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A modelling approach for the vibroacoustic behaviour of aluminium extrusions used in railway vehicles

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Abstract

Modern railway vehicles are often constructed from double walled aluminium extrusions, which give a stiff, light construction. However, the acoustic performance of such panels is less satisfactory, with the airborne sound transmission being considerably worse than the mass law for the equivalent simple panel. To compensate for this, vehicle manufacturers are forced to add treatments such as damping layers, absorptive layers and floating floors. Moreover, a model for extruded panels that is both simple and reliable is required to assist in the early stages of design. An statistical energy analysis (SEA) model to predict the vibroacoustic behaviour of aluminium extrusions is presented here. An extruded panel is represented by a single global mode subsystem and three subsystems representing local modes of the various strips which occur for frequencies typically above 500 Hz. An approximate model for the modal density of extruded panels is developed and this is verified using an FE model. The coupling between global and local modes is approximated with the coupling between a travelling global wave and uncorrelated local waves. This model enables the response difference across the panels to be predicted. For the coupling with air, the average radiation efficiency of a baffled extruded panel is modelled in terms of the contributions from global and local modes. Experimental studies of a sample extruded panel have also been carried out. The vibration of an extruded panel under mechanical excitation is measured for various force positions and the vibration distribution over the panel is obtained in detail. The radiation efficiencies of a free extruded panel have also been measured. The complete SEA model of a panel is finally used to predict the response of the extruded panel under mechanical and acoustic excitations. Especially for mechanical excitation, the proposed SEA model gives a good prediction compared with the measurement results.

1. Introduction

In recent years, aluminium extrusions have been used widely as the main structural component of railway passenger trains. These are usually composed of two outer skins and an interconnecting plate lattice, to give a stiff, light construction. However, the acoustic performance of such panels is less satisfactory, with the airborne sound transmission being considerably worse than the mass law for the equivalent simple panel [1–3].

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To compensate for this, vehicle manufacturers are forced to add treatments such as damping layers, absorptive layers and floating floors.

To model the interior noise of trains at an early design stage, the vibroacoustic behaviour of the vehicle, including the aluminium extrusions, needs to be modelled. The method of statistical energy analysis (SEA) is generally considered to be the most appropriate approach for predicting the interior noise [1,2]. The objective of this paper is to establish a modelling approach for aluminium extrusions, that is both simple and reliable, which enables an extruded panel to be represented in terms of a small number of subsystems and be incorporated into an overall SEA model for interior noise of vehicles.

The vibration of an extruded panel is dominated by global behaviour at low frequency while local motions of individual strips are dominant at high frequency. A poor acoustic performance, where the sound transmission loss is much lower than that predicted with the mass law, has been observed but this is not well modelled in previous studies [1–3]. For both acoustic and mechanical excitations, there is a vibration level difference across an extruded panel for frequencies where local motions are dominant. It is therefore essential for the model to predict such a characteristic of the structure successfully.

Shaw [1] first studied extruded aluminium plates from railway vehicles in terms of SEA. His work concentrated on the prediction of both the air-borne transmission loss and the noise radiated by mechanically excited vibrations. His SEA model of the extruded panel consisted of five subsystems, two representing each of the two outer skin plates and one for a tie plate between them. It was assumed that the tie plate subsystem represents a plate of the same size as the combined area of all interconnecting plates. All the structural connections were considered to be 90° T-type junctions. However, this model cannot take account of the global behaviour of extruded panels at low frequencies. Geissler and Neumann [2] calculated the transmission loss of extruded panels by using the AutoSEA software. In their study, the equivalent sandwich panel and the rib-stiffened plate formulations were used to represent extruded panels as subsystems in the SEA model. The calculation results using either model showed considerable deviation from the measured data. Pang [4] has recently produced a more successful SEA model using commercial SEA software.

Extruded panels are also modelled with other methods. Kohrs [3] presented a hybrid method for transmission loss prediction consisting of finite element (FE) calculations of the extruded profile and an analytical transmission loss model for an orthotropic single plate. Pang [4] considered a spectral finite element model of an extruded panel. Pezerat et al. [5] presented an analytical model for extruded panels, in which the motion is decomposed in terms of the assumed mode shapes of the two outer plates. Continuity of displacements and rotations at each connection leads to a system matrix equation.

In general, current studies on extruded panels are not sufficient to provide an efficient model. Consequently, more investigations are required in this area. This paper presents an SEA model for extruded panels and prediction results for mechanical and acoustical excitations. Comparisons are made with experimental results.

2. SEA model

It has been shown in an FE modal analysis [6] that only the first few modes of the cross section of an extruded panel appear to involve global behaviour of the structure. At higher frequencies the mode shapes become complicated and local motion of a single plate strip begins to dominate the modes. Global modes may be considered as those in which the energy is distributed over the whole system, involving motion of the whole panel, whereas ‘local modes’ involve mainly motion of one (or more) members without much motion of the whole panel. Normally, the local modes appear first at the widest strips. The global modes are dominant at low frequencies while local modes occur for frequencies typically above 500 Hz. The vibration energy of extruded panels is distributed in the global modes of the whole panel and local modes that are dominated by individual strips.

In constructing the SEA model for an extruded plate, global modes, local modes on the two sides and those of the interconnecting strips of the panel may be treated as separate SEA subsystems. Xie et al. [7] have proposed a three-subsystem SEA model. In this model, it was assumed that there is no direct coupling between the local modes on the two sides of the extruded panel, either through the air or foam between them or through the connecting ribs. Instead, the connection between them is assumed to act through the global modes of the structure. In the present paper, the intermediate connecting ribs are considered as an additional

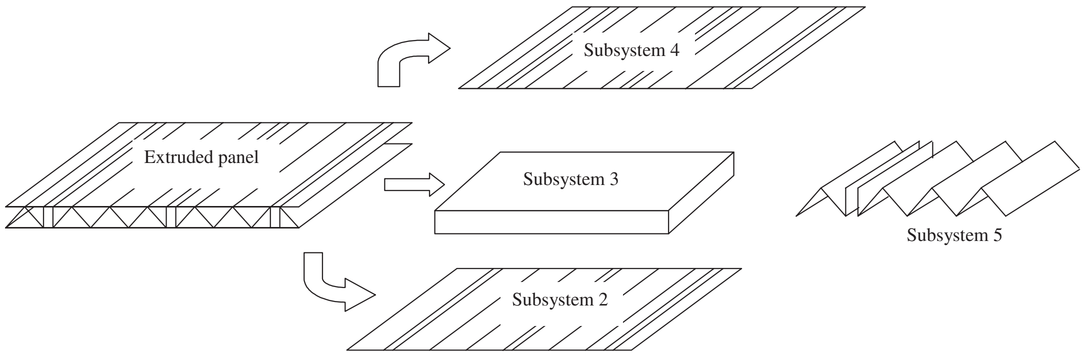

Fig. 1. SEA subsystems of extruded panels. Subsystem 3 represents global modes, subsystems 2, 4 and 5 represent local modes.

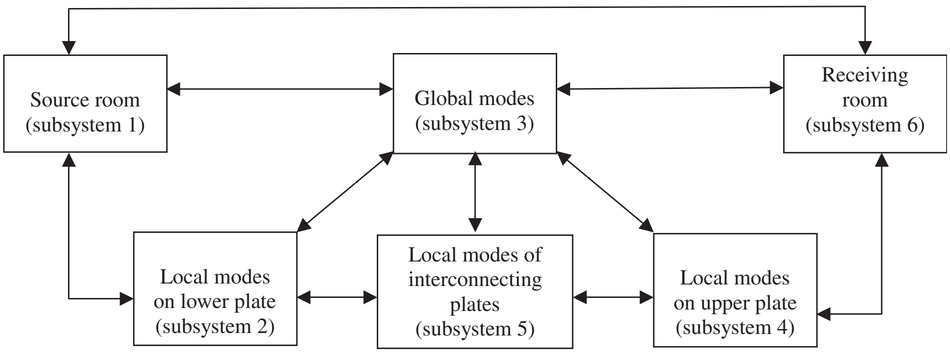

Fig. 2. SEA block diagram for extruded plate in a transmission suite showing the couplings among subsystems.

subsystem, which has direct couplings with the two local modes subsystems. This gives a four-subsystem SEA model for an extruded panel, as illustrated in Fig. 1. The main tasks here are to model the modal density of these subsystems and the coupling between them and to apply SEA models to estimate the response of the panel. It is worth pointing out that the air in the extrusion cavity is not modelled as a SEA subsystem in the present model. The coupling through the air has been analysed in Ref. [6] based on the theory of the sound transmission through a double-leaf partition. The analysis showed that the coupling through the air is much weaker than that through the structure. Indeed the stiffness of the tie plates is much greater than the air in the extrusion cavity and therefore dominates the coupling of the vibration of the two outer plates. Pang [4] found a difference in results when the air layer was included, but in that case the air layer provided considerable damping due to the presence of foam in the air layer. In the present model this effect is accounted for simply by applying a higher damping loss factor to the structural subsystems.

When the sound transmission through the panel in a transmission suite is considered, the appropriate couplings between acoustic subsystems and the global and local mode subsystems are required. This also forms a major task in the present work. The coupling mechanisms in the SEA model are illustrated in Fig. 2.

3. Modal density

3.1. Model

The modal density is one of the most important parameters required to define a subsystem within an SEA model. In order to represent an extruded panel in a simple SEA model, it is necessary to determine its modal densities. For basic structures such as a rod, a beam or a plate, analytical expressions have been available for many decades. However, for complicated structures like extruded panels, there is no theoretical expression available to calculate their modal densities. The modal density of extruded panels can be considered as the sum of the modal density of the global and local modes. This is expressed by

$$n(\omega) = n_g(\omega) + n_l(\omega), \tag{1}$$

where $n_g(\omega)$ and $n_l(\omega)$ represent the modal densities of global and local modes, respectively.

The global modes can be considered on the basis of an equivalent uniform plate. The modal density of the local modes can be obtained from that of a large plate, having an area equal to the combined area of all the strips, with a number of line constraints applied. The global modes have much smaller wavenumbers than the local modes and the modal density of global modes is expected to be much lower than that of local modes.

It has been shown by Xie et al. [8] that boundary conditions and intermediate constraints on a plate have systematic effects on the mode count and modal density of the structures. Assuming a simply supported plate, an approximate analytical expression for the modal density of the global modes has been given by [7]

$$n_g(\omega) = \frac{S}{4\pi}\sqrt{\frac{M''}{B_g}} - \frac{1}{4}\left(\frac{M''}{B_g}\right)^{1/4}\left(\frac{L_x + L_y}{\pi}\right)\omega^{-1/2} \tag{2}$$

where L_x and L_y are the dimensions of the extruded plate, S is its surface area, M'' is the equivalent mass per unit area, and B_g is the equivalent bending stiffness for the global modes [6]. The first term corresponds to the plate area and the second term (which depends on the boundary conditions) corresponds to the perimeter length. Similarly for local modes, if the plate is represented by a rectangular plate divided by p intermediate constraints,

$$n_l(\omega) = \sum_{1}^{p+1} n_i, \tag{3}$$

where n_i is given by

$$n_i = \begin{cases} \dfrac{l_i L_y}{4\pi}\sqrt{\dfrac{m''}{B}} - \dfrac{1}{4}\dfrac{l_i}{\pi}\left(\dfrac{m''}{B}\right)^{1/4}\omega^{-1/2} - \dfrac{1}{2(p+1)}\left(p\delta + \dfrac{1}{2}\right)\dfrac{L_y}{\pi}\left(\dfrac{m''}{B}\right)^{1/4}\omega^{-1/2} & \text{for } N_i > 0, \\ 0 & \text{for } N_i < 0, \end{cases} \tag{4}$$

where m'' is the mass per unit area of the strips, B is the bending stiffness of the strip, $p+1$ is the number of strips, δ is the boundary effect of the intermediate constraints between strips (1 for clamped, $\frac{3}{4}$ for simple support and 0 for free condition [8]), l_i is the length of the shorter edge of each strip, L_y is the length of the longer edge and N_i is the corresponding mode count, which is given by

$$N_i = \frac{l_i L_y}{4\pi}\sqrt{\frac{m''}{B}}\omega - \frac{1}{2}\frac{l_i}{\pi}\left(\frac{m''}{B}\right)^{1/4}\omega^{1/2} - \frac{1}{p+1}\left(p\delta_{\text{pinned}} + \frac{1}{2}\right)\frac{L_y}{\pi}\left(\frac{m''}{B}\right)^{1/4}\omega^{1/2}. \tag{5}$$

The first term in Eq. (4) is proportional to the area of strips, which represents the modal density of a plate without including the effects due to boundary conditions. The remaining terms represent the effect from the boundary conditions. The condition $N_i > 0$ implies that the modal density is physically meaningful only when modes are present. To use Eqs. (3)–(5) to calculate the modal density of the local modes, it is necessary to know the dimensions of all strips. These equations can be applied to the whole panel or to local mode subsystems on the outer or intermediate plates.

3.2. Results

An example extruded plate has been used for the purpose of this paper. It is of aluminium and has dimensions $2.016 \times 1.0 \times 0.07$ m. It is composed of 77 strips, the thickness of each strip being 0.003 m. The widest strip is 0.16 m wide while the narrowest strip is 0.046 m wide.

Fig. 3 presents the modal density calculated from Eqs. (1)–(5). Simple supports are assumed for the line constraints between strips in these calculations, i.e. $\delta = 3/4$. The result from an FE analysis is also given,

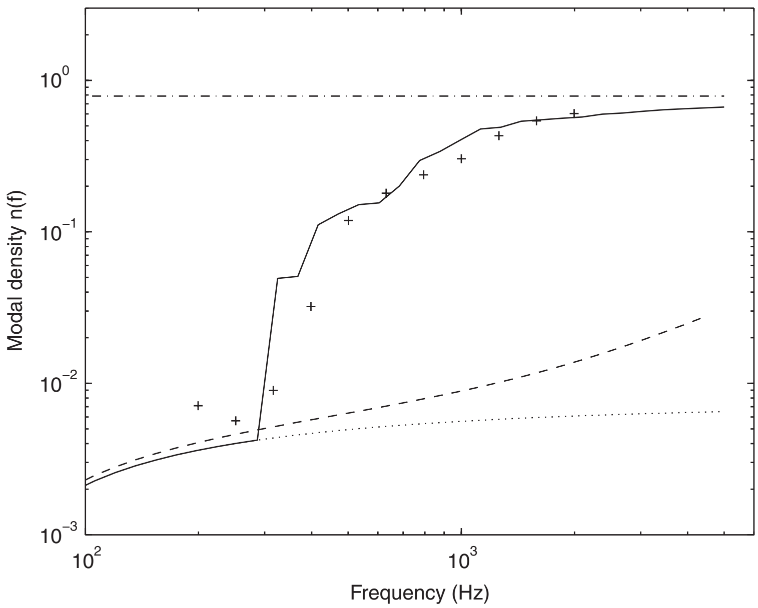

Fig. 3. Modal density of the extruded plate. —, Estimated by Eqs. (1)–(5); +, FEM in overlapping octave bands; · – ·, simple plate; – –, global modes by thick plate model; · · · · ·, global modes by thin plate model.

expressed in overlapping octave bands in order to smooth the data at low frequencies. The calculated result generally agrees well with that from the FEM model.

The global modes were modelled using both thin and thick plate theory. The modal density obtained using thick plate theory is about four times higher than that based on thin plate theory at 3 kHz. However, compared with the modal density of local modes at this frequency, the modal density of global modes is much smaller. For the purpose of illustration, the result using a simple plate, which simply considers the extruded panel as a plate having the area of all strips without including the effects of the boundary conditions, is also shown in Fig. 3. The local mode behaviour converges to this result only at very high frequency.

4. Coupling between global and local modes

Due to the low modal density of global modes and the complexity of the structure, it is quite difficult to model the coupling between them and local modes in terms of SEA. Similar problems have been studied to address the dynamic behaviour of complicated or built-up structures over recent decades. In general, these studies were concentrated on beam–plate systems [9,10]. Normally, vibration sources are supported by stiff beams. The power injected into and transmitted around structures is controlled by long-wavelength waves generated in these beams. As these long waves propagate along the stiff beams they generate short-wavelength flexural waves in the attached flexible plates. The coupling between global and local modes within extruded panels is physically similar to that between beams and plates within beam–plate systems. There is no general approach available to deal with the vibration behaviour within a complicated built-up structure comprising long- and short-wavelength components. However, one important conclusion from investigations of beam–plate systems is that the short wave components, or plates, act principally to introduce damping [9,10] and present a locally reacting impedance to the long waves at the structural joints [11], provided the difference between the wavelengths of beams and plates is large. Also, apart from this damping effect, the dynamic behaviour of the beams is not greatly affected by the presence of the attached plates. For extruded panels, the local modes within strips can be considered to be driven by the global modes. This can form an analogue to beam–plate systems. Here, global modes are equivalent to the beams and local modes to the plates.

The coupling between the global and local modes has been modelled based on analysing the input power of a set of independent finite beams driven by kinetic excitations at their two ends [6]. The kinetic excitation is modelled with travelling global waves. The local modes in each strip of the extruded panel are assumed to be independent, which is reasonable if they are not too similar in width. Analogous to the beam–plate case [11], an expression for the coupling loss factor from global modes to local modes has been given by [6]

$$\eta_{gl} \cong \frac{m''(p+1)}{M''kL}, \tag{6}$$

where $p+1$ is the number of strips in the length of L of the extruded panel, k represents the wavenumber of bending waves in the strips, M'' is the equivalent mass per unit area of the whole panel and m'' is the mass per unit area of the strips.

5. Coupling between local modes

It has been shown in Ref. [6] that the coupling between the two outer plates of the extruded panel through air can be neglected compared with the coupling through the global modes. However, the coupling from the global modes to the local modes is insufficient by itself to predict the response of the panels correctly when using a three-subsystem SEA model to represent extruded panels [7]. The intermediate connecting ribs between the two face plates can actually act as a bridge to connect the two local mode subsystems on the two sides of the extruded panel. The coupling between local modes is modelled using standard expressions for various structural joints.

The extruded panel can be considered to be made from a combination of many substructures, each of which has a similar form and consists of a number of strip plates as shown in Fig. 4. The typical structure is plotted in solid lines. The joints connecting the intermediate ribs and two face plates have two basic forms: ‘T’ joint and ‘$\underline{\vee}$’ joint. Coupling between the three local mode subsystems occurs through the interaction of waves at these junctions. Plates 1 and 2 correspond to the local modes on the source room side (subsystem 2). Plates 3 and 4 form the intermediate local modes (subsystem 5). Plates 5–7 form the local modes on the receiving room side (subsystem 4). The coupling between local modes on the face plate and the intermediate local modes can be obtained by working on the typical structure shown in solid lines in Fig. 4.

Using expressions for the transmission coefficients for various joints according to Craik [12], the coupling loss factors between these local mode subsystems have been given by [6]

$$\eta_{si} = \frac{6c_g\tau_{\text{cross}}}{\omega\pi(l_1+l_2)}, \quad \eta_{is} = \frac{6c_g\tau_{\text{cross}}}{\omega\pi(2l_3+l_4)}, \tag{7}$$

$$\eta_{ir} = \frac{c_g(2\tau_{\text{Tee}C}+4\tau_{\text{cross}})}{\omega\pi(l_4+2l_3)}, \quad \eta_{ri} = \frac{c_g(4\tau_{\text{Tee}}+8\tau_{\text{cross}})}{\omega\pi(l_5+2l_6+2l_7)}, \tag{8}$$

where the subscripts s, i and r denote the source side, intermediate and receiving side local mode subsystems. For instance, η_{si} is the coupling loss factor from the source side local modes to the intermediate local modes. τ_{Tee}, $\tau_{\text{Tee}C}$ and τ_{cross} correspond to the transmission coefficients given by Craik [12]. For the present case, all strips are assumed to have the same material and thickness and hence the values $\tau_{\text{cross}} = 0.0838$ and $\tau_{\text{Tee}} = \tau_{\text{Tee}C} = 0.149$ are found. c_g is the group velocity of the bending waves in the strips.

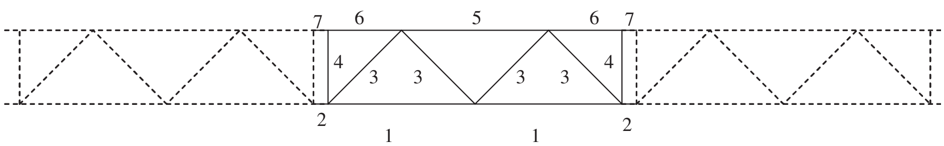

Fig. 4. SEA model of the joints for the coupling between local modes of extruded panels.

6. Coupling with air

The coupling between the extruded panel and the source and receiving rooms requires the coupling loss factors between the structure and an acoustic subsystem. In SEA this is modelled in terms of the average radiation efficiency of the structure, σ. The coupling loss factor from a plate to an acoustic cavity is given by

$$\eta = \frac{\rho c \sigma}{\omega m''}, \tag{9}$$

where ρ is the density of air, c is the sound speed in air and m'' is the mass per unit area of the plate. The coupling loss factor from a cavity to the structure can be obtained using the consistency relationships between subsystems in SEA, $n_i\, \eta_{ij} = n_j\, \eta_{ji}$.

The radiation of a double-skinned extruded plate is far more complicated than the well-known results for a rectangular plate. In the present SEA model, the coupling between global modes and air and that between local modes and air are considered separately.

6.1. Coupling between global modes and cavity

The radiation of global modes is similar to that of rectangular plates [13]. The critical frequency of global modes is usually quite low due to the high bending stiffness so that the radiation efficiency of global modes is generally close to unity for frequencies above 200 Hz [6].

6.2. Coupling between local modes and cavity

The radiation efficiencies of the two outer plate subsystems have been modelled in Refs. [6,14]. Considering an average of all possible excitation positions, the radiation efficiency of the one side of the panel is given by

$$\sigma_l = \frac{1}{S} \sum_i S_i \sigma_i, \tag{10}$$

where S is the area of the surface of the extruded plate, i under the summation indicates the ith strip on the surface of the extruded plate, S_i is the area of the ith strip, and σ_i is its radiation efficiency. In this it is assumed that the average vibration of each strip is equal and that the vibration of each strip is uncorrelated with the others.

The radiation efficiency of strips, which have extreme aspect ratios, is not well modelled using the usual model of rectangular plates [13] in the acoustic short-circuiting region. A set of approximate expressions has been given in Ref. [14] for various frequency regions, based on a study using a modal summation approach. These are used here.

6.3. Comparison with experimental results

The radiation efficiency of a sample panel under unbaffled conditions has been measured using a reciprocal test method [6]. This panel had dimensions 1.5×1.0 m and differs slightly in geometry from that used for the earlier results in Section 3. Results have been obtained corresponding to excitation positions on strips and on stiffeners. The results are compared in Fig. 5 with predictions from the model described above. The predicted radiation efficiencies for the panel include the contributions from global and local modes based on the modal summation approach. It is given by [6]

$$\sigma = \frac{\overline{W}_{\text{global}} + \overline{W}_{\text{local}}}{\rho c S \overline{\overline{\langle u^2 \rangle}}} = \frac{\sigma_g \overline{\overline{\langle u_g^2 \rangle}} + \sigma_l \overline{\overline{\langle u_l^2 \rangle}}}{\overline{\overline{\langle u_g^2 \rangle}} + \overline{\overline{\langle u_l^2 \rangle}}}, \tag{11}$$

where u_g is the velocity due to global modes, u_l is the velocity due to local modes and σ_g is the radiation efficiency of global modes. For this the measured velocities u_g and u_l are used. Simple supports of the plate

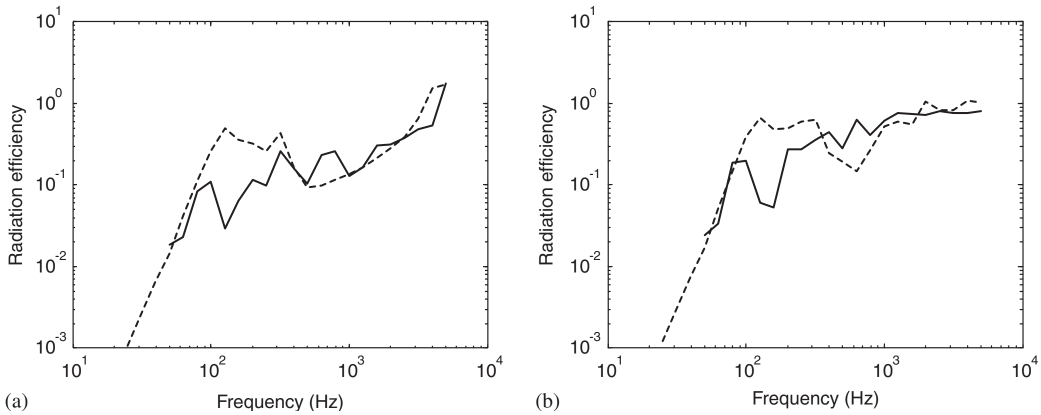

Fig. 5. Predicted and measured radiation efficiency of an extruded panel. —, Measured; – – –, predicted: (a) excitation on a strip; (b) excitation on a stiffener.

strips are used for the calculation of radiation efficiencies. This gives the cut-on frequency of local modes at 240 Hz.

In Fig. 5 the predictions agree with the measured results in most frequency bands. Both measurements and predictions show a lower radiation efficiency for excitation on a strip than on a stiffener. The dip found in the measurements at the first mode at 128 Hz is believed to be due to the cancellation between front and back under the unbaffled conditions; this is a twisting mode not accounted for in the model, but would also not be present in a vehicle.

7. Applications of SEA model

The proposed SEA models for extruded panels are used to predict the response of the panel for both mechanical and acoustical excitations. The results from SEA are then compared with experimental results.

7.1. Mechanical excitations

A set of experiments has been carried out on the same sample of extruded panel, which was forced at a point on its surface using an electrodynamic shaker. The response of the panel was measured using a scanning laser vibrometer. The spatially averaged transfer mobility obtained from vibration measurements has then been used to estimate the kinetic energy of the global mode subsystem and local mode subsystems. The average responses on either side of the panel correspond to the sum of the response contributions from global and local modes. The response scanned in the region of the stiffeners corresponds to the contribution from global modes. The response of local modes is therefore obtained by subtracting the contribution of global modes from the overall response. The interest here is to see whether the SEA model can give a reasonable prediction for the response on the two sides of the panel.

The coupling loss factors used are from the above models. Damping loss factors, obtained from measurements, of 0.045 and 0.1 are used for the global and local modes, respectively. The half-power bandwidth method was used to determine the modal damping loss factors at low frequency for global modes. The attenuation of vibration propagation with distance due to the damping effect was measured along the driven strip at higher frequencies and the damping loss factors in one-third octave bands for the local modes were derived from this [6]. Responses of the panel predicted from both three- and four-subsystem SEA models are then compared with the measurement results. For the three-subsystem SEA model, the predicted result for the receiving side of the panel (not shown here) is much lower than that from measurements [6,7]. After introducing the local mode subsystem for the intermediate connecting strips, the response of the panel is well predicted, as shown in Fig. 6(a). The vibration level difference across the panel is overestimated by the

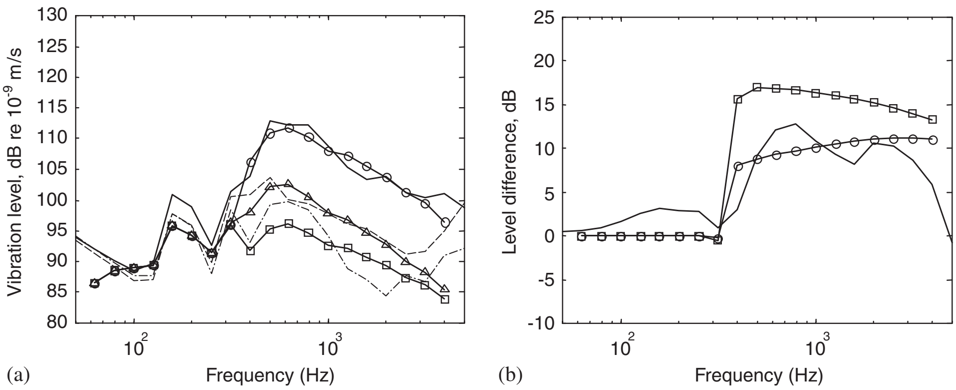

Fig. 6. (a) Predicted vibration levels of the extruded panel using four-subsystem SEA model for mechanical excitations. –○–, SEA-sending; –Δ–, SEA-receiving; –□–, SEA-global —, experiment-sending; – –, experiment-receiving; – · –, experiment-global. (b) Vibration level difference across the panel for mechanical excitations. –○–, SEA (four subsystems); —, experiment (mechanical); –□–, SEA (three subsystems).

three-subsystem model but more reliably predicted by the four-subsystem one (Fig. 6(b)). This shows that it is not sufficient to use only coupling from global modes to local modes to predict the response of the panel.

7.2. Acoustical excitation

The response of the panel under acoustical excitation in a transmission suite is predicted and compared with experimental data obtained from a vehicle manufacturer on a large sample of the same panel used in the above mechanical measurements [4]. From the SEA calculations, the spatially averaged sound pressure in the two rooms can be obtained. The sound transmission loss of the panel can therefore be calculated by

$$R = L_{p1} - L_{p6} + 10\log_{10}\left(\frac{S}{A_6}\right), \tag{12}$$

where L_{p1} and L_{p6} are the sound pressure level in the source room and receiving room, S is the area of the panel and A_6 is the total Sabine absorption in the receiving room.

By adjusting the input power into the source room, the sound pressure level for the source room is made the same as that from experimental data. This and the predicted sound pressure level in the receiving room are presented in Fig. 7(a). Both three- and four-subsystem SEA models underpredict the sound pressure level in the receiving room for frequencies above 315 Hz. There is not much difference between the results from the two SEA models. The sound transmission loss is therefore overpredicted compared with the measured one, as shown in Fig. 7(b).

The vibration response of the panel to acoustic excitation is also underestimated in the SEA models, as shown in Fig. 8. For frequencies below 250 Hz, where global modes are dominant, the SEA model gives a reasonable estimate. The local modes become dominant for frequencies above 250 Hz as seen in the measured vibration levels. The predicted vibration level for the local mode subsystem adjacent to the source room (subsystem 2) is much lower than the measured results. In the three-subsystem model for the extruded panel, the vibration level on the receiving side is close to the response of the global modes (not shown in Fig. 8(a)). Due to adding the intermediate local mode subsystem, the response of the local modes on the source room side and the receiving room side increases a little. The actual vibration level on the receiving side is higher than that of the global modes alone. This underestimation of the vibration of the panel leads to an overprediction of the sound transmission loss. The reason for the higher amplitude of the response of the extruded panel is not clear at this stage.

It has been shown in Ref. [6] that the non-resonant vibration due to airborne excitation is much lower than the resonant vibration for the values of damping loss factor used in the calculations. The non-resonant vibration, which is not contained in the vibration of the SEA subsystem, can therefore be neglected as the

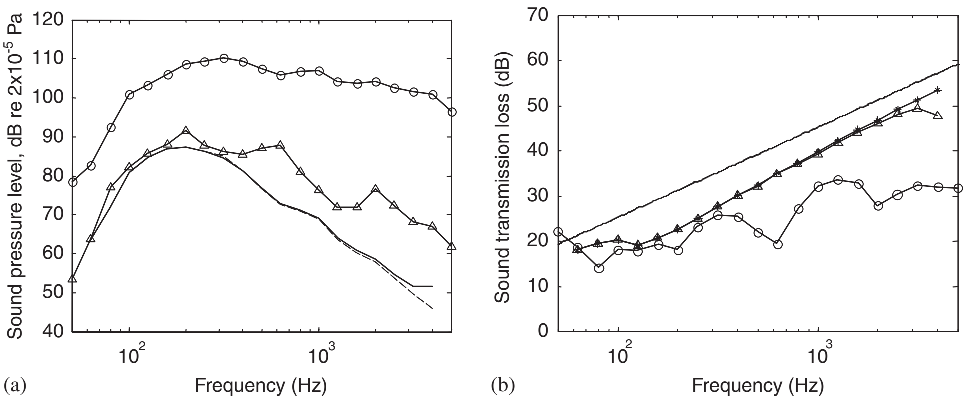

Fig. 7. (a) Sound pressure levels in two rooms predicted using SEA models. —, Four-subsystem SEA model; – –, three-subsystem SEA model; –○–, source room; –Δ–, measured receiving room. (b) Sound transmission loss predicted from SEA. —, Mass law; –○–, experiment; –Δ–, four-subsystem SEA model, –*–, three-subsystem SEA model.

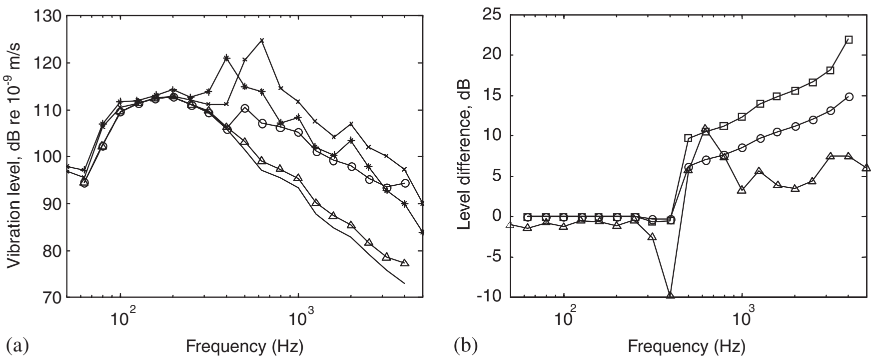

Fig. 8. (a) Predicted and measured vibration level of the extruded plate for the acoustical excitation in the four-subsystem SEA model. –○–, SEA sending panel; –Δ–, SEA receiving panel; –*–, experiment sending panel; – × –, experiment receiving panel; —, SEA global. (b) Vibration level difference between the two sides of the extruded panel predicted using SEA models. –○–, SEA (four subsystems); –Δ–, experiment (acoustical); –□–, SEA (three subsystems).

reason for the higher vibration amplitude. The damping of the extruded panel used in the SEA analysis is based on the experiments for the mechanical excitations on the smaller panel in Section 7.1. The values of the damping loss factor used for the extruded panels have a significant influence on the predicted results. Fig. 9 presents the results obtained using values of 0.1, 0.01 and 0.001 for the damping loss factor of local modes. It can be seen that the transmission loss decreases as the damping loss factor decreases. The result for a damping loss factor of 0.001 is much closer to the experimental result. However, from the vibration level differences across the panel using different damping loss factors, as shown in Fig. 9(b), it appears that a damping loss factor higher than 0.01 is required to give the best prediction of the level difference. To obtain better understanding of the predicted results from the SEA model, more accurate damping values are required. The actual radiation efficiency for the case of acoustical excitation also appears higher than that measured for mechanical excitation [6]. This also has effects on the response of the panel. Therefore, a more accurate model for the radiation efficiency under acoustical excitation is also required.

8. Conclusions

A modelling approach for the vibroacoustic behaviour of aluminium extrusions used in railway vehicles has been presented in this paper. Extruded panels are represented using global and local mode subsystems. An

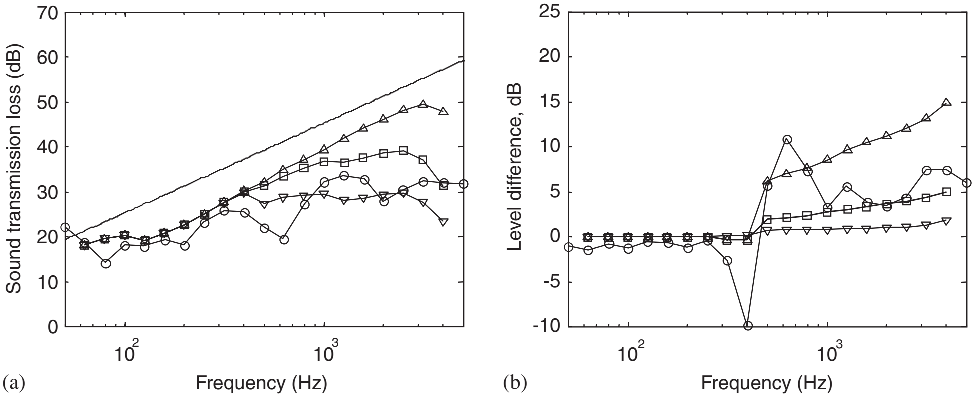

Fig. 9. (a) Sound transmission loss predicted using different damping loss factor for local mode subsystems; (b) vibration level difference of the two sides of the extruded panel predicted using four-subsystem SEA model with different damping loss factors for the local mode subsystems. $-\Delta-$, $\eta = 0.1$; $-\square-$, $\eta = 0.01$; $-\nabla-$, $\eta = 0.001$; $-\circ-$, measurement.

approximate model for the modal density of extruded panels has been developed and verified using an FE model. The coupling between global and local modes is approximated by a travelling global wave which drives the edges of the local subsystems. The interactions between local modes are modelled using standard couplings between plates joined at various structural joints. For the coupling with air, the average radiation efficiency of a baffled extruded panel is modelled in terms of the contributions from global and local modes. SEA models of extruded panels consisting of global mode and local mode subsystems are proposed. These models enable the response difference across the panels to be predicted. The predicted responses of the panel under mechanical and acoustic excitations have been compared with experimental results. The proposed SEA model gives a good prediction for mechanical excitation. The sound transmission loss of extruded panels has not been so well predicted with the present SEA model. This requires further investigations on the structural vibration of extruded panels and their sound radiation.

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