# ESTIMATION OF VIBRATION LEVELS AT LAUNCH ON EQUIPMENT MOUNTED IN SATELLITES

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## Introduction

Satellite structures comprise essentially of a central tube with attached equipment platforms and appendices. The whole structure being surrounded by a shroud for protection. During launch of the spacecraft, vibration energy is transmitted into the structure. Energy sources are broad band random in nature, they include:

(i) Pressure fluctuations around the shroud (turbulent boundary layer flow, rocket efflux).

(ii) Direct mechanical inputs across the launcher interface (rocket motor forces).

Higher frequency vibrations on the equipment platforms may cause dangerously high accelerations in the mounted components. Equipment manufacturers must therefore qualification test during development. A knowledge of realistic vibration levels is important as overtesting equipment may result in heavier or more expensive constructions (damping treatments, isolators). Under testing may result in subsequent component failures during flight.

## General Response Estimates

A computer program using Statistical Energy Analysis (1) is being developed to provide estimates of the high frequency vibration levels on the platform without attached equipment. The next stage of the investigation is to determine the effect of the presence of the equipment on the overall platform vibration energy levels and to estimate the vibration levels at the mounting points using impedance data and theory for coupled systems.

## Earlier Work

(a) A general investigation of vibration responses on a model satellite, the EX1, had shown the effect of adding simulated equipment boxes to the platforms (2). On the addition of a box, the platform rms vibration levels in the vicinity of mounting points (directly underneath the box) fell by amounts up to 50% and proportional to the box mass. At points on the platform midway between mounted boxes, levels fell only by about 5%.

The localised effect around the mounting points resulted from the box/platform coupling impedance mismatch. An overall reduction in platform vibration level could be due to damping effect (absorption of energy) of the boxes, and/or a change in platform decrease impedance levels resulting in a change of vibration power flow input.

In another related study, equipment platforms were represented with aluminium plates and equipment with point masses. From this work it was deduced that if the equipment box had N attachment points 1/Nth of the total mass M should be associated with each mounting point. The total mounting point impedance could be obtained by summing the measured resonant plate impedance to the non-resonant (imaginary) pure mass impedance $i\omega M/N$.

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## The Current Investigation - Experimental Arrangement

To investigate further the above results a simplified EX1 configuration was set up. It consisted of a uniform honeycomb platform attached to the centre of a plain cylinder. Vibration power was put in by an electromagnetic vibrator at a point on the cylinder and measured with an impedance head. This gave a fairly uniform power input to the platform via the coupling ring. Four accelerometers were attached to the platform at midway between box positions to give a spatial average platform vibration level. Changes in this level could then be monitored for the addition of up to 7 boxes.

The simulated equipment units had an even mass distribution, thus mounting point impedances and vibration levels for a given box were all about the same. Impedance and vibration levels were measured at two mounting points and a single average value used.

## Theory

### Platform/Equipment box acceleration ratio

For a given spatial average platform vibration level, the mounting point accelerations of the box are given by:

$$\overline{a_b^{\,2}}(t) \;=\; \langle \overline{a_p^{\,2}}(t) \rangle \left| \frac{Z_p}{Z_p + Z_p} \right|^2 \tag{1}$$

where $Z_p$ is the platform impedance at the equipment mounting point. $Z_p + Z_b$ is the total mounting point impedance with the box added. $Z_b = Z_m$ for a box behaving as a pure mass, and $Z_b = Z_n$ for a resonant box.

### Power Flow

For equipment items where significant resonant coupling with the platform occurs, the average power flow into the mounted component, $\bar{P}$ can be written as:

$$\bar{P} \;=\; N \left\langle \frac{\bar{a}'^{\,2}(t)}{\omega^2} \right\rangle \left| \frac{Z_p}{Z_p + Z_n} \right|^2 \quad \mathrm{Re}\,(Z_n) \tag{2}$$

where $\langle \bar{a}'^{\,2}(t) \rangle$ is the platform acceleration after the box is added (from Ref.1). The resulting fall in platform vibration energy can be deduced using an energy balance expression for the structure, and measured power input data.

## Results

Figure 1 shows the platform acceleration levels with and without the seven equipment boxes. Results are frequency averaged values for 100 Hz constant bandwidths (normalised to power input). There is an overall reduction of 2 dB in platform vibration energy on adding the boxes (20% drop in rms vibration level). This is a larger change than was seen in previous work, but still indicates low power absorption i.e. Re $(Z_n)$ in equation (2) is small, (Re $(Z_n)$ = 0 for pure mass $i\omega M$).

Figure (2a) shows $|Z_p|$ for the honeycomb platform. Fig.(2b) shows $|Z_p + Z_n|$ for the combined box + platform (mass lines at $|Z_p + Z_m|$ and $|Z_p + Z_m/N|$ . Figure (2c) shows $|Z_n|$ for the box alone, suspended with other points free (for this box $M = 4.5$ Kg, $N = 6$, mass lines at $|Z_m|$ and $|Z_m/N|$. Equation (1) was then formed with this point impedance data. Measured and predicted responses at the box feet are shown in fig. 3. The measured resonant $Z_n$ and the measured $(Z_p + Z_n)$ for the combined system were used. In the overall frequency average the predictions are low by about 3 dB.

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## Conclusions

(i) Adding components may not significantly change the platform vibration levels (frequency range 50 Hz - 2 kHz in these experiments).

(ii) First attempts to predict vibration levels for this case using point impedances gave reasonable agreement. The pure mass theory gave good agreement at low frequencies. Measured resonant box data was better at high frequencies but not as good at low frequencies. A combination of the two sets of data may give the best results.

## References


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3. B.L. Clarkson, R.J. Pope. Progress Report for ESA Contract No. 4118/79.


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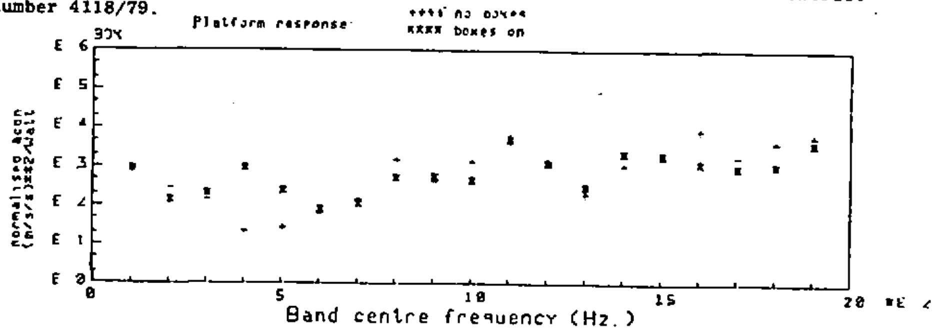


Figure 1: Average mean square acceleration of platform with and without boxes

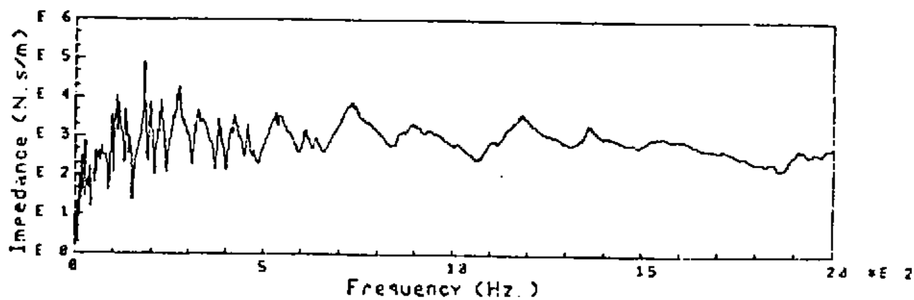


Figure 2: Modulus of Mounting Point Impedance of Honeycomb Platform

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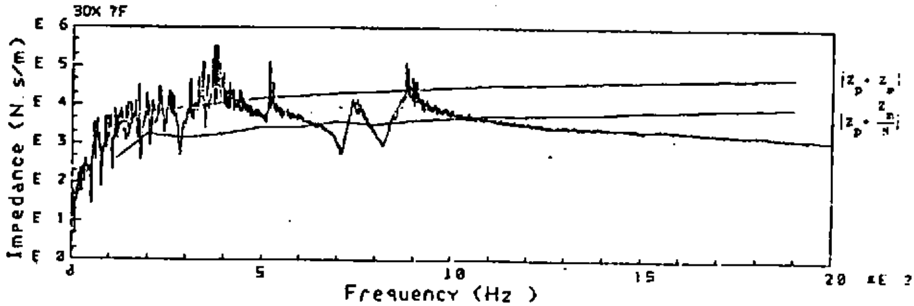

Figure 2(b): Modulus of Mounting Point Impedance of Platform + Box.

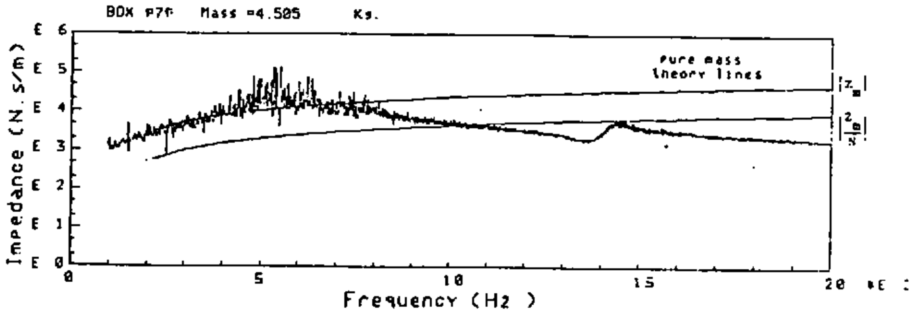

Figure 2(c): Modulus of Mounting Point Impedance of Box (other points free).

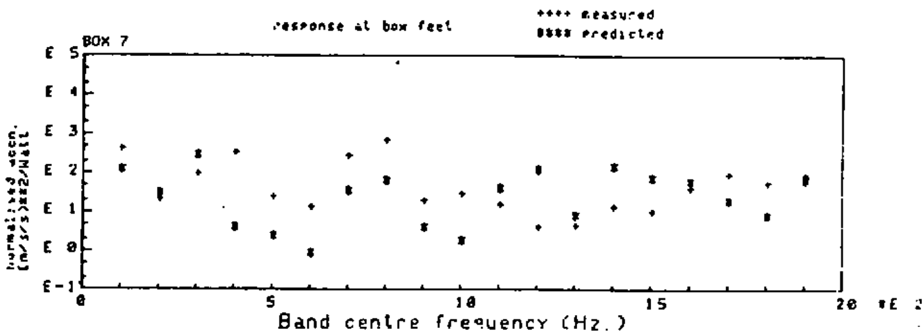

Figure 3: Measured and predicted average mean square accelerations at box feet.

# VIBRATIONAL ENERGY DISTRIBUTION AND PROPAGATION MEASUREMENTS IN HONEYCOMB AND CORRUGATED SPACECRAFT COMPONENTS

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## 1. THEORY

The Statistical Energy Analysis method of predicting vibration levels in spacecraft structures is formulated in terms of frequency and spatial averages of vibrational energy levels within the structural components. The theory applied in ISVR |1| was developed from a consideration of the energy balance in the structure. For example, the loss factor is indirectly estimated from experimental results using the expression:

$$\eta(f) \quad = \quad \frac{\overline{F^2\ (t)}}{\langle\overline{v^2\,(t)}\rangle} \quad \frac{n(f)}{8\pi f M^2 A^2} \qquad (1)$$

where $\eta(f)$ is the loss factor, F(t) the point force excitation, $\langle\overline{v^2(t)}\rangle$ is the spatial average of the mean square velocity, n(f) is the modal density of the structure, M.A is the total mass, and f is the centre frequency of the band of interest.

When structural components 1 and 2 are connected together the energy transfer from components 1 to 2 is described by the coupling loss factor $\eta_{12}$:

$$\eta_{12} \;=\; \eta_2 \quad \frac{n_2\ E_2}{(n_2 E_1\ -\ n_1 E_2)} \qquad (2)$$

where $\eta_2$ is the dissipation loss factor of the receiving component, $n_1$ and $n_2$ are the modal densities of components 1 and 2. $E_1$ and $E_2$ are the vibrational energies in components 1 and 2:

$$E_1 \;=\; M_1 A_1 \quad \langle\ \overline{v_1^{\,2}(t)}\ \rangle \qquad (3)$$

$$E_2 \;=\; M_2 A_2 \quad \langle\ \overline{v_2^{\,2}(t)}\ \rangle \qquad (4)$$

The estimations of loss factors (1) and coupling loss factors (2) are based on measurements of the spatial average mean square velocities.

In typical structures there may be significant spatial variation in levels. To investigate this effect a series of measurements were carried out on real spacecraft components.

## 2. NEAR FIELD EFFECTS

When a structure is excited by a broad band point force the average acceleration levels at the excitation point tend to be somewhat higher than the acceleration levels at other parts of the structure. It is thought that this 'near field' effect caused by non propagating waves will not affect the measured input power because this is derived from the real part of the mobility. The magnitude of the non propagating waves will be related to the imaginary part of the mobility. In the case of honeycomb structures the situation is complicated by the small scale local discontinuities. Figure 1 shows the position of the excitation and measurement points.

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## 3. TYPES OF MEASUREMENT MADE

As the objective of the tests was to determine the change in average vibration level with distance from the driving point and also the effect of the different mounting sub structures, the acceleration data is not converted into velocity. Thus the comparison is made on the basis of the inertance i.e.

$$\mathrm{mod}\left\{\frac{A(i\omega)}{F(i\omega)}\right\}$$

Figure 2 shows the modulus of the driving point inertance at point DN1. Similar results were obtained for the other two driving points. A typical transfer inertance is shown in Figure 3.

The variation of the acceleration levels as normalised to the input force can be investigated by normalising the transfer inertance curve. i.e. by calculating the ratio $R(\omega)$

$$R(\omega) = \frac{\mathrm{mod}\left\{\frac{A_j(i\omega)}{F(i\omega)}\right\}}{\mathrm{mod}\left\{\frac{A(i\omega)}{F(i\omega)}\right\}} = \mathrm{mod}\left\{\frac{A_j(i\omega)}{A(i\omega)}\right\}$$

However as not all the measurements of $A_j(i\omega)$ can be made simultaneously several runs are required.

## 4. SPATIAL VARIATION IN AVERAGE VIBRATION LEVELS

In order to give a clear picture of the acceleration levels throughout the structure the $R(\omega)$ curves for all the points were averaged into 200 Hz bands, and then the values within each band were plotted against the successive positions of the measuring points. The effect of the different driving point sub structure is clearly shown in the curves. In the frequency range up to about 600 Hz the average vibration levels are approximately the same over the whole structure. Above 800 Hz the levels begin to fall off in the furthest half of the panel, (Fig. 4). Above 1400 Hz driving directly onto the face plate gives much lower acceleration levels throughout the structure (Fig.5).

## 5. EFFECT OF SPATIAL VARIATIONS ON LOSS FACTOR ESTIMATIONS

The honeycomb platform and the corrugated cylinder and cone were coupled and the effects of incorrect estimation of energies due to the limited number of accelerometers used were examined. In the case of typical spacecraft components where there are significant spatial variations in levels the careful selection of the measurement positions is crucial. Analysis shows that inadequate distribution of accelerometers can produce even negative coupling loss factors in certain bands.

## REFERENCES

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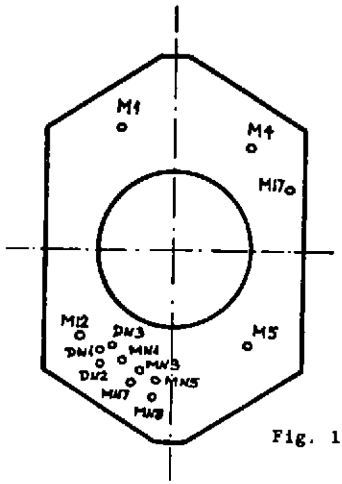


| Position | Distance from DN1 (mm) |
|---|---|
| MN1 | 30 |
| MN3 | 115 |
| MN7 | 125 |
| MN12 | 140 |
| MN5 | 235 |
| MN8 | 230 |
| M5 | 830 |
| M1 | 1290 |
| M4 | 1380 |
| M17 | 1265 |

Fig. 1 Position of Measurement Points on Honeycomb Platform

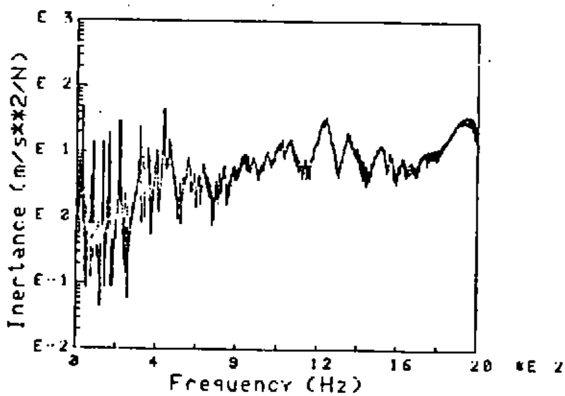


Fig. 2 Modulus of driving point inertance at position DN1

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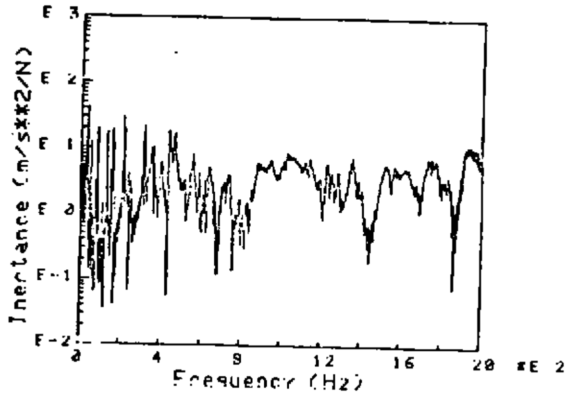
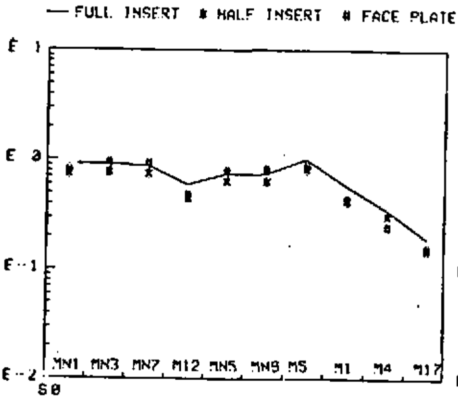



Fig. 3 Modulus of transfer inertance between driving point DN1 and the remote point M1



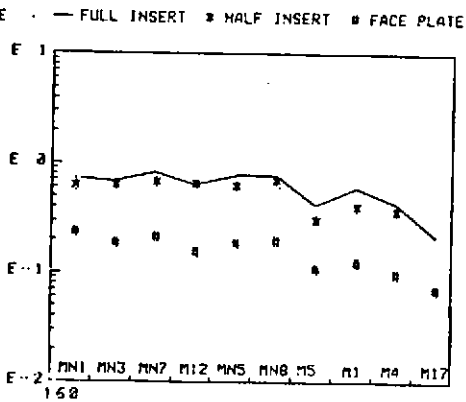




Fig. 4 Normalised Transfer Inertances across the panel. Average 800-1000 Hz.

Fig. 5 Normalised Transfer Inertances across the panel. Average 1600-1800 Hz.