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STRUCTURAL COUPLING AT JOINTS IN MOTOR VEHICLES

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1. INTRODUCTION

Statistical Energy Analysis is attracting increasing interest as a technique for predicting mid to high frequency structural vibration levels in motor vehicles. Modern motor vehicle bodies are constructed largely of thin sheet metal pressings spot welded together. One of the challenges in applying SEA is the calculation of CLF's across the complex joints produced by this construction method. A method proposed by Langley & Heron[1] using "wave dynamic stiffness matrices" is used in conjunction with finite strip elements[2] to construct three joint models representing parts of a small passenger vehicle shown in Fig. 1. Measured and predicted results for coupling at joints and mobility at points in a vehicle are shown.

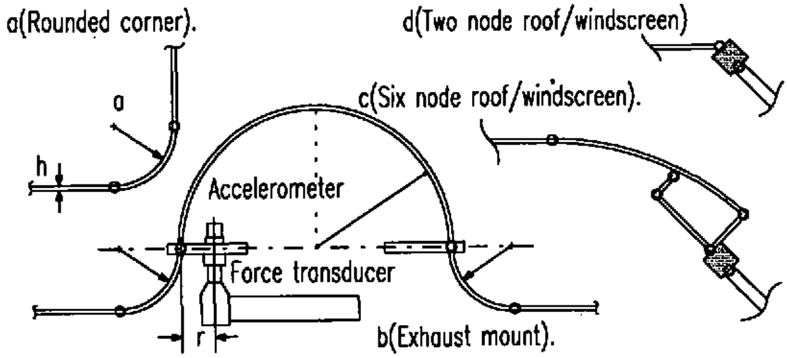

Fig. 1, Two four and six node joint models.

2. DYNAMIC STIFFNESS MATRICES

Following the work of Langley and Heron [1] & Langley [2,3] a joint modller has been constructed which allows the assembly of an arbitrary number of flat or curved semi-infinite panels along a straight line. These panels can be orientated at arbitrary angles to one another. In addition to this the modeller incorporates finite width elements for curved and flat panels of a type described by Langley [2]. An elastic interlayer element identical to that used by Mees & Vermeir [4] has also been used. All the element types are shown in Fig. 1, assembled along their nodal lines. Four degrees of freedom exist at each node allowing both in-plane and out of plane motion. The four displacements and there corresponding forces are assumed to vary as travelling waves having the same wave number along the boundary. The boundary force and displacement vectors, $\mathbf{f}$ and $\mathbf{d}$ respectively are related at an elements boundary by Eq. 1.

$$\mathbf{A} . \mathbf{d} = \mathbf{f} \tag{1}$$

Where $\mathbf{A}$ is the "wave dynamic stiffness matrix" [1] of the element. This matrix has been calculated for the panel elements using a wave approach. On assembling the stiffness matrix for the complete joint model, $\mathbf{B}$ from those of its constituent elements it is possible to calculate the response of the joint to incident waves. All panel elements are based on thin plate theory which will be applicable to nearly every body panel of a modern motorcar.

3. TRANSMISSION AT ROUNDED CORNER JOINTS

The corner joint shown in Fig. 1. part a with h/a=.01 has been modelled by connecting two semi-infinite flat panels to a two node curved panel element . In Fig. 2, results for bending wave transmission coefficients at the rounded corner joint are shown, diffuse wave field and a wave incident at 20° are compared with results for a sharp "L" joint [5]. The *x* axis is labelled f/f1 where f is frequency and f1=395Hz is the plane stress ring frequency [3] for the curved section. At low frequencies the rounded corner diffuse field value is the same as a sharp "L" corner as expected, it then drops rapidly to a minimum of 0.08 as f/f1 approaches 0.0017, rising gradually to 0.9 when f/f1= 3.4. For waves incident at 20° the two results are similar at low frequencies. The rounded corner result drops rapidly below 0.1 between f/f1 equal to 0.0001 and 0.4 above which it rises to 0.99. Investigation has shown that the sharp rise occurs at higher frequencies as the angle of incidence increases. This explains the more gradual transition to total transmission shown by the diffuse field value. Large errors could result from this switch like behaviour when using the SEA diffuse field assumption to calculate coupling across a rounded joint of this type, if the field in the transmitting system were not diffuse. Transmission of energy in the form of in-plane waves is greatly reduced for the rounded joint modelled.

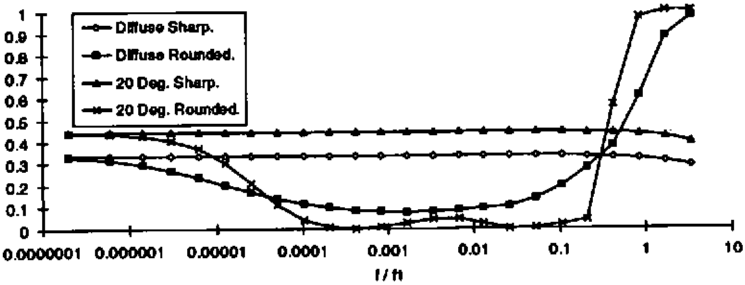

Fig. 2, Bending Wave Transmission Coefficient.

4. ROOF TO WINDSCREEN COUPLING

The CLF between the roof and the windscreen of a small passenger car has been predicted using the joint models shown in Fig. 1(c,d). Damping measurements of the windscreen were made using the reverberant.decay method giving typical values for total loss factor of around 108 dB over the frequency range considered. Acceleration level differences between the roof and the windscreen were measured by hitting the source with a plastic hammer and measuring the acceleration levels of both subsystems. The coupling loss factor from the roof to the windscreen was calculated using the energy flow relations of a simple two subsystem SEA model.

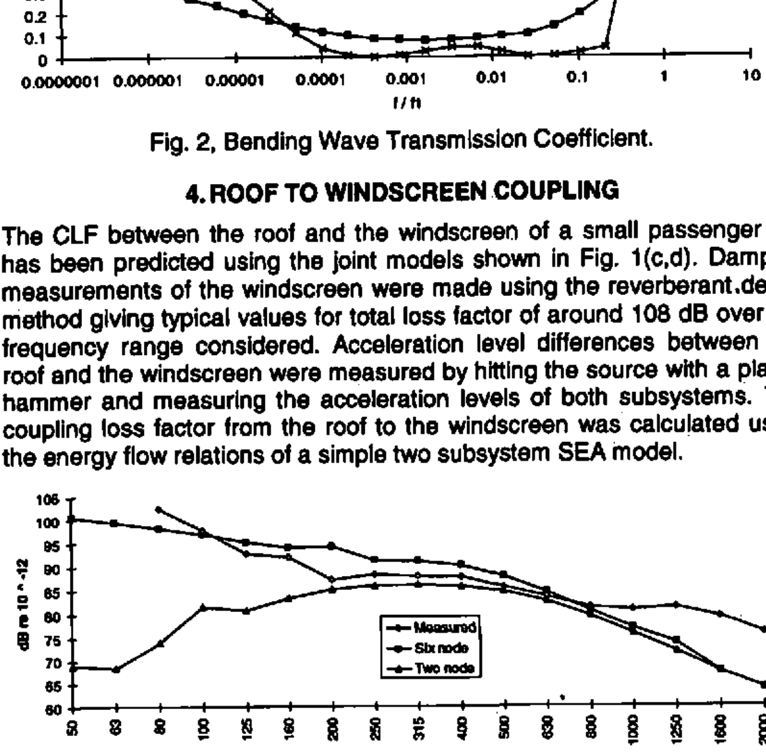

Fig. 3, Measured & Predicted CLF from Roof to Windscreen.

Fig. 3, shows predicted CLF's. for six node and two node models alongside measured results. The six node model shows better agreement with measured results bellow 200 Hz, above 800 Hz both predictions are less than measured.

5. POINT MOBILITY

In a motor vehicle point loading may occur at door hinges and at engine gearbox and suspension mounts. The joint modeller assembles the wave stiffness matrix of the entire joint structure from those of its constituent parts. This matrix can be inverted to produce the wave mobility matrix

$Y = i\omega B^{-1}$. The point mobility of the structure may then be calculated in a similar manner to the method used by Eichler[6] except that the fourier integral over wave number space must be evaluated numerically. This method has been tested against Eichler's analytical solutions for plate edge mobility, generally showing good agreement. Excellent agreement is obtained with Cremer's[3] infinite plate mobility. Fig. 1(b), shows the four node model used to calculate the exhaust mount mobility. Mobility was also measured using an accelerometer and an impedance hammer. Fig. 4, shows the real parts of the measured mobility compared with two predicted results. The flat plate prediction is given by $Y_C \times r^2$ where Y_C is Cremer's[3] infinite plate moment mobility. Both predicted mobility's are lower than measured.

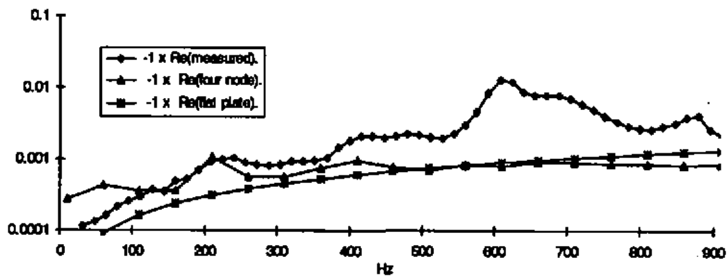

Fig. 4, Exhaust mount point mobility (m/Ns).

6. CONCLUSIONS

The theoretical joint modeller has been applied to studies of sound transmission and response in a real motor vehicle. Measured and predicted results for sound transmission show good agreement at frequencies bellow 1000Hz.

7. REFERENCES

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