Session B5 : Isolation et amortissement vibratoire dans les joints et assemblages (Projet Brite OSCAR) – Vibroinsulation and damping in structural joints (Brite OSCAR project)

New models of joints for finite element dynamic calculations of assembled structures

De nouveaux modèles de joints pour le calcul dynamique des structures assemblées par éléments finis

Brian P. Jarvis,
Eatec Ltd,
bjarvis@eatec.co.uk

Finite element modelling of the dynamic response of fabricated assemblies is a well-established and accepted technique. Assigning stiffness, mass and damping values for the joints is difficult, however, since their properties cannot be calculated from geometry alone. This paper describes a method, developed within the OSCAR project, to model the effects of traditional and novel joints. The sensitivity of response characteristics to joint parameters has been investigated on beam-like specimens. This study indicated the type of model that could represent joint behaviour. An experimental procedure was established to characterize the joint, and determine the model parameters. Viscous dampers were used as part of the model and so the method was extended to give the closest modal damping factors, thus enabling solutions relying on model superposition techniques.

La modélisation par éléments finis de la réponse dynamique des structures assemblées est une technique bien établie et acceptée par tous. Toutefois, attribuer des valeurs de rigidité, de masse et d'amortissement à des joints est difficile, puisque leurs propriétés ne peuvent être calculées à partir de leur géométrie seule. Ce papier décrit une méthode de modélisation, développée dans le cadre du projet européen OSCAR, des effets des joints traditionnels et actuels. La sensibilité des caractéristiques de réponse à partir des paramètres d'un joint a été recherchée sur des exemples similaires à des poutres. Cette étude présente un type de modèle qui peut définir le comportement du joint. Une procédure expérimentale a été établie pour caractériser le joint et déterminer les paramètres du modèle. Les amortisseurs visqueux ont été utilisés comme une partie du modèle et ainsi la méthode a été étendue pour donner les facteurs modaux d'amortissement les plus proches autant que des solutions possibles reliées aux techniques de superposition des modèles.

Introduction

All thin plate assemblies have to be joined in some way, either by traditional methods of bolting, welding and bonding or by novel fastenings such as smart joints. It is known that when subjected to dynamic loading, energy is lost in the joints, and if this property can be optimised, noise transmitted from structures can be reduced. Understanding the relationship between joint design and the noise reduction was the objective of the OSCAR project.

A separate work package within OSCAR was devoted to the modelling of joints for use with finite element analysis. If this could be achieved, then it would be possible to predict vibration levels at the design stage and select the most beneficial joining method.

Approach to joint modelling

The requirements of the joint models were limited by the scope of the OSCAR project. Principally :

- joints between thin plates (<3 mm)
- frequency range 0-500 Hz
- joint models to be simple (masses, springs, dampers)
- joint models to be linear
- joint models to be compatible with commercial FE codes

The procedure for assigning values to the joint models relied on experimental testing, first without, then with the joint, and ascribing the difference to the joint behaviour. Stiffness, mass and damping parameters would be adjusted until the response from the FE model and the test agreed.

It was assumed that there would be different parameters for each specified joint in which material properties, overlap, fastener type and spacing could all vary.

Traditional joints in beams

Before considering the more complex problem of plates, joints between two beam-like thin strips were investigated experimentally. Each beam was 1 000 mm x 50 mm and generally lmm thick. The joint was formed with a single M4 bolt and an overlap that could take the values 10, 20 or 30 mm. The beams were suspended vertically on soft supports and the upper end excited by a shaker (Figure 1). Responses were measured at 10 positions along the length of the beam. Figure 2 shows a typical FRF from the three overlap values and clearly shows that the overlap magnitude did not greatly affect the frequency of response. Bolt torque was varied and this too had no effect. This part of the study concluded that joint stiffness was not significantly influenced by the parameters of the joint.

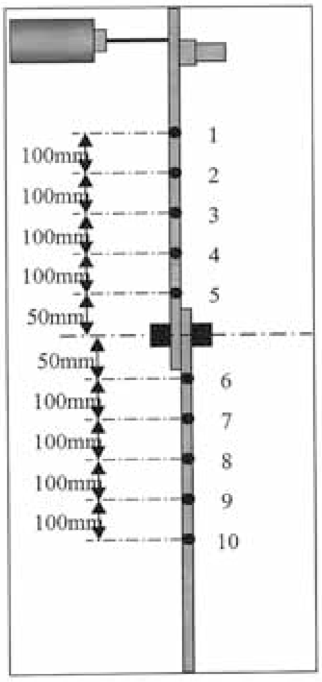

Fig. 1 : Experimental setup for tests on beams

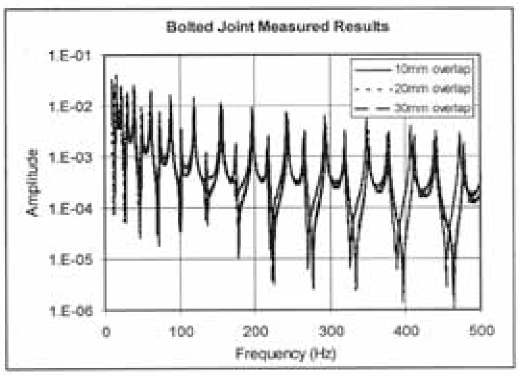

Fig. 2 : FRSs from overlaps between 10 and 30 mm

Smart joints in beams

Smart joints use a layer of viscoelastic material within the joint, and their construction would suggest a greater influence on stiffness and damping. A variety of smart joint types were tested and two examples are shown in Figure 3. Testing showed that, compared with un-jointed beams there was no change in natural frequencies implying that even with joints designed to be flexible, there was no relative motion across the joint. There was, however, some influence on damping, which can be seen in Figure 3. The un-jointed beam approximately followed a Rayleigh damping, but with the jointed beams the level of damping oscillated with increasing mode number.

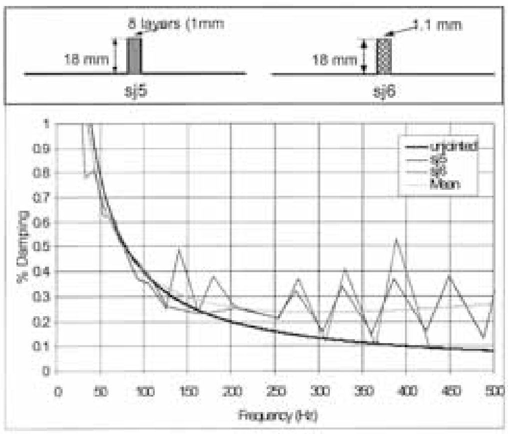

Fig. 3 : Damping caused by smart joints

Bonded joints in glass beams

Bonding of glass components was of particular interest in the OSCAR project. The fragility of glass required shorter, thicker strips to be used, dimensions 500 x 50 x 3 mm. Pairs of strips were joined with a soft adhesive with overlaps of 10, 20 and 30 mm. The natural frequencies compared with an un-jointed geometry were only affected for the 10 and 20 mm overlaps showing that for these values the flexibility of the joint itself was significant. The damping, expressed as a percentage of critical, was highly mode dependent (Table 1). The mode shapes could be divided into those in which there was a large rotational difference across the joint and those where it was small (Figure 4). The modes with the higher damping factors corresponded to those where the rotational difference was largest. This suggested a possible mechanism for modelling the damping in a joint without having to allow relative movement across the interface.

Mode	% Critical damping
1	0.3
2	2.4
3	0.3
4	1.6
5	1.4
6	0.5
7	1.1

Table 1 : Mode dependent critical damping

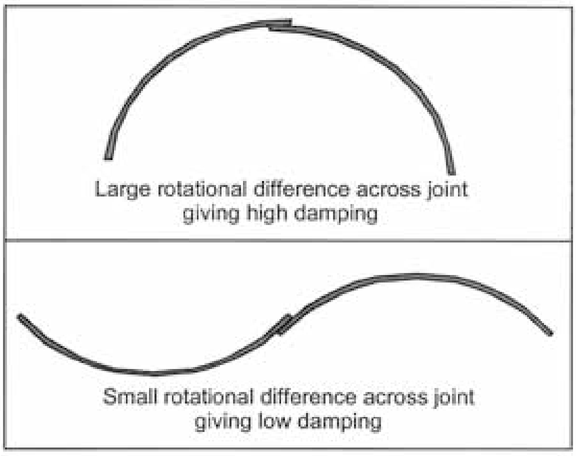

Fig. 4 : Proposed damping model

Proposed joint model

The knowledge gained in the tests on strips suggested a type of joint model that could also be used with plate assemblies. The joint model should represent the mass effects of both the fasteners and the thickness increase due to the overlap. Along the joint, the overlap would add some stiffness, although this is only important for joints between plates. Mass and stiffness effects can be deduced by geometry alone and are adequately represented by adding a beam fmite element along the line of node forming the centreline of the joint.

The damping model proposed was a rotationa viscous damper linking nodes either side of the joint centreline, separated by a distance Δx (Figure 5).

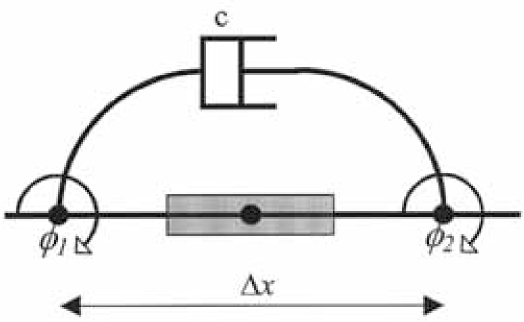

Fig. 5 : Proposed damping model

The damping constant for each damper would come from the characteristic value "**c**" of damping density expressed in Nm/(rad/sec)/m. In this way, only one value would be needed to give the damping effects of a particular joint.

The "**c**" value must be found from tests, and -since only one parameter is needed, a procedure was necessary to reduce the quantity of measured results. The technique devised was based on energy principles and involved creating a spectral function termed the Cumulative Response Function. (CRF).

A reference jointed plate assembly is suspended in soft supports to give the free-free boundary condition. Random excitation in the frequency range of interest is applied, and the mobility V_i (*f*) determined at *N* points spread across the plates. The spatial average square mobility is calculated as :

$$\overline{V^2}(f)=\frac{1}{N}\sum_{i=1}^{N}V_i^2(f) \qquad 1$$

The CRF, $\overline{V_c^2}(f_j)$ can then be defined as :

$$\overline{V_c^2}(f_j)=\frac{1}{F_R}\sum_{k=1}^{j}\overline{V^2}(f_k)\delta f_k \qquad 2$$

Where F_R = Frequency range of test
f_k = Discrete frequency value
δf_k = $\frac{1}{2}(f_{k+1}-f_k)$

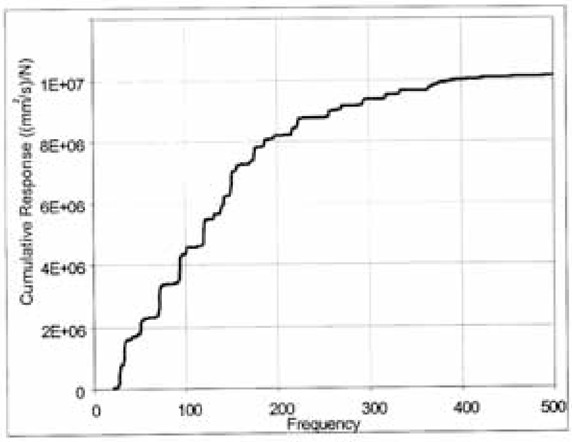

Fig. 6 : Typical CRF

The step increases in the CRF correspond to response frequencies, and indicate the contribution of each mode to the overall response.

A model of the reference assembly is then constructed and the CRF predicted with a best guess "**c**" value. The "**c**" value can then be tuned until the prediction gives a satisfactory match to the experimental results.

Figure 7 shows the CRF's for the test and model of a pair of plates bolted together with a 50 mm spacing between bolts and 20 mm overlap. Comparison is made with the CRF from an unjointed plate of the same overall size.

While a single characteristic value is all that is necessary to give the damping properties of the joint, that value does

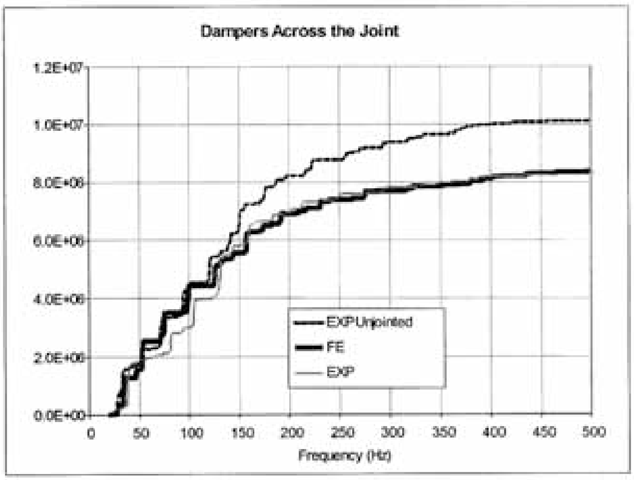

Fig. 7 : CRF values with and without joint

depend on the separation of the damper nodes across the joint. The relationship between these two quantities has been found to take the form :

$$c = Ae^{b\Delta x^{p}} \quad 3$$

If the joint model is to be used for different mesh densities, then the parameters in this relationship must be first determined from FE models.

Converting to modal damping factors

The characteristic value of damping to be applied to viscous dampers is useful because only one value is needed to describe the energy loss in any particular joint. However, the use of viscous dampers prevents the computationally efficient modal superposition methods from being employed. The equivalent modal damping factors can be calculated from the un-damped natural frequencies and mode shapes. Damping values will now become structure dependent, but modal superposition methods are now possible.

The expression for the modal damping factor ξ_j for mode j is given by :

$$\xi_j = \frac{1}{N} \times \frac{cL}{2\overline{m}_j \omega_j} \sum_{p=1}^{N} \varphi_{pj}^2 \quad 4$$

N = number of dampers along a joint line
L = length of joint line
c = viscous damper value as defined above
ω_j = frequency of mode j (rad/sec)
$\overline{m}_j$ = modal mass in mode j (output by most FE codes
φ_{pj} = rotational difference input to damper p in mode j (radians)

Acknowledgements

The author would like to acknowledge the support of the EC for the OSCAR project, and the experimental work carried out by partners KTH University, Stockholm and CETIM, Paris that contributed to this modelling of joints study.