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DEVELOPMENT OF AN IN-SERVICE DYNAMIC MODEL OF A DOUBLE DECK PASSENGER TRAIN

David Hanson[1], Mark Winton[2], Ross Emslie[1], Graham Brown[1], Bob Randall[3]

[1]Structural Dynamics Group, Sinclair Knight Merz,
St Leonards, Sydney 2065, Australia
[2] Structures and Dynamics, United Group – Rail, Broadmeadow, Australia
[3]Mechanical and Manufacturing Engineering,
The University of New South Wales, Sydney, Australia
E-mail: dhanson@skm.com.au

Abstract

This paper presents the findings of an investigation into the structural properties of a double deck passenger railcar. The objective of the investigation was to develop an in-service dynamic model about an operational point where the car was travelling at 80km/h over track used in everyday service. A detailed grid of response measurements, providing a spatial definition of the car, were recorded consisting of nearly three hundred accelerometer records in three axes distributed over the structure in more than forty measurement batches while the car was running down the track. Using these measurements, an in-service modal model was constructed by operational modal analysis using the enhanced frequency domain decomposition technique.

Of particular interest to this investigation were the primary vertical and lateral bending modes which were found to be coincident, with the lateral mode dominating. The application of whole body vibration weightings however, produced an equivalence of the largely decoupled vibration due to each mode. In this context, the in-operation modal properties are compared with those predicted by a workshop modal test and the operational deflection shapes of the car. The relative merits of these approaches for the modelling of the car in-service are discussed.

It is intended that the in-service dynamic model can be used in the prediction and refinement of ride comfort.

1. INTRODUCTION

The modal properties of a railcar, the natural frequencies, modal damping and mode shapes, have traditionally been obtained using Experimental Modal Analysis (EMA), in which the responses of the structure to a contrived and measured input, such as impact hammer excitation, are recorded. Techniques for EMA have been well documented (for example [1]) but face a fundamental problem, which is that the excitation rarely represents the force distribution, magnitude and type of forcing the railcar will experience while in operation. Moreover, the properties of the railcar might also be different in service, for example because of the effects of aerodynamic or passenger loading. One example related to the rail vehicle described in this paper is that the bogie suspension of the train is known to be non-linear and so will behave differently running down the track than when stationary in the workshop. The result is that the modal properties estimated from EMA may be poor estimates of the train's in-service behaviour.

Ideally, to obtain properties that apply when the railcar is in service, EMA would be performed under operational conditions. Unfortunately it is not possible in this situation to measure the excitations, only the responses, so traditional input-output modal analysis techniques cannot be applied. Instead, a blind identification technique is required, so called because it uses only the responses. When applied to measurements obtained in service this is termed Operational Modal Analysis.

Operational Modal Analysis (OMA), blind identification applied to mechanical systems, has become increasingly popular in recent years. New techniques have become available which allow the properties of a system to be identified for the case when the system is in service, thereby ensuring that the properties identified correspond with the working environment of the system. Some popular techniques which are available in commercial software packages include Frequency Domain Decomposition (FDD) [2, 3], Stochastic Subspace Identification (SSI) [4, 5] and Operational PolyMAX [6]. This paper presents an application of the enhanced FDD algorithm to the operational modal analysis of a double-deck passenger car and compares the modal model with models obtained by other traditional techniques.

2. THEORETICAL OVERVIEW

2.1 Enhanced Frequency Domain Decomposition

The Enhanced Frequency Domain Decomposition (EFDD) algorithm for OMA employed in this paper is part of the Bruel & Kjaer Pulse® commercial software suite. For completeness, the EFDD technique is summarised below, but interested readers are encouraged to refer to[2] for a detailed overview.

At a particular frequency ω, only a few modes $sub(\omega)$ will contribute to the response significantly, allowing the cross power spectral density of a lightly damped structure to be expressed as:

$$\boldsymbol{G}_{lm}(j\omega) = \sum_{k=sub(\omega)} \frac{d_k \varphi_{lk} \varphi_{mk}^T}{j\omega - \lambda_k} + \frac{\bar{d}_k \bar{\varphi}_{lk} \bar{\varphi}_{mk}^T}{j\omega - \bar{\lambda}_k} \tag{1}$$

where $\boldsymbol{G}_{lm}$ is the cross power spectral density between responses at locations l and m, d_k are scalar constants related to the modal participation factors, φ_k are the mode shapes, λ_k are the complex resonance frequencies, and the over bar $(\bar{\bullet})$ represents complex conjugate. A matrix of auto and cross spectral density functions can be decomposed at each frequency using singular value decomposition to yield:

$$\hat{\mathbf{G}}(j\omega_i) = \mathbf{U}_i\mathbf{S}_i\mathbf{U}_i^{\,H} \tag{2}$$

where $\mathbf{U}_i$ is a matrix of singular vectors and $\mathbf{S}_i$ is a diagonal matrix of singular values at frequency ω_i. If only one mode is dominating at a peak in a plot of the largest singular value then only one mode is significant in (1) and the first singular vector is an estimate of the mode shape, i.e. $\hat{\varphi} = \boldsymbol{u}_{1i}$. A section around this peak where this mode can be said to dominate, i.e. where the modal assurance criteria [1] between $\hat{\varphi}$ and the singular vectors $\boldsymbol{u}_{1\hat{i}}$ at nearby frequency bins $\hat{i}$ are above a certain threshold (say 0.8), can then be selected. Converting this section of the largest singular value back to the time domain by inverse Fourier transformation, the natural frequency and damping can be determined by curve fitting the zero crossing times and logarithmic decrement respectively of the resulting autocorrelation function.

2.2 Applying the EFDD Technique to a Railcar

The EFDD algorithm is based on the assumption that the system is subjected to excitation that is both frequentially and spatially white, i.e. the autospectum of the input is not a function of frequency and the input is evenly distributed across the structure. As an approximation, it is assumed that this generally applies to the case of the railcar with the dominant excitation sources, the track induced vibration and aerodynamic loading both being frequentially white in the frequency range of interest (for example [7]), and the overall levels of excitation being distributed across the car body.

2.3 Test Methodology

The responses on the structure of the car were measured in 45 batches using six hot-glue mounted triaxial accelerometers. The measurement locations are shown in Figure 1 where they can be seen to span the upper and lower deck floor, cabin roof, and vestibule roof and floor. Two measurements were also recorded on one of the bogies at two opposing diagonal corners (designated 9991 and 9992 below).

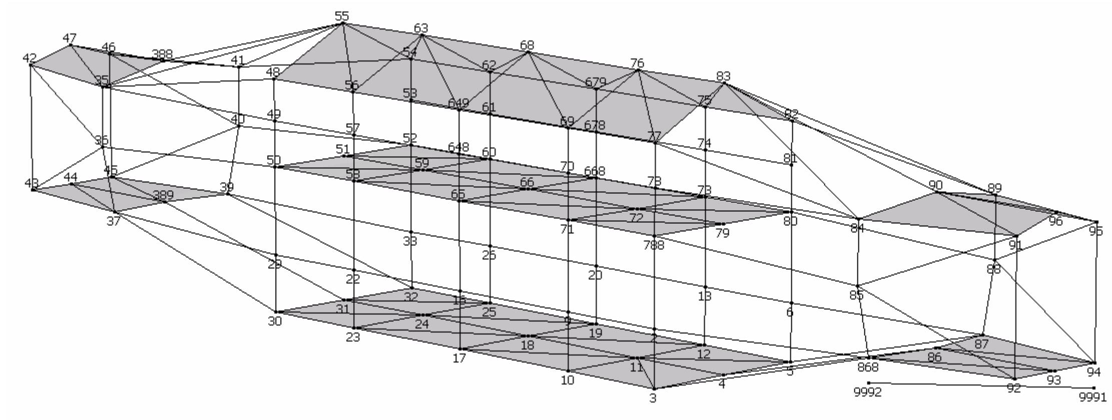

Figure 1 Measurement locations on the railcar

The measurements of between three and four minutes were recorded onto digital audio tape at a sampling frequency of 1000Hz. The measurements were downsampled to 125Hz before processing.

3. RESULTS

3.1 Operational Modal Analysis

The modal properties identified through the OMA are contained in Table 1. The modal parameters were estimated for each measurement batch, producing a statistical distribution. The standard deviations of these distributions are also included in Table 1 and it is interesting to note the degree of uncertainty in some of these parameter estimates. Principal among these is the whole body transverse shear mode which exhibits a standard deviation of 0.4Hz. This distribution of parameter estimates is due largely to small changes in the structure between measurement sets, including the location of the test personnel, the non-stationarity of the excitation highlighting small non-linearities in the car, and possibly other environmental factors.

The results of a workshop experimental modal test are also included in Table 1. The workshop modal test failed to excite the pitch mode or the lateral bending mode. A comparison of the other modes shows significant differences between the resonance frequencies of the stationary car compared to the car on-track. This comparison highlights the importance of OMA, in that the *in-service* modal properties are estimated, compared to a workshop test.

Table 1 Modal properties of the railcar

Mode	OMA				EMA
	Frequency (Hz) *	Std. Dev. Frequency (Hz)	Damping (%) *	Std. Dev. Damping (%)	Frequency (Hz)*
Pitch	0.91	0.15	22	7	-
Bounce	1.5	0.1	26	6.7	3
Lateral Bending	13.7	0.17	3.5	1.2	-
Vertical Bending	13.7	0.18	2.9	1.2	16
Whole Body Shear	15.3	0.42	3.2	1.2	17

* Frequencies and damping have been distorted by a constant scaling factor to preserve commercial confidentiality

Of primary interest in this investigation were the bending and shear modes. The mode shapes corresponding to these modes are contained in Figure 2- Figure 4.

It is interesting to note that the bending modes appear to be coincident, although the distribution of their estimates may mean that they are distinct in any particular measurement batch. This was not immediately evident from the operational deflection shapes because the lateral bending mode dominated the vertical bending mode. This will be discussed below.

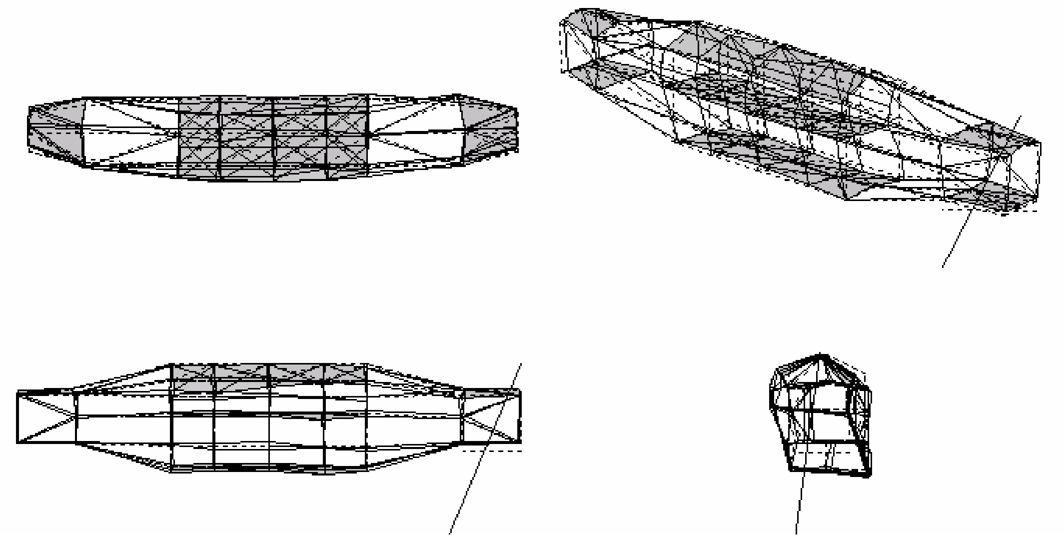

Figure 2 Lateral Bending Mode

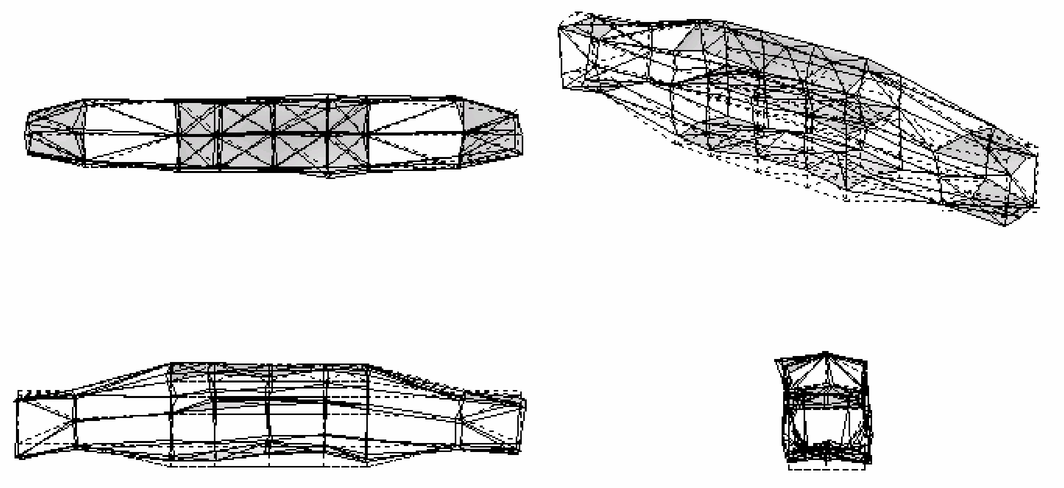

Figure 3 Vertical Bending Mode

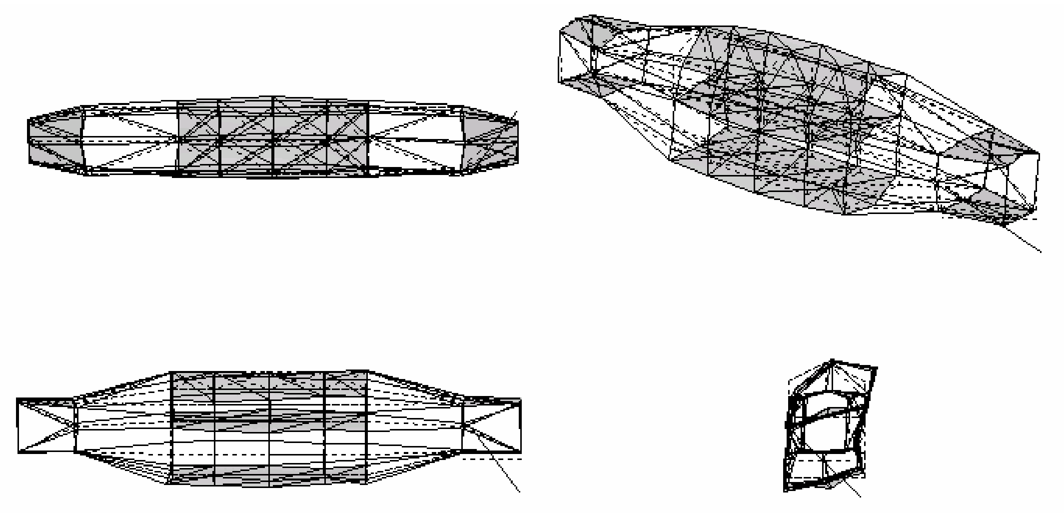

Figure 4 Whole Body Shear Mode

3.2 Operational Deflection Shapes

A commonly employed tool for assessing the in-service behaviour of the railcar is operational deflection shape (ODS) analysis. This analysis is based on so called ODS Frequency Response Functions (FRFs) which derive their magnitude from the auto-spectrum of each response, but their phase is that of the cross spectrum between each response and a reference response. The operational deflection shapes are obtained by plotting the acceleration predicted by these ODS FRFs at a particular frequency.

Unlike the OMA, which creates a modal model of the railcar, the ODS represent the summation of all the modal contributions at each frequency. Consequently, dominant modes such as the lateral bending mode shown above can tend to mask the presence of less active nearby modes, such as the vertical bending mode. This can be seen in Figure 5 which shows the ODS of the railcar at 13.7Hz (factored frequency). The dominant lateral bending mode is clear, particularly in the top view, but there is little indication of the existence of the coincident vertical bending mode.

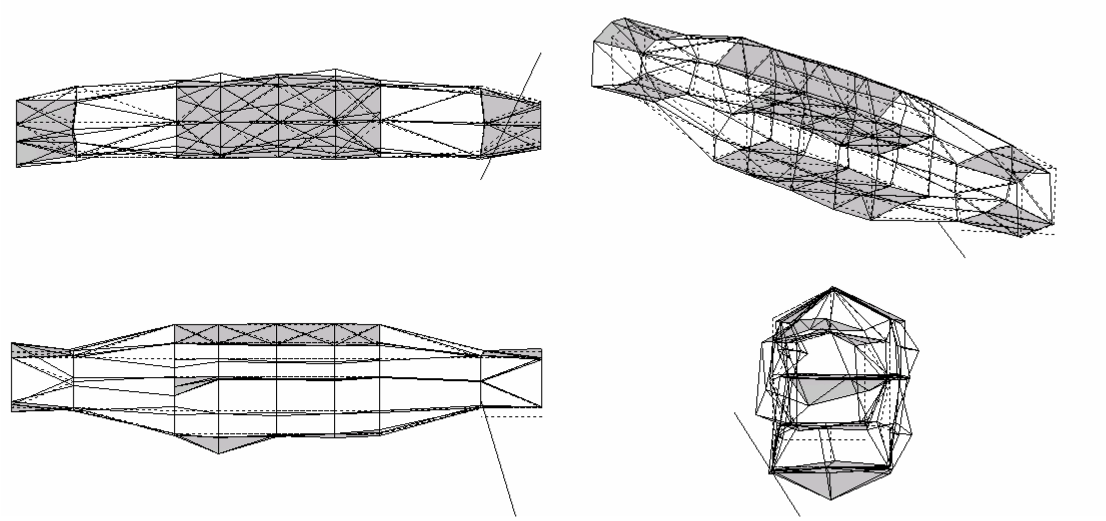

Figure 5 Operational Deflection Shape – 13.7Hz (factored frequency)

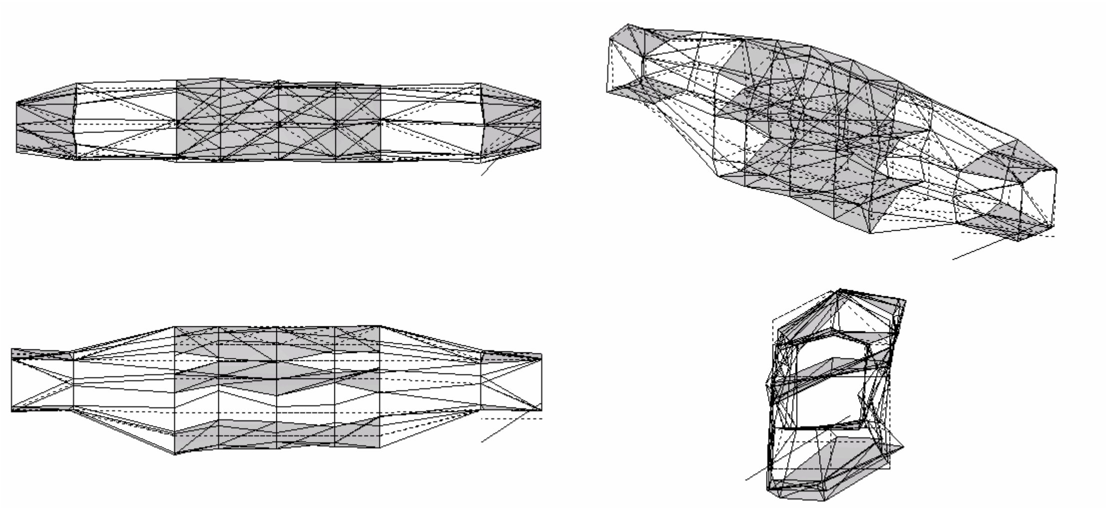

Figure 6 Operational Deflection Shape – 15.3Hz (factored frequency)

The importance of this deficiency in ODS analysis may become apparent when an assessment is made of the ride comfort based on whole body vibration weighting curves. These curves (see e.g. [8]) place a greater weighting on the vertical component of acceleration than on the lateral and longitudinal component, which in this case, equalises the contribution of the lateral and vertical bending modes to passenger comfort. Predictions of structural modifications, based say on a finite element model modified to suit the "mode shapes" predicted by the ODS, might then fail to suitably address passenger comfort criteria, highlighting the importance of OMA compared to ODS analysis.

These problems are not encountered near the whole body shear mode. The ODS at 15.3Hz, shown in Figure 6, provides a good representation of this mode.

4. DISCUSSION

This paper discussed an operational modal analysis conducted on a double-deck passenger car. The analysis employed the EFDD technique to extract the modal properties from response measurements obtained while the car was running at approximately 80km/h on track. Three modes were identified in the region around 14Hz (factored frequency); the closely spaced lateral and vertical bending modes, and the whole body shear mode. The lateral bending mode dominated the vertical bending mode which was therefore not apparent in the ODS at this frequency. The application of whole body vibration weightings made these two modes equally important for ride comfort assessments however, thus demonstrating the importance of OMA compared to ODS analysis. The OMA results were also compared to those of a workshop modal test. Significant differences in predicted resonance frequencies existed between the two tests, reinforcing the utility of OMA in that the actual *in-service* modal properties are obtained.

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