration verses displacement shown in Figs. 4 and 5 respectively for side and central cutout. Numerical studies shows that the cases with a side cutout with flap have the better potential compare to central cutout with flap. Hence, discussion will be limited to this particular case, for conciseness.



(a) First Target Frequency at 450 Hz.

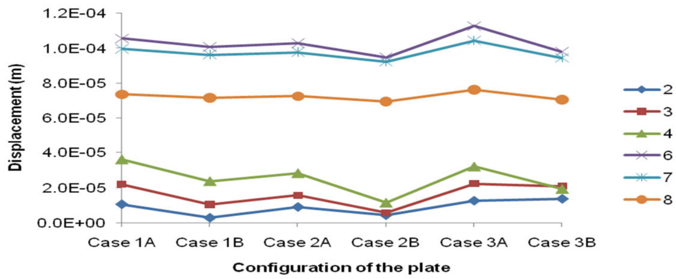


(b) Second Target Frequency 580 Hz.

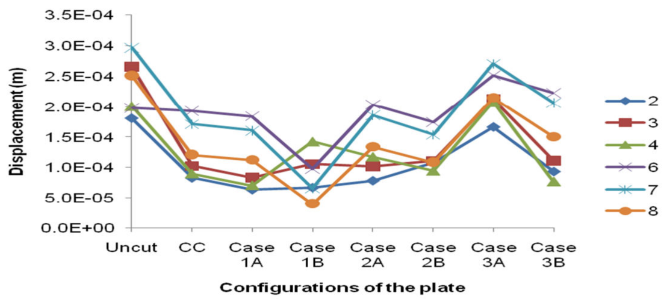
Fig. 4. Displacement of the Plate with Different Cases of Side Cutout.

The natural frequencies for plate with side cutout and various flap configurations are given in Table 1. This shows the natural frequency varies with location of the flap and also there will be slight variation in when flap bent towards or away from the cutout. It was observed from the table that the first frequency of the cutout plate with flap for the cases 1A, 1B, 2A, 2B and 5B are very nearer to the target frequency of the uncut plate. From the modal analysis it is concluded that placing flap near cutout and near fixed boundary seems to be quite ineffective.

Figure 5 shows the displacement of 10 cases with the side cutout and flap. It was observed that for the case 1A, 1B, 5B first frequency response was more and in case 2A, 2B, 5B the second frequency response was more as compared to remaining cases. This shows that placing flap near cutout and fixed boundary were quite ineffective. From modal and harmonic analysis it was concluded that for the cases 3A, 3B, 4A, 4B were more favourable.



(a) First Target Frequency at 450 Hz.



(b) Second Target Frequency 580 Hz.

Fig. 5. Displacement of the Plate with Different Cases of Central Cutout.

Figure 6 shows the peak response previous to first target frequency for all the cases are smaller in magnitude except 2A, 2B, 4B and 5A. Also the peak response is lowest between first and second target frequency for the cases 3B and 5B compared to other cases. Hence the case 3B has significantly smaller peak and one of the promising configurations based on peak response. The numerical study shows that plate with side cut-out and flap placed at centre of the plate - Case 3B most promising one and has been selected as best case to achieve better attenuations.

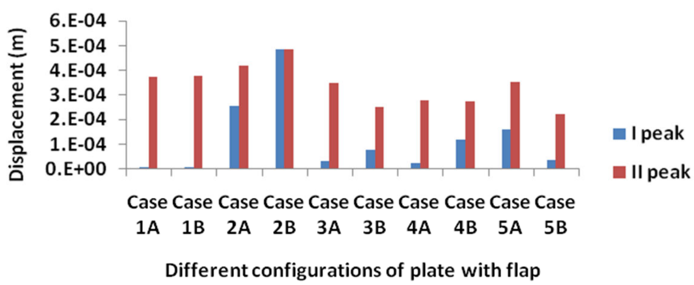
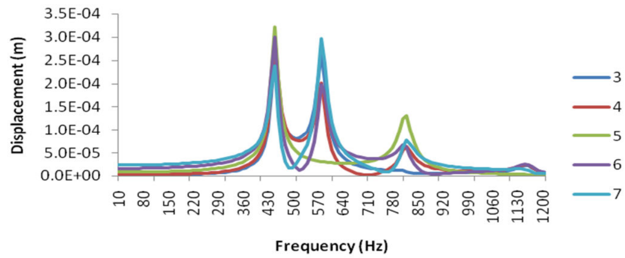
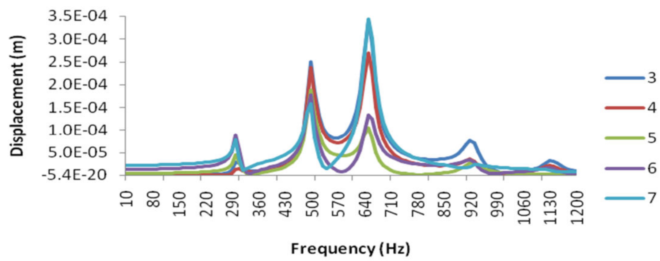


Fig. 6. The Peak Response of Different Configurations the Plate and Flap before First Target Frequency (I Peak) and between First and Second Target Frequency (II Peak).

The frequency response curves were obtained from the harmonic analysis at different measuring points on the plate and for different locations of the cut-out and flap, but presented in the report for the uncut plate and plate with cut-out and flap for best combination of locations. Due to the presence of cut-out, Points 2 and 3 corresponds to uncut plate are not available on the plate with side cut-out. The points located near cut-out are closer to these points and are considered for comparison. The displacements at points 1 and 9 are not presented in the report, as they were very small when compared to other points as they have located at very close to boundary. The frequency response curves for the uncut plate and plate with case 3B are shown in Fig. 7.



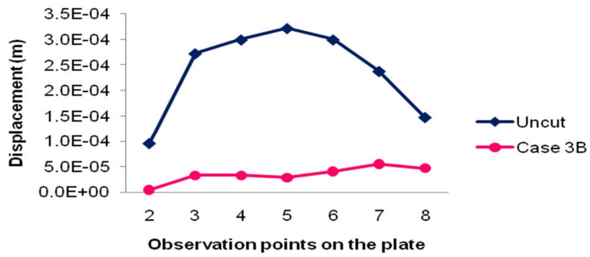
(a) Uncut Plate.



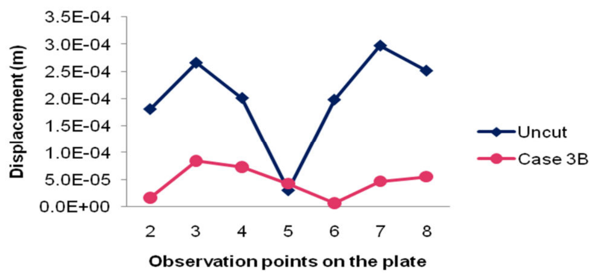
(b) Case 3B.

Fig. 7. Frequency Response Curves at Different Observation Points of the Plates.

Figure 8 shows the displacement at various locations of uncut plate, plate with side cut-out and plate with flap at best flap location (Case 3B). It was observed that the displacement of the plate configuration Case 3B at first and second target frequency is very much less compared to the uncut plate as well as plates with cut-out at all the points. The attenuation at all points are in the range of 65 - 95% except at point 5, where the attenuation is negative (-41%) at second target frequency, but with very smaller vibration amplitude. It can be observed from the table that the displacement values of the uncut plate at all the point except at point 5 are in the order of 1.8E-04 to 3.2E-04, whereas at point 5 the value is 4.29E-05. The displacement of the uncut plate at location 5 is about 15-20% of displacements at other points. Hence the increase in displacement of plate for the Case 3B may be neglected for selecting best flap location. The attenuation at various measuring points corresponding to first and second target frequencies along with the displacement of uncut plate and plate with flap and cut-out are given in the Table 2.



(a) First Target Frequency at 450 Hz.



(b) Second Target Frequency 580 Hz.

Fig. 8. Comparison of Displacement at Different Locations of the Uncut Plate and Plate of Case 3B.

Table 2. Displacement and Attenuation Percentage of Uncut plate and Plate with Case3B.

Frequency (Hz)	Configuration of the plate	Displacement (m) at various Observation points on the plate						
		2	3	4	5	6	7	8
450 Hz	Uncut	9.58E-05	2.73E-04	3.00E-04	3.22E-04	3.00E-04	2.38E-04	1.48E-04
	Case(3B)	4.37E-06	3.31E-05	3.30E-05	2.89E-05	4.08E-05	5.56E-05	4.75E-05
	Attenuation (%)	**95.43**	**87.87**	**89**	**91.02**	**86.4**	**76.64**	**67.9**
580 Hz	Uncut	1.81E-04	2.66E-04	2.01E-04	3.03E-05	1.98E-04	2.97E-04	2.51E-04
	Case(3B)	1.65E-05	8.48E-05	7.34E-05	4.29E-05	6.86E-06	4.71E-05	5.55E-05
	Attenuation (%)	**90.9**	**68.1**	**63.48**	**-41.6**	**65.35**	**84.14**	**77.9**

When compared with the displacement of the uncut plate, attenuations may be observed in the displacement of the plate for best case at both first and second target frequencies. These attenuations are in the order of 83% and 75% respectively on the basis of the maximum displacement on the plates. This is shown in Table 3.

The inference of the above analysis is that when the cutout is made at a location 25% of the length of the plate and the flap is placed near the cutout and boundary the resulting configurations of the flap are ineffective at first or second target frequency. On the other hand when the flap is placed closer centre of the plate on the other side of cutout, between 50 and 75% of length of the plate the resulting configurations are more favourable. The analysis shows that better attenuations can be obtained over entire plate when flap is placed at centre of the plate. The minimum and maximum attenuations are of 68%-95% in the first target frequency and 63% - 91% in the second target frequency. The attenuation on the basis of the maximum displacement for the cutout plate with flap compared with uncut plate are 83% at first and 75% in second target frequency of uncut plate are shown in Table 3.

Table 3. Attenuation on the Basis of the Maximum Displacement.

Target Frequency (Hz)	Maximum displacement in the plate (m)		Percentage Attenuation of vibration
	Uncut Plate	Case 3B	
450	3.22E-04	5.56E-05	83
580	2.97E-04	7.34E-05	75

4. Conclusions

The vibration control of clamped rectangular isotropic plate with cutout and vibratory flap placed at different locations subjected to harmonic excitations was investigated numerically using finite element analysis. The following conclusions were drawn from the analysis.

- The optimally placed vibratory flap reduces the vibration of the uncut plate at tuning and higher frequencies without experiencing amplification in its response as compared uncut plate.
- The plate with side cutout and flap has better attenuation of vibration compared to plate with central cutout and flap.
- The attenuation of vibration of a plate with cutout depends upon the flap location and flap orientation.
- The cutout made at a location of 25% of the length of the plate and the flap placed near the cutout and boundary is ineffective.
- If the flap placed between 50 and 75% of length of the plate, the resulting configurations are more favourable.
- The best attenuation of vibration of plate has been obtained at first and second target frequencies by making cutout at location 25% of its length and placing the flap at centre of the plate.
- The attenuations have been obtained over the entire plate which varied from 68-95% and 63-91% respectively at first and second target frequencies.
- The attenuation of vibration on the basis of maximum displacement for the cutout plate with flap is 83% and 75% respectively at first and second frequency of the plate.
- If the application demands central cutout, the flap placed at a location 25% and 75% of its length gives better attenuations.

References

1. Yu, S.D. (2009). Free and forced flexural vibration analysis of cantilever plates with attached point mass. *Journal of Sound and Vibration*, 321(1-2), 270-285.

2. Amabili, M.; Pellegrini, M.; Righi, F.; and Vinci, F. (2006). Effect of concentrated masses with rotary inertia on vibrations of rectangular plates. *Journal of Sound and Vibration,* 295(1-2), 1-12.
3. Jacquot, R.G. (2001). Suppression of random vibration in plates using vibration absorbers. *Journal of Sound and Vibration,* 248(4), 585-596.
4. Arpaci, A.; and Savci, M. (1987). A Cantilever beam damper suppressing rectangular plate vibrations, *Journal of sound and Vibration*. 115(2), 225-232.
5. Ozguven, H.N.; and Çandir, B. (1986). Suppressing the first and second resonances of beams by dynamic vibration absorbers. *Journal of sound and Vibration*, 111(3), 377-390.
6. Curadelli, R.O.; Ambroshini, R.D.; and Dhanesi, R.F. (2004). Vibrations control by attaching masses to a plate excited by rotating machinery. *Journal of sound and Vibration,* 273(4-5), 1087-1100.
7. Ji, J.C.; and Zhang, N. (2010). Suppression of the primary resonance vibrations of a forced nonlinear system using a dynamic vibration absorber. *Journal of Sound and Vibration*, 329(11), 2044-2056.
8. McMillan, A.J.; and Keane, A.J. (1997). Vibration isolation in a thin rectangular plate using a large number of optimally positioned point masses. *Journal of Sound and Vibration*, 202(2), 219-234.
9. Avalos, D.R.; and Laura, P.A.A. (2003). Transverse vibrations of simply supported rectangular plates with two rectangular cut-outs. *Journal of Sound and Vibration*, 267(4), 967-977.
10. Ranjan, V.; and Gosh, M.K. (2005). Forced vibration response of thin plate with attached discrete dynamic absorbers. *Thin Walled Structures* 43(10), 1513-1533.
11. Aida, T.; Kawazoe, K.; and Toda, S. (1995). Vibration control of plates by plate-type dynamic vibration absorbers. *Journal of Vibration and Acoustics*, 117(3A), 332-338.
12. Kerlin; R.L. (1988). Predicted attenuation of the plate-like dynamic vibration absorber when attached to a clamped circular plate at a non-central point of excitation. *Applied Acoustics*, 23(1), 17-27.
13. Ulz, M.H.; and Semercigil, E.S. (2008). Vibration control for plate-like structures using cut-outs. *Journal of Sound and Vibration*. 309(1-2), 246-261.
14. Mahadevaswamy, P.; and Suresh, B.S. (2011). Effect of position and tilt angle of the flap on vibration control plate - A finite element analysis. *International Journal of Mechanics and Solids*, 6(1), 27-38.
15. Mahadevaswamy, P.; and Suresh, B.S. (2012). Experimental study of effect of tilt angle of the flap on transverse vibration of plate. *International Journal of Engineering Research and Technology*, 1(10), 1-7.
16. Rao, S.S. (2003). *Mechanical vibrations* (4^{th} ed.). Pearson Education.