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MODELLING THE VIBRATIONAL CHARACTERISTICS AND RADIATED SOUND POWER FOR A Y25-TYPE BOGIE AND WAGON

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A theoretical investigation has considered the vibrational characteristics of the bogie and wagon to quantify the noise freight wagons. This study presents the analysis of the bogie by finite element methods (FEM) to cover the lower-frequency range. The wagon has been analyzed using a statistical energy analysis (SEA) approach. The accelerances on the bogie have been calculated and the sensitivity to excitation evaluated. The calculation of sound power for force inputs has also been determined. Complementary experimental results show the predictions to be in good agreement. The wagon has not been examined specifically to identify detailed response and radiated sound; instead it is possible to consider the response of larger identifiable structural areas such as the doors, walls, roof, etc. The response of these larger areas can also be used to predict the transfer functions from radiated sound power for forces applied at the connection points. Good agreement has been obtained with experimental measurements.

1. INTRODUCTION

This study covers the analytical investigation of the response of a Y25-type bogie (Figure 1) on to which is attached a wagon (Figure 2) of type tombereau. The comparison with the experimental findings [1] concentrates on the predicted response of the point and transfer accelerances on the bogie, and the response of the wagon to force and moment excitations at a location where the wagon connects to the bogie.

Various analytical methods were considered. The two that were chosen are:

1. Finite element modelling (FEM) [3, 4]

An FE model of a bogie can be used to predict the modal characteristics, (for example, natural frequencies and corresponding model shapes) and, knowing damping, the forced vibration response. Whilst it is feasible to model the bogie, the wagon is too complex and a statistical energy analysis has been performed.

2. Statistical energy analysis (SEA) [5]

The structure is described by a number of subsystems that are connected together. Whilst the subsystems are close representations of the actual components, there is a lack of detail and the responses are predicted in a band averaged ($\frac{1}{3}$ octave) and spatially averaged sense. A typical requirement for statistical confidence is that each subsystem should, within the chosen frequency bands, exhibit of the order of five or more modes as an individual structure. For the lower frequencies, the finite size and thickness, and hence stiffness, mean that such an approximation would not be realistic below 500 Hz for the bogie frame whilst for the wagon it is likely that this restriction is reduced to below 50 Hz. Hence, the SEA model of the wagon has been considered for the typical frequency range of 50–5000 Hz. A commercially available SEA software package [6] was used.

2. ASSEMBLY OF PREDICTIVE MODELS

2.1. FE MODEL OF THE BOGIE

The models generated used four-noded quadrilateral shell elements (SHELL 63 in ANSYS [3]) and a mesh density more than sufficient for the frequency range examined. The suspension and braking mechanisms were too complex to be analyzed theoretically. Hence, the analysis here has focussed on examining the bogie frame above the suspension.

To create a realistic simplified model, it has been necessary to reduce the amount of structural detail that is modelled. Likewise, given the limitation on the size of the numerical model that can be analyzed, it has not been appropriate to model the whole of the bogie frame using a FE representation. A quarter model of the bogie frame has been created instead and one uses the assumptions that there are planes of symmetry about two perpendicular vertical planes passing through the centre of the bogie. These planes are in the fore and aft direction and in the lateral direction. The view of the bogie in Figure 1 shows that the structure with the braking mechanisms, etc., is not exactly symmetric. The reduction using this approach, and

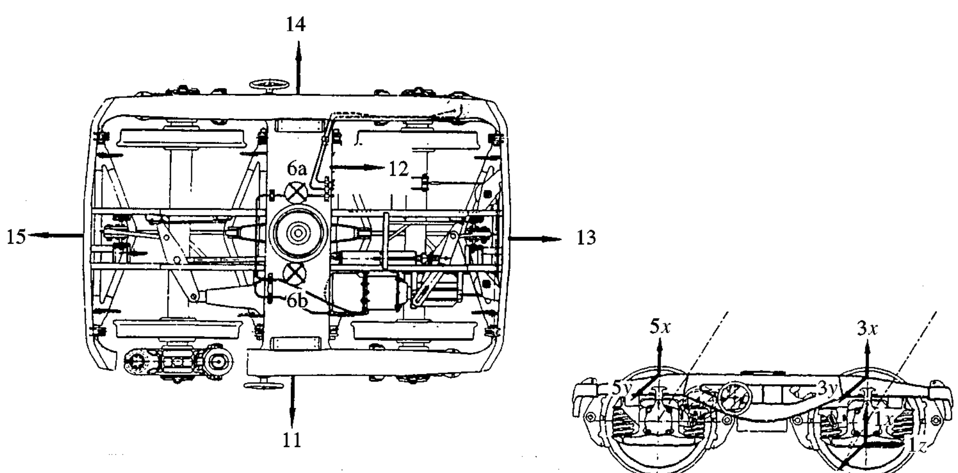

Figure 1. Plan and side view of a Y25-type bogie.

applying symmetric or antisymmetric boundary conditions on these planes, reduces the problem and the modes of the whole structure are then the total number of modes of the individual models. Whilst there may be some minor differences between the predicted and actual natural frequencies, these differences will not significantly alter the $\frac{1}{3}$ octave band responses except for the lowest order bands, where the modal density is low.

The modes for the four models, below 1 kHz, were calculated and used in a modal superposition method to calculate the forced response, at discrete frequencies, over the frequency ranges of 25–500 Hz third octave bands. Measured modal loss factor equal to 0·02 (or 2%), was adopted. The forced response was calculated for each model separately and then combined to give the total response of the bogie. (See reference [4] for an explanation of applying non-symmetric loads, such as at one suspension point, on a symmetric structure).

The results were calculated at measured degrees of freedom with additional points on various sections. Extra responses were subsequently used to calculate estimates of the mean square velocity on the sections and radiated sound power contributions using approximate radiation efficiencies for beam and plate-like structures.

2.2. SEA MODEL OF THE WAGON

An overview of the wagon (Figure 2) shows the main structural details:

(i) steel corrugated side panels (3920 mm × 2175 mm × 3 mm)
(ii) steel corrugated doors (2070 mm × 2300 mm × 3 mm)
(iii) aluminium rolling roof (12 000 mm × 2690 mm × 2 mm)
(iv) steel front and rear end panels (2690 mm × 2200 mm × 3 mm)
(v) steel sills and posts
(vi) wooden (oak) floor (12 000 mm × 2690 mm × 45 mm)
(vii) steel subframe supporting the floor and attached to the bogie.

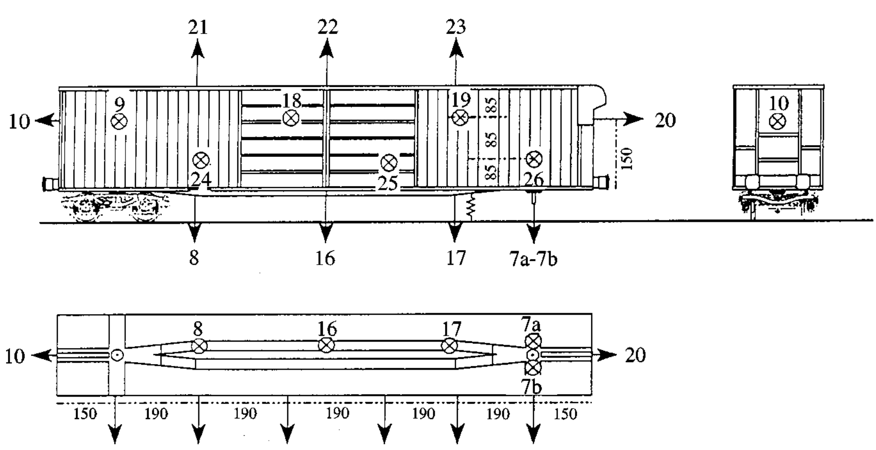

Figure 2. Side and end view of the wagon.

The SEA model is shown in Figures 3 and 4. The subframe model is attached to the main wagon body model at points where there is a common structure; for example, at the sills. The subframe is stiff and the beam subsystems used have low modal densities at low frequencies. Hence, the predicted behaviour of the framework may

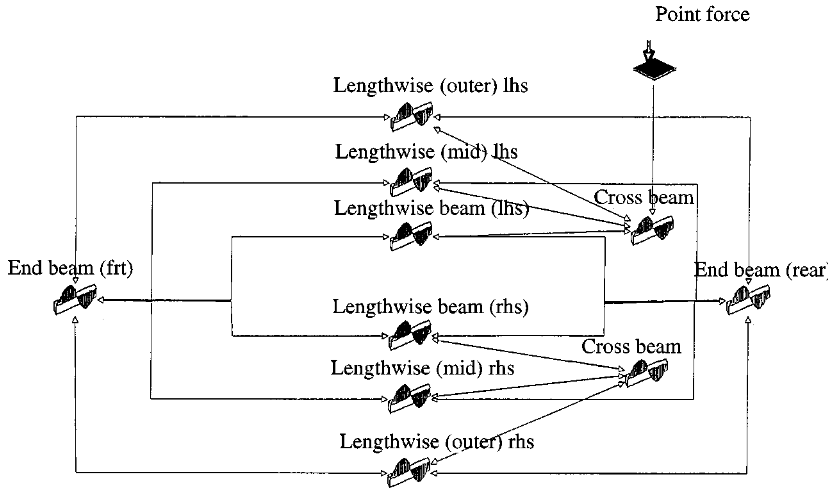

Figure 3. SEA model of the wagon subframe.

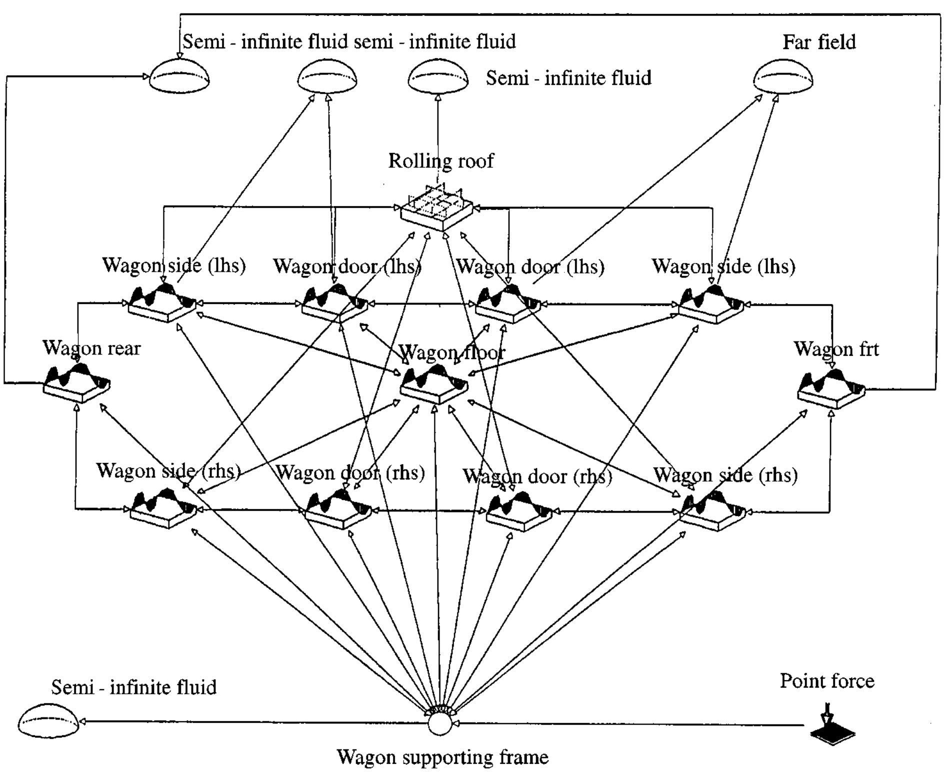

Figure 4. SEA model of the wagon.

have a low statistical confidence. Another limitation relates to the mounting points of the wagon on the bogie, at which the excitation forces apply, which are at a junction of many structural components.

The main corrugated panels and doors are highly orthotropic. One possible model is an orthotropic plate subsystem, as the stiffness in the two orthogonal directions can be estimated. This type of subsystem has not been implemented as the connection and the coupling strength of corrugated or orthotropic plates would require more extensive investigation. It is also not appropriate to model the corrugations (wavelength 490 mm, 200 mm wide top and bottom and corrugation depth 62 mm) using a rib-stiffened plate model. For this study equivalent isotropic plates have been adopted. The effect of the increase in the stiffness, due to the corrugations, will be to decrease the modal densities at low frequencies and also result in different coupling loss factors. As frequency increases, due to the size of the corrugations, the plate will have a modal density that approaches the flat plate result. The radiation efficiency of such a configuration will be higher than a flat plate of the same thickness as there will be a lower critical frequency in the direction parallel to the corrugations due to the higher stiffness in that direction. The acoustic radiation from corrugated panels will be strongly dependent on both frequency and direction, and these results, based on simple radiation models, will not be able to reflect this.

3. PREDICTIONS AND COMPARISONS WITH MEASUREMENTS

3.1. FINITE ELEMENT PREDICTIONS (BOGIE)

These results can be classified as point and cross accelerance above the suspension, transfer accelerance across the bogie, including to the wagon connection point, and radiated sound power per unit squared force. In each case measurements [1], up to 5 kHz, on a non-rolling bogie are shown.

Figure 5 shows the point accelerance above the suspension point to axial and vertical excitation respectively. The agreement for the axial direction is very good. The structure is more responsive in this direction than in the vertical direction. The vertical point accelerance is in good agreement in the 125–500 Hz third-octave bands. Below 125 Hz the structure has few modes and this leads to a large fluctuation in the FE predictions in third octave bands. The point cross accelerance predictions, not shown, are in reasonable agreement with the measurements reasonable, except at lower frequencies.

The transfer accelerance to points on the cross member (point 12), front beam (point 13) and side wall (point 14) are shown in Figures 6(a) and (b). Below 125 Hz there is some overprediction of the transfer accelerance, but above 25 Hz the agreement is much improved with generally a higher response due to axial excitation in both measurements and predictions. The transfer accelerance to the connection point area shows similar agreement and for axial excitation follows the measured trend well.

The predicted total power radiated for the axial force compared to the measurements is shown in Figure 7. The prediction includes the cross member as well as the

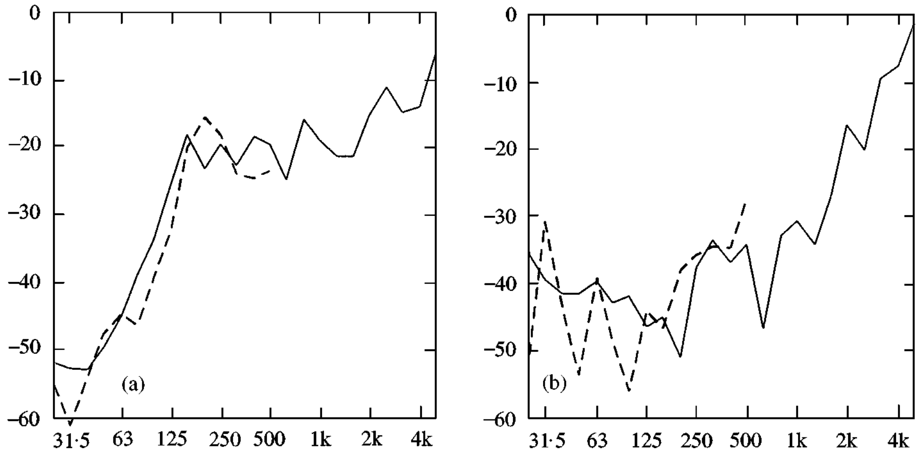

Figure 5. Comparison of the measured and predicted point accelerance for axial excitation and response (a) and vertical excitation and response (b) above the suspension point: ——, measured; - - - - -, predicted.

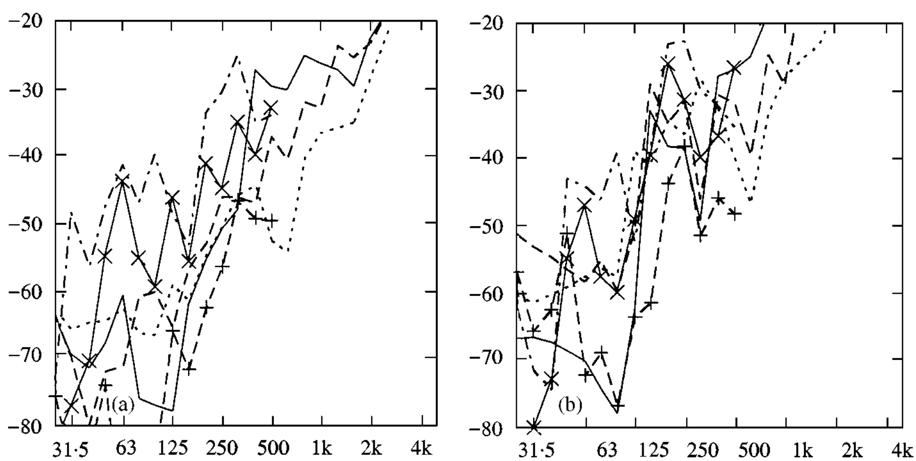

Figure 6. Comparison of the measured and predicted transfer accelerances for vertical excitation (a) and axial excitation (b) above the suspension point: ——, cross member; -----, front beam; and – – –, side wall measured. – · – ·, cross member; —×—, front beam; and —|—, side wall predicted.

side and end sections. This agreement is much better than for the individual bogie sides and ends, except below 63 Hz where measurement limitations may exist. The radiation efficiency values used [1] are very low at the lower frequencies where short-circuiting can occur around the beam-like elements.

3.2. PREDICTIONS OF THE WAGON RESPONSE

The SEA subsystem response is assumed to be uniformly distributed; the response at individual locations cannot therefore be investigated. Further refinement,

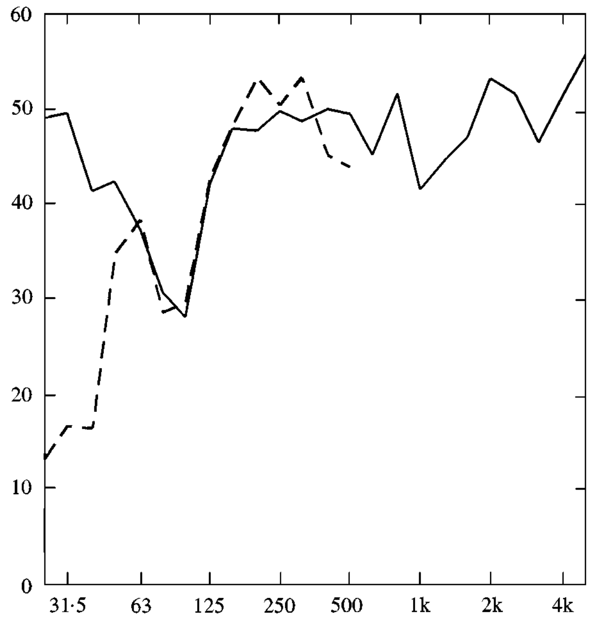

Figure 7. Comparison of the —— measured and – – – – predicted total bogie sound power transfer functions for an axial force.

by splitting a subsystem into a number of smaller subsystems does not usually produce any improvement. Some of the experimental results, especially for the framework, show an attenuation of response with distance from the applied source and this effect cannot be replicated with the type of model under consideration.

3.2.1. *Prediction of the frame response*

Figures 8(a) and (b) show the measured point accelerance on the subframe, to force and moment excitation respectively, compared with the predictions on the main lengthwise frame at the connection point. The predictions are based on the theoretical formulae for the point impedance of infinite beam structures [2] and is generally valid at the higher frequencies where the finite length and boundary conditions become less important.

For force excitation the accelerance increases at a rate of 8·4 dB per octave in the measurements and 3 dB per octave in the predictions. This indicates that the connection point is less stiff, but at low frequencies there may be significant structure governing the low-frequency response. Below 500 Hz the predictions are higher than the measured response and above 1600 Hz the predictions are lower than the measurements. For moment excitation there is very good agreement with the measured point accelerance over the whole frequency range.

3.2.2. *Prediction of the wagon body response*

The results are derived from predictions in which the power input to the wagon is calculated using an input impedance based on a model of the two main beams

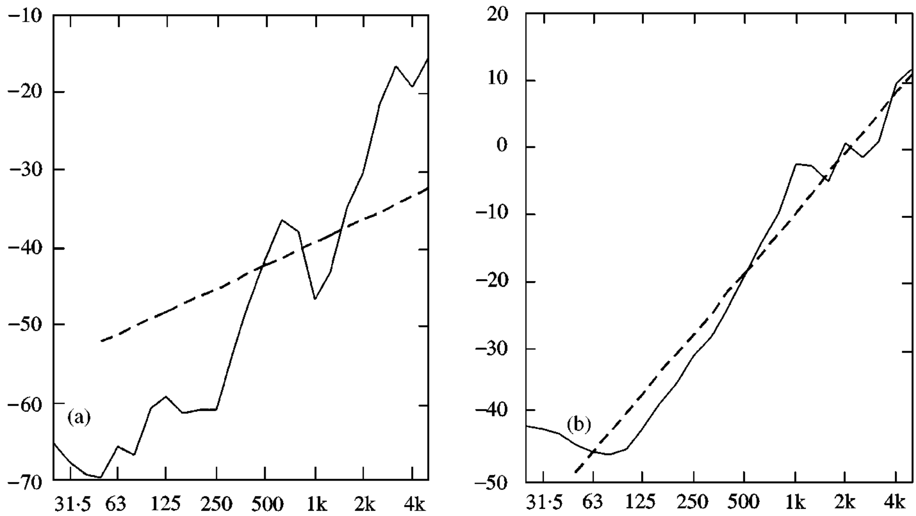

Figure 8. (a) Accelerance of bogie mounting point for vertical force input. (b) Angular accelerance of bogie mounting point for moment input: (——, measured; – – – predicted).

connected together; i.e., a modified prediction which replicates more realistically the connection point and produces more accurate estimates of the power input levels to the subframe.

3.2.3. *Side panels*

The SEA model predicts similar levels for all the side panels of the wagon. The results for the side wall are given in Figures 9(a) and (b) for force and moment excitation respectively. The overall levels and general trend versus frequency are predicted correctly for both cases. There is a higher response to unit moment excitation compared with a unit force excitation.

3.2.4. *Doors*

The main problem has been the connection of the doors to the sills and side walls. The simplest idealization, using a number of point connections, was adopted. The predictions, given in Figure 10, are good. Given these assumptions it is not surprising that the measured higher-frequency response is lower than that predicted as it is likely that the doors are very poorly coupled at these frequencies.

3.2.5. *Rolling roof*

Predictions for the roof modelled as a stiffened plate (Figure 11) follow the same trends as the measurements but are generally 10–20 dB higher at the higher frequencies. The most likely explanations are that the coupling between the roof and side walls, etc. is less strong than modelled or that the stiffened plate formulation is not appropriate and that the roof is responding as a set of loosely connected plate subsystems. There is, however, a reasonable fit to the measured trend.

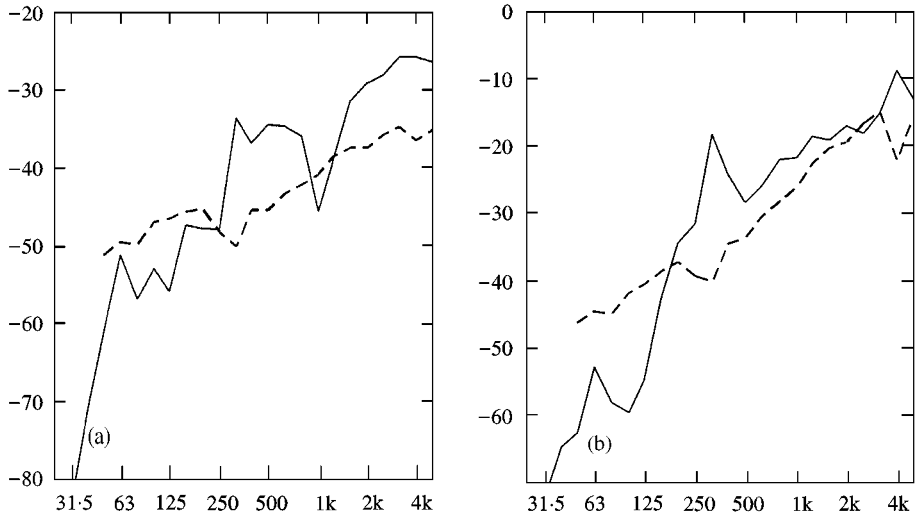

Figure 9. (a) Response of side wall for unit vertical force input at bogie connection point, (b) for unit moment input at bogie connection point: measurement at one point compared with prediction of spatially averaged response; (—— measured; – – – predicted).

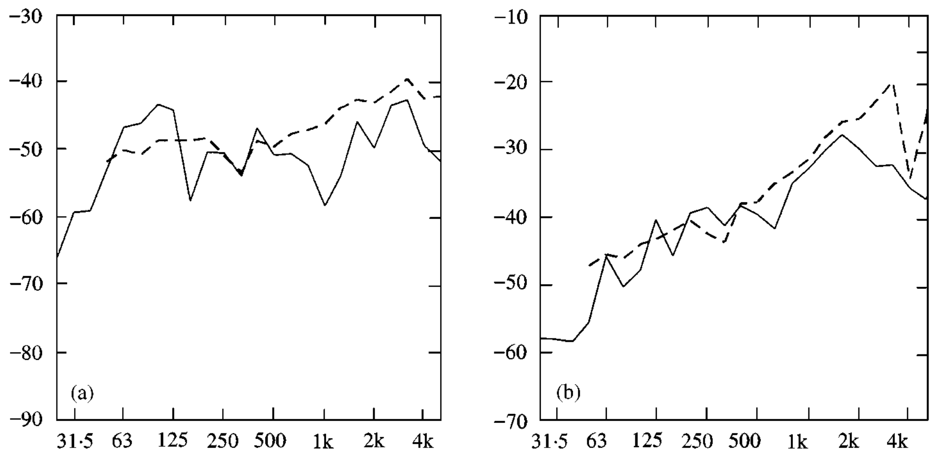

Figure 10. Response of wagon doors for (a) unit vertical force and (b) unit moment at bogie connection point (—— measured; – – – predicted).

3.2.6. *Front and rear ends*

The comparisons with the measurements are given in Figures 12(a) and (b) for force and moment excitation respectively. There appears to be a significant variation with frequency in the measured response. This may be caused by the stiffened nature of these ends, but the stiffening is not regular and therefore not easily amenable to modelling. Generally, the predictions are a little lower than measurements but still reasonable.

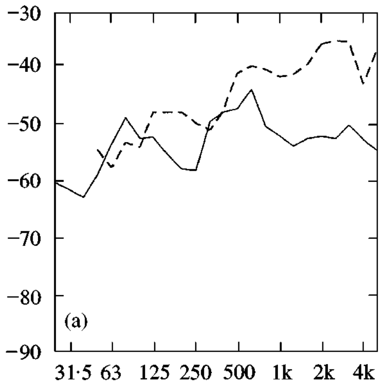

Figure 11. Response of wagon roof for (a) unit vertical force and (b) unit moment at bogie connection point (—— measured; — — predicted).

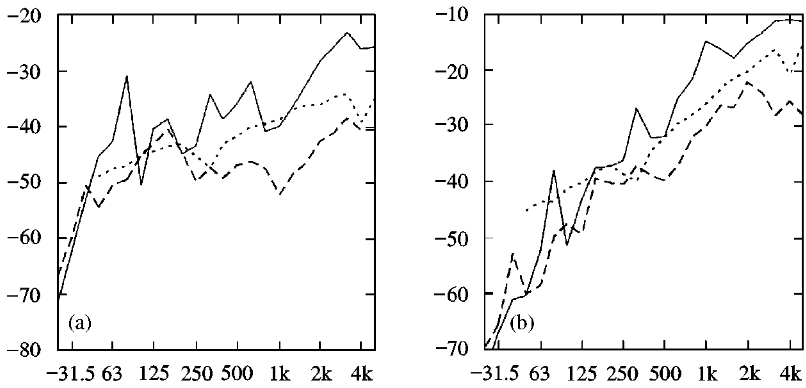

Figure 12. Response of wagon end walls for (a) unit vertical force and (b) unit moment at bogie connection point (—— measured rear wall; — — measured from wall; - - - - - predicted).

3.3. PREDICTION OF THE FORCE TO SOUND POWER TRANSFER FUNCTION

Figures 13(a–d) show the predictions of the sound power contribution for a unit force applied to the wagon mounting point for sound radiated by a side wall, door, the whole roof and the end walls respectively. Each relevant subsystem is assumed to radiate into an acoustic half space. The radiated power is dependent upon the radiation efficiency [2] calculated using the radiating area, perimeter and type of the subsystem. The radiated power is given by the product of the radiation efficiency, the acoustic impedance of air, the radiating surface area and the mean square velocity, itself calculated to include the acoustic radiation losses.

The predictions are in good agreement with the measurements for the side wall and the door and both structures show a significant increase above 2 kHz. This is likely to be due to an increase in the radiation efficiency at this frequency rather than due to an increase in the response. The critical frequency of a 3 mm steel plate is approximately 4 kHz, which can be seen as a peak in the predicted results in Figures 13(a–d). The power radiated from the roof (Figure 13(c)) is greatly

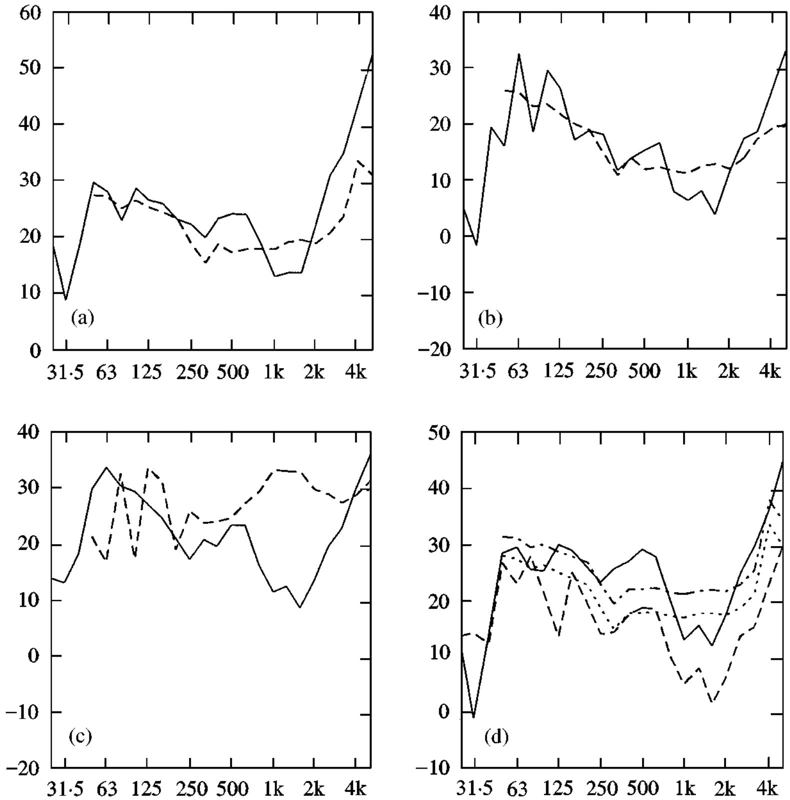

Figure 13. Sound power radiated from (a) right-hand side wall, (b) wagon door, (c) roof and (d) end walls, for unit vertical force input of bogie connection point; —— measurement for right-hand side compared with ······ predictions for (a)–(c) —— and – – – for measured on front and rear ends compared to ······ and – · – · predicted respectively.

overpredicted above 500 Hz. This discrepancy corresponds to the differences in vibration response already seen for this component. The predictions for the sound power transfer function for the end walls show a similar level of agreement to those for the side wall and door. The reason for the differences found (for the wall) is partly due to the underprediction of the response in the mid frequencies, 250–800 Hz (see Figure 12).

3.4. THE PREDICTION OF SOUND PRESSURE DUE TO OPERATIONAL FORCES

The response of the wagon has been calculated using the SEA model, with the force and moment at the bogie connection point set to the values equal to the equivalent forces calculated from the measurements [1]. The acoustic powers are assumed uncorrelated and are simply added together. The sound pressure at the trackside location, 7·5 m from the track centreline and at a height of 1·5 m above the rail head was evaluated using a simple radiation model, based on monopole radiation directivity. Adjustment is made for the radiation efficiency of corrugated panels using the simple factor $1 + 2P/B$ to multiply the radiation efficiency below the critical frequency, where P and B are equal to the total length of the corrugations and B the total panel perimeter. The effects of ground reflection (+ 3 dB) and the forcing from the second bogie (+ 3 dB) are also taken into account in the predictions. The comparison between the predictions based on experimental hybrid and reciprocal appraoches [1] and the SEA prediction is given in Figure 14. Between 500 and 2500 Hz the SEA model is in excellent agreement with the reciprocal measurement approach. The difference between the approaches is of the order of 5 dB below 500 Hz, where the SEA model underpredicts the pressure, and above 2·5 kHz the SEA model overpredicts compared to the measurement-based estimates. There is significant response and radiated sound due to the equivalent moment above 2 kHz and from a predictive approach this cannot be ignored.

4. CONCLUSIONS

The finite element model of the simplified bogie has shown reasonably good agreement in predicting the dynamic response up to and including the 500 Hz third-octave band. It would be feasible to improve the agreement at the lower frequencies by increasing the amount of detail modelled. The lower-frequency bands have very few modes and the natural frequencies are sensitive to detail.

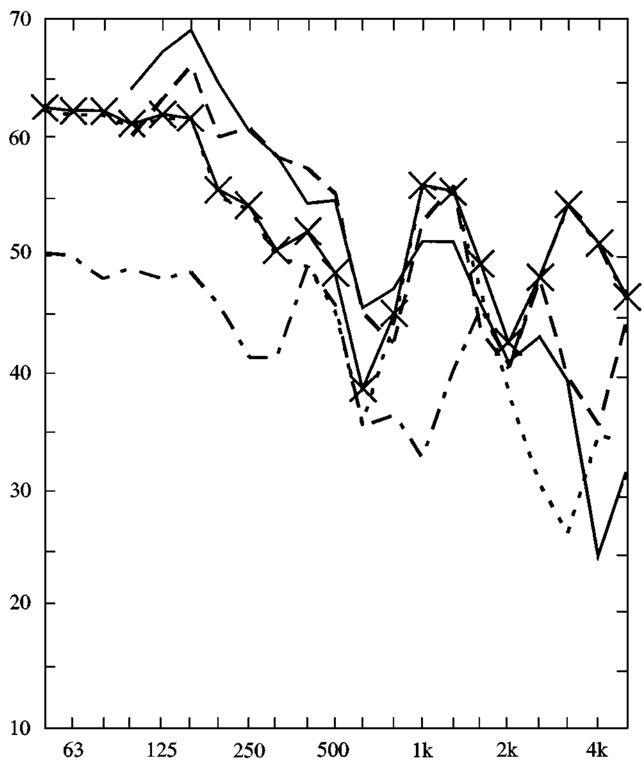

Figure 14. Comparison of the sound pressure level for the wagon body from measurement based approaches, —— hybrid and – – – reciprocal, and SEA predicted, using the ····· equivalent force, – · – · – moment —×— and combined.

Predictions of the sound power radiated by the bogie have been made, but it is difficult to extract the power radiated from the individual components in the measurements and the latter was restricted to axial excitation. The total power radiated, summed over the experimental measurements and for the predictions for the sides and ends, is in very good agreement and the effect of the differences for the individual components is reduced. Below the 63 Hz third-octave band there may also be some reduced confidence in the sound power measurements.

The results of the simplified SEA model show reasonable agreement with the measured responses and sound powers for a unit force or moment excitation at the connection point to the bogie. For SEA models a difference of less than 10 dB between the predictions and the measurements, can be considered to be good and for most of the wagon structure better than this has been obtained. The main limitation on the predictions and the approach has been the limited detail on the wagon subframe. This resulted in an overprediction of the subframe response although from other responses it appears that the input mechanical power to the models is reasonable. One development to overcome this is to have a more detailed model of the mounting point, possibly using a FE representation, to predict more accurately the input mobility and resulting power input.

The evaluation of equivalent forces, between the wagon and bogie, from running measurements on similar wagons and bogies will in future, allow comparison and predictions of the noise from other types of wagon structures to be assessed without the need for extensive experimental studies. Also it will be possible to identify the effect of design changes to existing configurations and quantify the actual contribution of superstructure noise to the noise levels measured in pass-by tests where the rolling noise may dominate the measurements.

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