Experimental Study Of Effect Of Tilt Angle Of The Flap On Transverse Vibration Of Plate

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Abstract

Experiments have been carried out to determine the response of transverse vibration of a clamped rectangular isotropic plate with vibratory flap. Vibratory flap is like a dynamic vibration absorber and is used to reduce the vibration of the plate. An attempt has been made to discover the effect of tilt angle of the flap on natural frequency and response of the plate by placing flap at different tilt angles on the bare plate. The response of the plates was obtained at various locations of the plate through sine sweep test. The test result shows that the natural frequency and response of the plate varies with tilt angle of the flap. Investigating the effect of tilt angle on plate with flap revealed that there is a critical value where vibration suppression is effective. The vibration of plate with flap at tilt angle of 45° has been reduced by 93% at target frequency compared with bare plate. Experimental results were shown to be in reasonable agreement with numerical results.

Keywords

Transverse vibration, rectangular plate, vibratory flap, tilt angle.

1. Introduction

Thin plates are common structural elements employed in many engineering applications such as automotive, machine bodies, railway engine etc. and are subjected to a wide variety of excitations. Vibration of such plates can be controlled by using a dynamic vibration absorber (DVA) owing to its simplicity and effectiveness. A dynamic vibration absorber is a device which consists of an auxiliary mass which is relatively small in comparison with the mass of the vibrating structure. A similar concept has been used in the analysis of plates by using an additional plate called as flap which can vibrate independently, when it is attached as a cantilever plate on the main plate**.** The optimally placed vibratory flap reduces the vibration of the plate 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. The effect of mass ratio and location of the point mass analyzed [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 vibrations of rectangular plates carrying concentrated masses. Jacquot [3] presented a Numerical method to predict the effectiveness of application of damped dynamic vibration absorbers to suppress stationary random vibration of rectangular simply supported plates. Nagaya et al. [4] discussed a method of vibration control of a structure by using the vibration absorber without damping, which cannot be applied to the structure subjected to variable frequency loads. In this method, a variable stiffness vibration absorber is used for controlling a principal mode. Jedol Dayou and Brennan [5] investigated the global vibration of host structure using multiple tunable vibration neutralizers and have proposed a method to determine the optimum mass for the neutralizers. Arpact and Savci [6] 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 [7] employed a procedure to determine the optimum parameters of two dynamic vibration absorbers for the first two resonances of a cantilever beam. Curadelli et al. [8] reported the numerical and experimental study to determine the efficiency of the attached masses of the vibration control system. Dahlberg developed [9] an optimal design of dynamic absorber made by polymer-laminated steel sheets based mass, damping ratio and tuning frequency

Vinayak 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. Manfred H Ulz and S Eren 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 [14] 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 and reported that good attenuation in vibration can be obtained for tilt angle of 30 to 50^0.

The objective of this work is to study the transverse vibration response of a clamped rectangular plate with vibratory flap subjected to harmonic excitations through experiments. Vibratory flap is like a dynamic vibration absorber to keep the natural frequency of the resulting system away from the excitation frequency and hence to reduce the vibration of the plate structure during its operating conditions. The present investigation is focusing on the effect of tilt angle of the flap on the response of the plate-flap assembly. The study explores the effect of tilt angle for constant mass ratio of flap to plate and a critical value of tilt angle where vibration suppression is effective. For this purpose the highest displacement on the bare plate and plate with flap are compared, corresponding to target frequency and peak displacements of plate with flap .The sine sweep experiments were conducted for plates with the flap at different tilt angles.

2. Experimentation

The **Fig.1** shows the schematic diagram of the plate with flap used in the experimentation. The material selected for the plate is steel and size of the rectangular plate is 420 mm x 210 mm x 3 mm with aspect ratio (length to width) 2 .The flap of size of 80 mm x 80 mm x 3 mm is fixed on the bare plate using welded joint as a cantilever plate and the fixed side of the flap is perpendicular to longer side of the plate [14]. In structural applications bolted and riveted joints are used to introduce damping [1],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 size of the plate and flap kept constant to maintain constant mass ratio (7%) and the tilt angle of the flap was taken as variable parameter. The response of the plate was observed on the plate at different locations, at a distance of 1/4, 1/2 and 3/4 of the length of the plate as shown in **Fig. 2.**

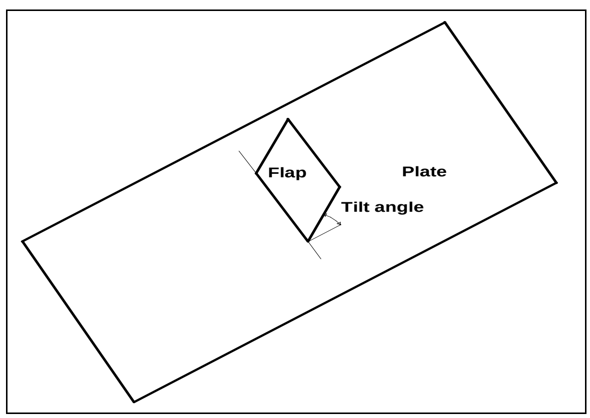

Fig. 1. Schematic diagram of the rectangular plate with flap placed at tilt angle.

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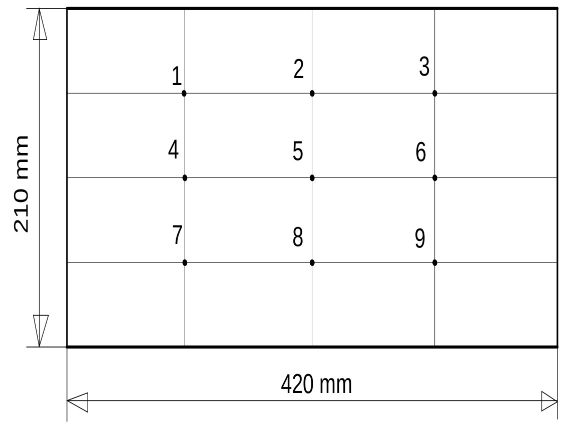

Fig. 2. Displacement measuring points

A sine sweep test was carried out using computerized FFT analyzer on rectangular bare plate and plate with flap placed at different tilt angles of 20º, 30º, 45º, 60º and 70º. **Fig. 3** shows the experimental setup used in the analysis. The plate was clamped along its edges to the fixture and is connected to the exciter at a location away from the centre of the plate. The plate was excited with the frequency varying from 0 to 1200 Hz. The power output from the amplifier is connected to the exciter which induces excitation on the plate. The piezoelectric accelerometers were attached on the plate which acts as sensing devices of vibration. The output of the accelerometer was fed to the analyzer through the amplifier. The acceleration of the plate at various locations was measured using accelerometers and the measured acceleration was converted into displacement using integration method.

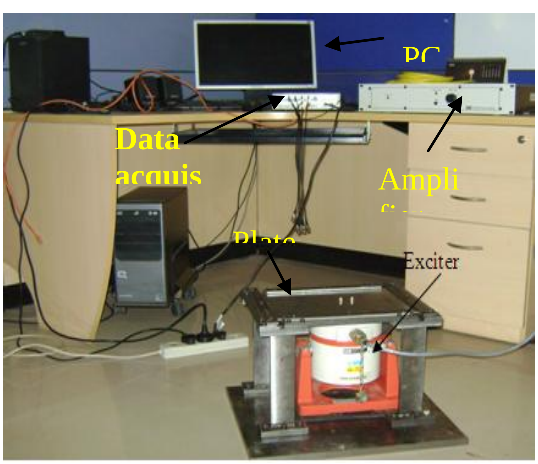

Fig. 3. Experimental Setup used in the investigation

3. Results and Discussion

The experiments were conducted for the rectangular plate with all the edges clamped using clamping devices and frequency response curves were obtained for the bare plate and plate with flap. The natural frequency of each plate was obtained from impulsive hammer test and the **Fig. 4** shows the Effect of tilt angle on first (PF1) and second (PF2) frequency of plate with flap comparing with target frequency of bare plate (P1). It was observed from the graph that the first frequency of the plate with flap increases with tilt angle, where as the second frequency decreases with tilt angle. It was also observed that for the tilt angle 20 and 45º the frequency range is more as compared to the tilt angle 60 and 70º.

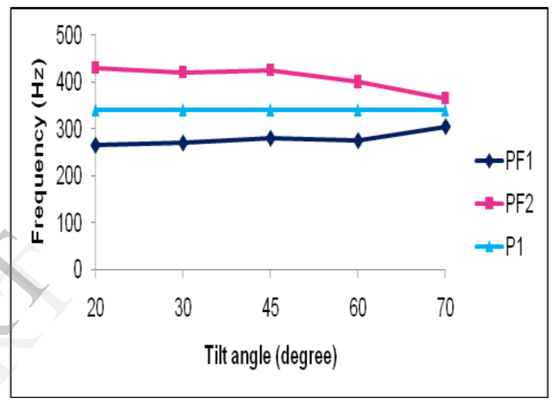

Fig. 4. Effect of tilt angle on first (PF1) and second (PF2) frequency of plate with flap comparing with first frequency of bare plate (P1)

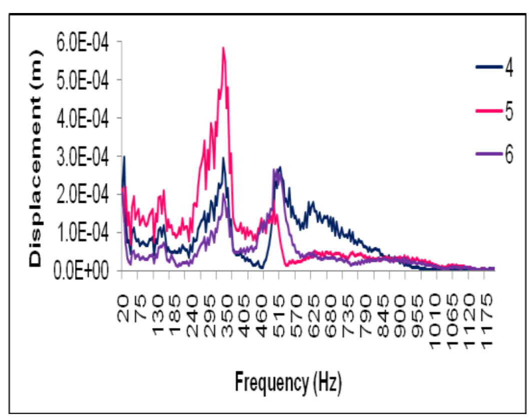

Fig. 5. Experimental frequency response curves for the bare plate at locations 4, 5 and 6

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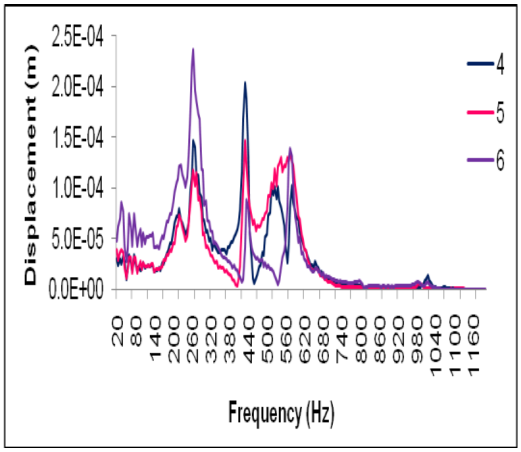
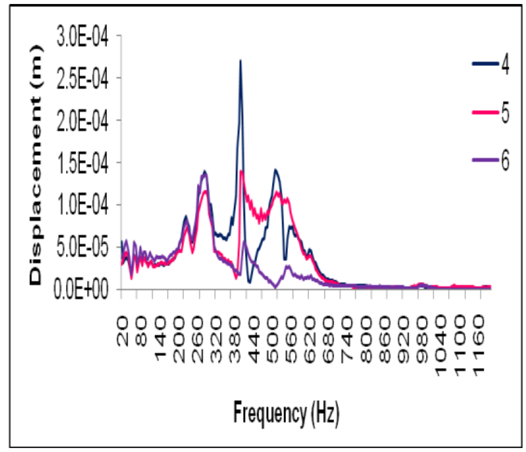
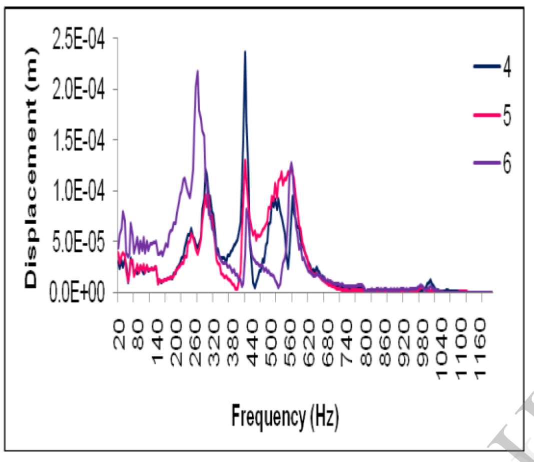


(a)



(d)



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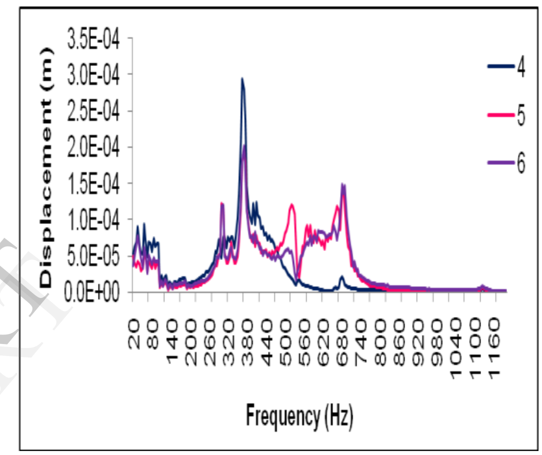
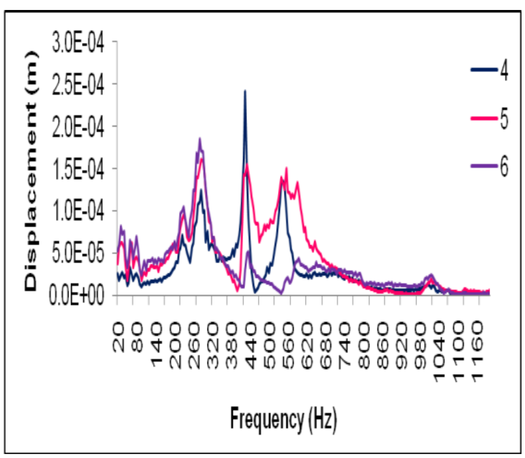


(e)



(c)

Fig.6. Frequency response curves for the plate with flap at different tilt angles at locations 4, 5 and 6. (a) Tilt angle 20º: (b) tilt angle 30º: (c) tilt angle 45º: (d) tilt angle 60º: (e) tilt angle 70º.

The frequency response curves for the bare plate obtained from sine sweep test are shown in **Fig. 5** and for the plate with flap placed at different tilt angles of 20º, 30º, 45º, 60º and 70º are shown in **Fig. 6(a-e).** From the response curves, the highest displacement in the plate in each mode is taken to plot the graph of displacement with tilt angle of the flap as shown in **Fig. 7.**

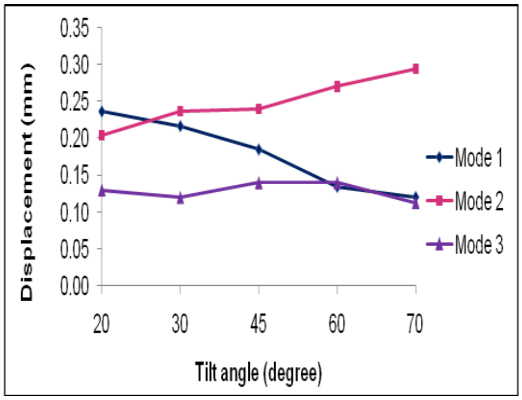


Fig. 7. Effect of tilt angle of flap on displacement of the plate.

It was observed from the graph of displacement with tilt angle that in mode 1 the displacement decreases as the tilt angle increases and in mode 2 the displacement increases as the tilt angle increases. But the displacement variation was very small in the range of 40º to 50º. In mode 3, no appreciable variation in the displacement with tilt angle. In higher modes the displacements were very small compared to first 3 modes. Higher displacements were observed at an angle of 20 to30º in mode 1 and 60 to 70º in mode 2, while at 45º in all modes the displacements were small compared to the angle 20, 30, 60 and 70º. From experimental results it was found that at target frequency, the attenuation of vibration of the plate at tilt angle of 45° was 93.7% and corresponding to first and second peak displacement, attenuation of vibration were respectively 72% and 61.4%. In mode 3 (third peak) comparatively poorer attenuation of 30-40 % was observed, but with smaller vibration amplitude. From the experiment it was concluded that the displacements were very high for the tilt angle 20 and 30° in mode 1 and for 60 and 70° in mode 2. It was also observed that for the tilt angle 20 and 45º the frequency range is more as compared to the tilt angle 60 and 70º.So it can be concluded that the tilt angle of 45º is the critical value to achieve the best attenuation.

4. Finite Element Analysis

In the published paper (14), results are available for aspect ratio 1.5 (length to width). In the present work, finite element analysis has been done for aspect ratio 2 for the purpose comparison of experimental results. In the finite element analysis, the material used for the plate and flap is steel having material properties of young's modulus 200 GPa, poisons ratio 0.3 and mass density of 7800 kg/m³. The size of the rectangular plate of 420 mm x 210 mm x 3 mm was used in the analysis. The flap size of 80 mm x 80 mm x 3 mm was used and attached on the plate as a cantilever plate. The flap was attached as a cantilever plate on the clamped rectangular plate. The plate with flap was modeled and meshed with shell 63 elements.

The harmonic analysis was carried out on a rectangular plate with and without flap. The frequency of the flap is tuned to the fundamental frequency of the plate (target frequency). A load of 50 N with a variable frequency of 0 to 1200 Hz was applied at a location away from the centre of the plate to avoid nodal lines. The displacements were observed on the plate at a distance of 1/4, 1/2 and 3/4 of the plate. The flap was fixed to the center of the plate as a cantilever with different tilt angles which varied from 0 to 90° in steps of 10°. The frequency response was recorded at different locations of the plate shown in **Fig. 2.**

5. Numerical results

From the modal analysis the frequencies of the plate with flap at different tilt angles were found and are used to find the effect of tilt angle on first and second frequency of the plate. The **Fig.8** shows the Effect of tilt angle on first (PF1) and second (PF2) frequency of plate with flap comparing with first frequency of bare plate (P1). It was observed from the graph that the first frequency of the plate with flap increases with tilt angle, where as the second frequency decreases with tilt angle. It was also observed that the range of frequency decreases with increase in tilt angle.

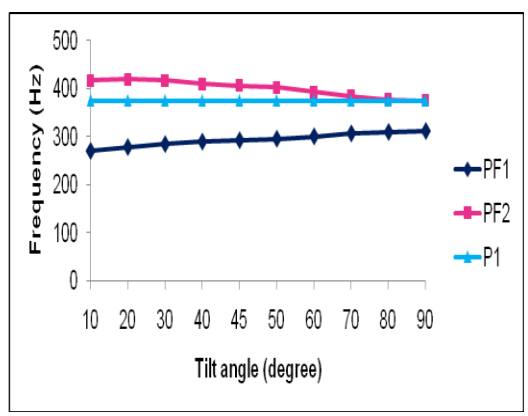


Fig.8. Effect of tilt angle on FEA first (PF1) and second (PF2) frequency of plate with flap comparing with first frequency of bare plate (P1).

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The frequency response curves were obtained from the harmonic analysis at different locations of the plate for different tilt angles. The highest displacement in the plate is observed for each tilt angle and is used to plot the curves of displacement with different tilt angle at mode 1, 2 and 3.

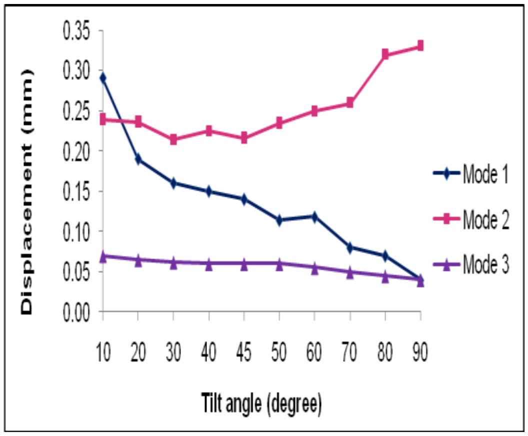
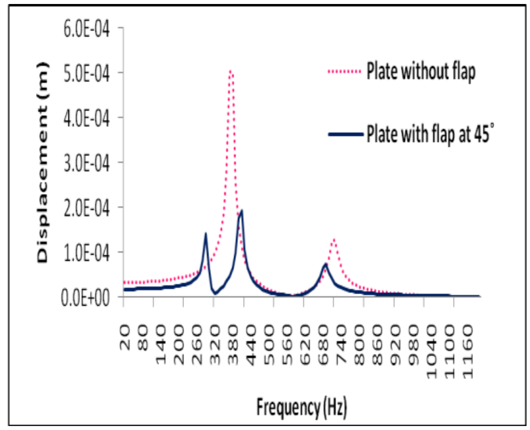


Fig. 9. Effect of tilt angle of flap on displacement of the plate according to FEA

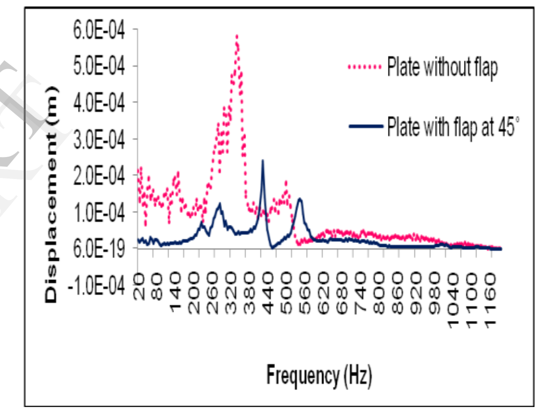
Referring to **Fig. 9** the effect of tilt angle on displacement of the plate, it was observed that in mode 1 the displacement decreases as the tilt angle increases and in mode 2 the displacement increases as the tilt angle increases. This shows the same trend as experimental results. From finite element analysis it was found that at target frequency, the attenuation of vibration of the plate at tilt angle of 45° was 90.7% and corresponding to first and second peak displacement, attenuation of vibration of the plate at tilt angle of 45° were respectively 72% and 61.4%.In mode 3 (third peak) comparatively poorer attenuation of 30-40 % was observed, but with smaller vibration amplitude. So it can be concluded from finite element analysis that, the tilt angle of 45º is the critical value to achieve the best attenuation.

6. Comparison of Experimental and FEA Results

The FEA and experimental frequency response curves to compare the displacement at location 5 (highest displacement position) with and without flap for tilt angle 45º are shown in the **Fig. 10.** It was observed from the graphs that at target frequency, the attenuation of vibration of the plate was 90.7%-93.2%. Corresponding to first peak displacement of plate with flap, attenuation of vibration of the plate was 72% -77.8% and for second peak 61.4%- - 62%. The comparison of displacement of the plate with and without flap at location 5 for tilt angle 45º and percentage reduction in vibration are tabulated in **table 1.**



(a)



(b)

Fig. 10. Comparison of displacement at location 5 with and without flap for tilt angle 45º: (a) FEA (b) Experimentation.

The inference of the above analysis is that the higher displacements were observed for the tilt angle 10-30° in mode 1 and 60-90° in mode 2. It was also observed that for the tilt angle 10 to 50º the frequency range more as compared to 60 to 90º. Between 40 -50 º in all modes the displacements were small with significant range of frequency compared to the angles 10 to 30º and 60 to 90º. This is due to the reason that the modal mass contribution varies with tilt angle of the flap. So it can be concluded that the tilt angle of 45º is the critical value to achieve the best attenuation.

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The difference in attenuation of vibration varies from 2 – 5 % from numerical to the experimental analysis. This is due to the limited trials in the experiment and difficulty in getting ideal boundary conditions.

Table 1.Comparision of displacement of the plate with and without flap at location 5 (Maximum displacement position)

Parameter	Method	Displacement of the plate without flap (m)	Displacement of the plate with flap (m)	Percentage reduction in vibration
At target frequency of the plate	FEA	5.03E-04	4.67E-05	90.7
	Expt.	5.82E-04	3.93E-05	93.2
At first peak displacement of the plate with flap (mode 1)	FEA	5.03E-04	1.41E-04	72.0
	Expt.	5.82E-04	1.29E-04	77.8
At second peak displacement of the plate with flap (mode 2)	FEA	5.03E-04	1.94E-04	61.4
	Expt.	5.82E-04	2.04E-04	62.0

7. Conclusions

The transverse vibration of clamped rectangular isotropic plate with vibratory flap for different tilt angles subjected to harmonic excitations was investigated experimentally. The following conclusions were drawn from the analysis.

- The first frequency of the plate with flap increases with tilt angle, where as the second frequency decreases with tilt angle.
- The best attenuation of vibration of a plate was obtained by placing the flap at center of the plate at a tilt angle of 45º in fundamental and at higher modes.
- At target frequency, the attenuation of vibration of the plate at tilt angle of 45° was 93.2%.
- Corresponding to first and second peak displacement of plate with flap, attenuation of vibration of the plate at tilt angle of 45° were 77.8% and 62% respectively.
- The experimental and finite element analysis showed the similar results with the small deviation in attenuation of 2 to 5%. This is due to the limitations in experimentation.

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