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REDUCING AIRBORNE SOUND RADIATION FROM MACHINE STRUCTURES

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Flat plates are used in constructing most of the machines and they are efficient radiators of airborne sound. Adding damping to these elements can reduce airborne sound in some cases, but adding damping is not always possible. This paper discusses the possibility of changing the flat nature of plates by introducing discontinuities in the form of corrugations. Several such designs of corrugations of various geometric configurations on plates measuring 480 mm x 300 mm were constructed and tested for insertion loss. Some of the configurations demonstrated an insertion loss more than 4 dB(A) in comparison to a flat plate of the same dimensions. The results of these experiments were compared with the vibro-acoustic simulations and experiments conducted on similar corrugated plates to establish the relationship between the higher insertion loss of some configurations with their geometry. And these results will be used to obtain an optimum configuration of the corrugations that can result in insertion losses higher than 4 dB(A) in comparison to a bare plate of the same dimension. The size of the plates used in the above experiments was kept smaller for the ease of conducting experiments. However, if similar corrugations are provided on much larger surface areas of machine structures, better insertion loss values can be expected.

Keywords: Vibro-acoustics, sound insertion loss, corrugated plate dynamics.

1. Introduction

Vibrating plate type structures are the main industrial noise sources. Especially, flat plate type structures commonly used in most of the machineries become sounding boards, when subjected to external disturbances. Sound is radiated because of the action of solid vibrating surfaces upon the fluid which comes in contact with such vibrating bodies. Most machine structures can be constructed using an assemblage of flat plates. Hence such complex structures can be modelled as simple plate like structures to study their vibroacoustic behaviour. Accurate prediction of sound radiation from and sound transmission through these type of complex structures, in the preliminary phases of the design process, helps a designer to treat the noise and vibration problems a priori. Because of varying geometrical forms, constructional features and material properties, sound radiation prediction from such vibroacoustic structures becomes extremely difficult. This is due to the fact that those mathematical modelling techniques which are used to predict the dynamic behaviour of such structures, are uncertainty prone due to damping distribution, boundary conditions, tolerance and structural joint properties. Even with the availability of sophistication in respect of computational software and hardware, large amount of model preparation and computation time pose infeasibility of the methods requiring large amount of data to be handled and processed. Experimental investigation thus helps one to correlate theoretical outcomes and validate the prediction results.

Various theoretical and experimental investigations have been carried out on sound transmission through isotropic single and double panel constructions in the past. For a transmission suite comprising of a room-panel-room, Crocker and Price [1] presented general power flow relationships. The power flow between two rooms was defined as the flow between non-resonant modes, when there are no modes excited in the panel in the frequency band under consideration. The prediction was made for the Sound Transmission Loss (STL), the radiation resistance and the vibration amplitude of a partition using (Statistical Energy Analysis) SEA. The “mass-law” sound transmission was observed to be due to non-resonant modal vibration, whereas the increased transmission was observed in the coincidence region, which was due to resonant modal vibration. Renji et al. [2] proposed a direct non-resonant coupling loss factor between a source room and a partition, and suggested a modified-SEA model to predict the non-resonant energy transfer between rooms and partition, by considering the sound transmission through a single-leaf aluminum plate. Cheng and Shyu [3] extended the work of Renji et al. [2] to further investigate sound transmission through double & triple leafs and estimated non-resonant energy transfer using a non-resonant coupling loss factor between a cavity and partition. Renji et al. [4] verified the modified SEA, which they had developed earlier, by acoustically exciting an aluminum plate in a reverberation chamber and estimated the response of the panels. Later, Cheng et al. [5] showed that the non-resonant response of the panel is insignificant at frequencies below the critical frequency, and through an experimental study, they suggested that the non-resonant response component should be included in the estimated response for enhancing prediction accuracy.

Large panels are generally made out of thin sheets and they have very low bending stiffness. Many forms of partition or enclosures employed in practice differ from plain, homogeneous and uniform plates for reasons of weight, stiffness to weight ratio, static stiffness or sometimes noise control requirements. Forming corrugations on thin flat plates is a very common approach to improve their static bending stiffness and buckling resistance. Sometimes, ribs directly attached to such thin flat plate structures or beads are introduced as out of plane deformations to increase their stiffness. This makes them orthotropic, giving higher bending stiffness in the two orthogonal directions. The critical frequency for bending waves travelling in the direction parallel to the corrugations is much lower in comparison to that of a uniform flat plate of equal thickness. Cordonnier et al. [6] investigated the influence of stiffening effect on corrugated steel structures and determined STL from sound intensity measurements and determined critical frequencies of orthotropic corrugated plates. The effects of coincidence on sound transmission loss and the acoustic resonances were studied by Ng and Zheng [7] through experimental investigation of double-leaf corrugated panel constructions by considering two separate leaves of a layer of reinforced concrete corrugated roof panel and a light weight profiled metal sheet. Because of high stiffness to weight ratios, recent trend, in aerospace structures, is to use composite sandwich constructions over flat metal structures to provide strength to the base structure. Such type of composite sandwich structures, however, suffer from the disadvantage of poorer sound transmission properties than metals [8].

Based on the fact that the radiation efficiencies of resonant modes are relatively small in comparison with the internal loss factors, in most of the research work it is assumed that there is a negligible contribution from the resonant transmission [7], which might be true for general cases. However, it is shown that the role of resonant transmission should not be neglected in some cases [9, 10, 11]. The neglect of resonant transmission is based on few quantitative investigations available on the relative contribution of resonant transmission components. The resonant transmission not only depends on damping but also on geometrical parameters such as thickness and size, a detailed investigation which includes the effect of those parameters is required.

By suitably altering the dynamic behaviour of flat plates, sound radiation efficiency can be reduced and in addition sound insertion loss can be improved. In the present work, flat plates are modified in order to explore the possibility of changing the flat nature of plates by introducing discontinuities in the form of triangular corrugations. The effect of attaching triangular stiffeners is not only providing strength and stiffness to the base structure, but also decreases the efficiency at which vibrations of the structure get coupled with the surrounding air. Several such designs of corrugations of various geometric configurations on plates were considered to get two different sets (9 configurations in each set) of corrugated plates and tested for insertion loss.

2. Experimental setup

With a view to capture an insight of the dynamic behaviour of corrugated plates, a bare plate of size 0.48 m × 0.3 m × 0.0035 m is considered. Different corrugated configurations are fabricated, by modifying bare plates with the help of inverted triangular shaped fabricated stiffeners of varying included angles and thicknesses. The corrugation configurations are enlisted in Table 1 for two different sets of plates and Figs. 1(a) & 1(b) show set-A & set-B corrugated plates, fabricated for performing experiments.

The experiments consisted of separate measurements in order to capture sound insertion loss of the flat bare plate (plate-1) as well as all the set-A & set-B corrugated plates (i.e. plates-2A to 10A & plates-2B to 10B). The plate parameters are so chosen to get varied corrugation configurations as to capture the wide range of dynamic behaviour of plates.

(a)

(b)

(c)

(d)

Figure 1: Background preparation for experimentation: (a) set-A corrugated plates, (b) set-B corrugated plates, (c) setup for insertion loss measurement and (d) microphone mounting arrangement.

3. Measurement of insertion loss

Sound insertion loss measurements on two sets (viz. set-A and set-B) of corrugated plates along with the flat bare plate have been carried out in an anechoic chamber of size 3.6 m × 3.6 m and having a cut-off frequency of 200 Hz. The inside of anechoic chamber consists of wedge blocks, on all six sides of the chamber, made of polyurethane having density of 30 kg/ m^3. Fig. 1(c) shows an experimental setup for the same. A marble cutter as a sound source was put inside a wooden enclosure and an arrangement was made to insert the plates at the opening. The wooden enclosure was properly sealed using a sealant. The sound in the test section was measured by using an array of four B & K 4939 free-field condenser type microphones of size 6.53 mm (1/4") having a flat frequency response of 4 Hz to 100 kHz. The dynamic range of the microphones is 28 dB to 164 dB and operating temperature -40^o C to 150^o C. The microphones were suitably placed on the curved wooden frame, which was then clamped with pre-fabricated frame as shown in the Fig. 1(d). The microphones were attached with a B & K pre-amplifier (type 2970) and were powered by four channel *NEXUS* type 2690-OS2 signal conditioning amplifier. The output was fed to the computer through a NI DAQ system PCI 4462. Using the LabVIEW environment, the microphone data (in Volts) were obtained. The plywood sheet of 12 mm thickness was placed underneath the box containing sound source and the experiments were performed for the measurement of Sound Pressure Level (SPL) for calculating the insertion loss of the plates. Initially all the four microphones were calibrated using standard calibration kit and the sensitivities (in V/Pa) were recorded for subsequent processing. Sound pressure levels were calculated from the sensitivities for the cases with and without inserting the plates.

Table 1: Parameters of fabricated corrugated plates for experimentation.

<table>
<tr><th>Plate No.</th><th>Overall size of corrugated plate X × Y × Z (m)</th><th>Included angle of Stiffener</th><th>Thickness of stiffener (mm)</th></tr>
<tr><td>1</td><td rowspan="10">0.48 × 0.3 × 0.0035</td><td colspan="2">Flat bare plate</td></tr>
<tr><td>2</td><td>70^0</td><td>1.5</td></tr>
<tr><td>3</td><td>70^0</td><td>2</td></tr>
<tr><td>4</td><td>70^0</td><td>3</td></tr>
<tr><td>5</td><td>90^0</td><td>1.5</td></tr>
<tr><td>6</td><td>90^0</td><td>2</td></tr>
<tr><td>7</td><td>90^0</td><td>3</td></tr>
<tr><td>8</td><td>120^0</td><td>1.5</td></tr>
<tr><td>9</td><td>120^0</td><td>2</td></tr>
<tr><td>10</td><td>120^0</td><td>3</td></tr>
</table>

The insertion loss was evaluated for one-third octave band frequencies for each plate using Eq. (1) given by Aygun and Attenborough [12] and then with the help of standard A-weighting values for the one-third octave band centre frequencies, corrected A-weighted sound insertion loss (in dB) was calculated.

$$IL = \overline{L}_{p_{II}} - \overline{L}_{p_I} \tag{1}$$

In Eq. (1), $\overline{L}_{p_I}$ is the spatial average sound pressure level in the frequency band in the test room, when the plates were inserted, and $\overline{L}_{p_{II}}$ is the spatial average sound pressure level in the frequency band in the anechoic room, when no plate was inserted.

The spatial average SPL has been calculated by measuring the local SPL at three different positions of the microphone array by varying the height of the curved wooden frame (shown in Fig. 1(d)) which was repeatedly clamped on the fabricated frame placed in front of the wooden enclosure.

The spatial average sound pressure level, $\overline{L}_p$, in dB, was calculated from the local sound pressure levels, L_{p_i}, using Eq. (2) given by [12]:

$$\overline{L}_p = 10\log\left[\frac{1}{n_m}\sum_{i=1}^{m}10^{\frac{L_{pi}}{10}}\right], \tag{2}$$

where n_m is the number of measurements.

4. Results and discussion

The results of A-weighted measured insertion loss obtained are shown in Fig. 2 for the corrugated plates and compared with a flat bare plate. Because of low frequency threshold of the anechoic chamber (as mentioned earlier), the measured insertion loss for frequencies more than 200 Hz are plotted. Figures 2(a)-2(c) show the comparison of measured insertion loss data for the first set of experiments for set-A corrugated plates with that of a flat bare plate. The theoretically calculated critical frequency of a flat bare plate is 3358 Hz. As the stiffeners are incorporated to the bare plate, the behaviour of the corrugated plate becomes orthotropic, i.e. there are two critical frequencies corresponding to the direction of bending wave travel. Due to the effect of corrugation and added stiffness, the critical frequencies shift towards the low frequency, which are clearly observed. The theoretical critical frequencies of set-A plates lie between 300 Hz and 340 Hz as the lowest critical frequency whereas highest critical frequency range calculated lie between 1386 Hz and 2100 Hz. Similarly, the theoretical critical frequencies of set-B plates lie between 215 Hz and 510 Hz as the lowest critical frequency whereas highest computed theoretical critical frequencies lie between 3200 Hz and 4100 Hz. The critical frequency dips are very close to the theoretically evaluated values and clearly seen in the experimental insertion loss results.

The results obtained clearly indicate that the stiffeners improve sound insertion loss at almost entire frequency range, except at and just below the coincidence range, i.e. between 400 Hz and 1 kHz, where a little deterioration is observed. This behaviour can be well explained by the effect of introduced low wavenumber components due to the effect of boundaries of stiffeners, which then subsequently increases the radiation efficiency below the critical frequency. As the radiation efficiency is equivalent to a normalized sound power with respect to the specific acoustic impedance of the surrounding media, plate area and vibrating normal velocity of the plate, it can be seen as a measure of the sound radiation in dependence of the geometry of the plate. Because of the variation in configuration of two set of plates, the radiation efficiency plays significant role just below the coincidence. At the critical frequency, i.e. when the plate bending wavelength equals the trace wavelength of acoustic waves, the plate vibration amplitude increases and the radiation efficiency also increases. Thus, at the critical frequency the sound transmission is high and is due to modes resonant around a band centred at this frequency. This behaviour is dominant for set-A plates than set-B plates because the radiation efficiency of the former is relatively more than that of the later due to extra edges and subpanels formed due to corrugation. As the frequency range between 400 Hz and 1 kHz being the region where edge modes contribute to sound radiation, more sound is radiated by set-A plates than set-B plates and hence sound insulation performance is little poor for set-A plates at and just below the coincidence region due to resonant sound transmission which is depicted in Fig. 2(a)-2(c). In the frequency region well below the coincidence, i.e. up to 400 Hz, it is found that the insertion loss of corrugated plates is more as compared to a flat bare plate due to the modes which are not resonant in this frequency band. This is because the vibration amplitude of resonant modes is low and the radiation efficiency is also low.

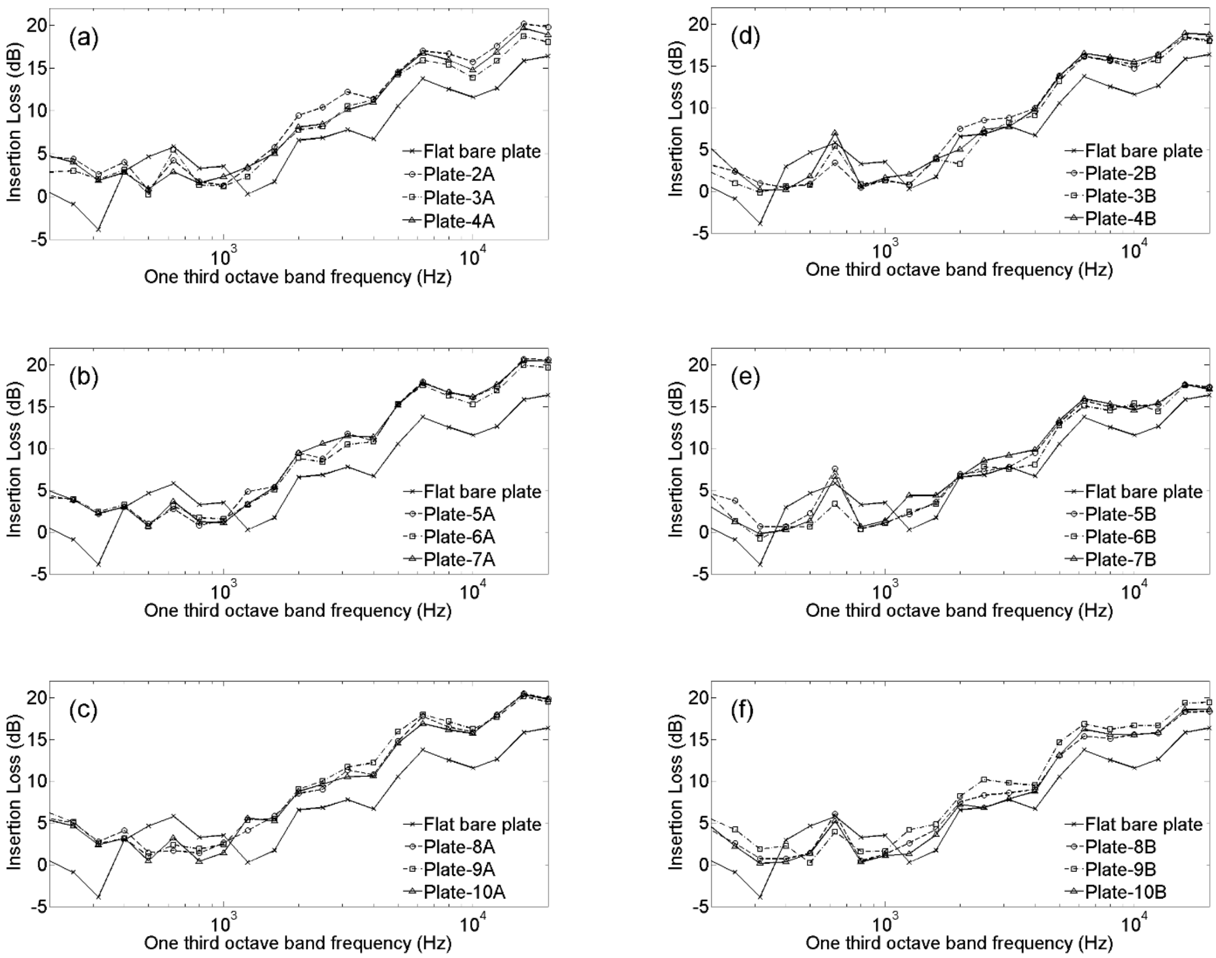

Figure 2: Comparison of experimental insertion loss of set-A & set-B corrugated plates with flat bare plate: (a) plates-2A, 3A & 4A; (b) plates-5A, 6A & 7A; (c) plates-8A, 9A & 10A; (d) plates-2B, 3B & 4B; (e) plates-5B, 6B & 7B & (f) plates-8B, 9B & 10B.

The enhancement of sound insulation effect due to corrugation is very distinctly observed at higher frequencies. Above 1 kHz frequency, all the set-A plates show better sound insulation performance (refer Figs. 2(a)-2(c)) as compared to set-B plates (Figs. 2(d)-2(f)) and even much better than a flat bare plate. This is due to reduced vibration levels and hence reduction in sound radiation efficiency because of increase in surface mass density of corrugated plates as compared to a flat bare plate. The insertion loss at higher frequencies is more dominant for set-A plates than set-B plates due to radiation efficiency of former being less at higher frequencies. Above 4 kHz frequency, the insertion loss separation between the set-A plates and a flat bare plate increases, specifically plates-9A shows very clear benefit of the optimum geometry amongst the population under consideration, giving an overall gain in sound insertion loss of 4.6 dB(A) over a flat plate of the same dimensions. The vibro-acoustic simulation results based on SEA for the prediction of sound transmission loss and radiation efficiency are also confirming same behaviour of corrugated plates.

A very significant effect of the periodicity of stiffeners is to divide the frequency range into a sequence of pass bands, where vibration can propagate, and stop bands, where it cannot. For the peaks and dips in the insertion loss this pattern of stop and pass bands is responsible, which can be clearly seen in the results. Especially in the high frequency region, the oscillations above the coincidence are due to these pass-stop frequency bands. The periodically distributed stiffeners with relatively narrow separations restrict the deformation of the corrugated plates, offering therefore the corrugated plates a larger stiffness than that of those plates with stiffener spacing being wider. Out of all the corrugated plates, the overall sound insertion loss of plate9A is found to be much better, indicating that there is an optimum configuration which gives better results on stiffening the bare plate. Not only the spacing between the stiffeners plays role in sound insulation, but also the thickness of stiffeners is important, as far as vibro-acoustic behaviour is concerned.

5. Conclusion

The possibility of changing the flat nature of plates by introducing discontinuities in the form of corrugations is attempted. Several designs of corrugations of various geometric configurations on plates were fabricated in order to conduct experiments to capture sound insertion loss. Both low and high frequency behaviours of the corrugated plates are affected by suitable choice of the stiffener configuration and the spacing between the stiffeners as well. The coincidence of bending waves in the direction of stiffeners and the acoustic waves have been occurring at relatively much lower frequencies. In the frequency range just below the coincidence, little deterioration in sound insertion loss is observed due to additional edges incorporated because of stiffeners. At low frequencies, due to relatively more sound radiation efficiency of set-A corrugated plates than set-B plates, the insertion loss of set-B plates shows only a small improvement. For the entire frequency range, set-A plates gives an overall sound insertion loss more than set-B plates. It is suggested through the experimental results that there should be an appropriate stiffener configuration which ensures maximum benefit of stiffening the plates as far as sound insulation is concerned.

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