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THE PREDICTION OF STRUCTURE-BORNE SOUND POWER THROUGH MULTIPLE CONTACT POINTS BETWEEN MACHINES AND FLOORS.

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1. INTRODUCTION

The prediction of the structure-borne sound power is simple only when source and receiver are connected through a single point and excitation is uni-directional. In the more realistic situation where multiple contact points and multiple excitation components are involved coupling between the points and between the components occur and prediction becomes complicated. To obtain the solution a mobility matrix approach is usually adopted. Though such can offer a solution the approach is often intractable and importantly it gives little engineering insight. An alternative approach has been suggested [2] in which an effective point mobility concept is introduced. If this can be successfully implemented it could reveal much to an engineer and if adopted would be an important engineering tool. In this paper implementation of the effective point mobility concept is considered via a case study where the total active power for an industrial fan mounted upon a floor is estimated.

2. THEORY

In the situation where the source is connected to the receiver via only one point and motion is constrained to one direction the structure-borne power at the point is simply given by [1];

$$Q=\frac{(V_{sf})^2}{|Y_s+Y_r|^2}Y_r \tag{1}$$

where V_{sf} is the mean square of the point free velocity and Y_s, Y_r are the source and receiver point mobilities respectively. Practically, the free velocity is measured with the machine suspended by very resilient supports and running under normal operating conditions.

In the more complicated situation where the source is connected

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via a number (N) of points and motion is general (ie. exists in all six degrees of freedom) the response of a point is influenced by the coupling between it and all other points and between the different components of motion. The extent of the influence is dependant upon the structure's response to the coupling and also upon the degree to which the coupling is excited. Thus the response of a point can be given by [2] the effective point mobility;

$$Y_{ii}^{nn\Sigma}=Y_{ii}^{nn}+\sum_{m=1 m\neq n}^{N} Y_{ii}^{nm}\frac{F_i^m}{F_j^n}+\sum_{j=1 j\neq i}^{6} Y_{ij}^{nn}\frac{F_j^n}{F_i^n}+\sum_{m=1 m\neq n}^{N}\sum_{j=1 j\neq i}^{6} Y_{ij}^{nm}\frac{F_j^m}{F_i^n} \qquad (2)$$

where Y denotes general mobility, F general force and the indices k, n specify location and i, j specify direction. The first term is the direct response, the sum that is the second term is the transfer coupling, the third sum set is the cross coupling and the fourth sum set the cross-transfer coupling.

Using the effective mobility concept for the multi-point-connected, multi-directional situation the power at a point is, analogous to Eq.(1), given by;

$$Q_j^n=\frac{(V_{sf_j}^{\,n})}{|Y_{s_j}^{\,n n\Sigma}+Y_{r_j}^{\,n n\Sigma}|^2}Y_{r_j}^{\,n n\Sigma} \qquad (3)$$

where $Y_{rj}^{nn\Sigma}$ is the effective mobility of the receiver point and $Y_{sj}^{nn\Sigma}$ the effective mobility of the source point.

The total active power into the receiver is subsequently obtained by summing the power contributions from each point and component and [1] taking the real part;

$$Q^{tot}=Re(\sum_{n=1}^{N} Q^n) \qquad (4)$$

Eq.(3) and Eq.(4) indicates therefore that the power can be calculated for the multiple point, multiple component situation providing the effective mobilities are known.

To obtain the effective mobility knowledge is needed, see Eq.(2), of the mobility functions and of the force distribution amongst the connecting points. The mobilities can be measured with the source and receiver separated. The force distribution however is dependant upon the connection of the source to the receiver. Hence to predict the total power via the effective mobility concept with coupling between the points taken into account an estimate of the force distribution must be made.

2.THE SYSTEM

The viability of using various assumed force distributions in the estimation of the total active power is considered via a case study; that of an industrial fan rigidly mounted upon a 20cm concrete floor. Data for the fan was obtained by measurement whilst for the floor a simply-supported condition was assumed along all four edges and the well known plate equation invoked [1]. A loss factor of 10^{-2} was used. Vertical translational excitation is assumed to be the dominant vibrational component and is only considered.

A schematic diagram of the case is shown in figure 1.

In figure 2 typical point and transfer mobility magnitudes of both fan and floor are shown. Three characteristic regions are observed with respect to the relationship between the point and transfer mobilities of the fan. A low frequency region up to 100Hz in which both are 'equal', a mid frequency region between 100Hz and 1000Hz in which the transfer is less than the point and a high frequency region above 1000Hz in which both point and transfer are resonant and of the same order of magnitude until divergence assumes importance. The low frequency region corresponds to where the structural response is dictated by the mass of the body, the mid frequency region where the stiffness of the mounting point is dominant and the high frequency region is where wave behaviour occurs in the body.

For the floor up to about 200Hz the point and transfer mobilities are approximately equal. At higher frequencies both transfer and point are resonant where again there is a relative decrease of transfer mobility due to divergence and internal losses.

In general the mobilities of the fan are typically 10^{+1} higher than those of the floor. Hence the fan mobilities will dominate in the denominator of equation (3) and the source can be considered a constant force source.

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4.THE EXACT SOLUTION

In order to assess the suitability of assumed force distributions an 'exact' solution of the total active power in the system was calculated by solving the matrix equation;

$$[F]=[Y_{sr}]^{-1}.[V_{sf}] \qquad (5)$$

to obtain the exact forces and substituting these into equations (2) through to (4). The 'exact' total active power is shown in figure 3 and is seen to decrease at approximately 12dB per octave from a peak at 100Hz. An estimate of overall power into the floor must thus have a high accuracy at the lower frequencies.

5.THE UNCOUPLED SOLUTION

The simplest assumption is that there is zero coupling between the contact points. The effective point mobilities then reduce to their point mobility terms and no estimate is needed for the force distribution. The 'uncoupled' estimate of the total active power is shown in figure 4 normalised with respect to the exact solution. In the stiffness controlled region between 100Hz and 1kHz the estimate is typically within +/-5dB of the exact solution whilst for both the mass and resonant regions discrepancies in excess of this are more common. In terms of engineering accuracy it is suggested therefore for a narrow band analysis the uncoupled solution is acceptable in the stiffness region whilst for both the mass and resonant regions it is not.

A reason for the estimate being more accurate in the stiffness region is because the coupling between contact points on the fan is, as indicted by a low transfer mobility cf. point mobility, smaller here than in the other regions. Hence the contact points are only weakly coupled and a point mobility only estimate is appropriate for the effective mobilities of the fan.

To improve the estimate in both the mass and resonant regions, coupling between the points must be taken into account.

6.FORCE RATIO ESTIMATES

To account for coupling an estimate of the force distribution needs to be made.

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The magnitude of the exact force ratio between two contact points, f2/f1 for the case considered, is shown in figure 5. The other force ratios were similar. The variation is in excess of +/-20dB but with an observed mean of about unity. Hence by ignoring the details it is suggested that a simple unity estimate for the force ratio magnitude can be made. The phase of the force ratios is known to vary 'randomly' between $+/-2\pi$ with a mean of zero. Hence an estimate of zero is suggested for the force ratio phase. The resulting 'unit magnitude, zero phase' total power estimate, again normalised with respect to the exact value, is shown in figure 6.

In all three regions, contrary to what might be expected, discrepancies between the estimate and the exact have increased. In the stiffness region the contact points of the fan are known to be only weakly coupled. Hence the influence upon the fan effective mobilities of using a unit magnitude, zero phase force ratio assumption as opposed to an uncoupled assumption will be small in the region. The increased discrepancies in the stiffness region have therefore been introduced through the effective mobilities of the floor. A reason is that the unit magnitude, zero phase force ratic estimate assumes equal and in phase coupling between all the contact points. In fact the coupling may be somewhat more random ie. between points 1 and 2 it may be strong with a phase of π whilst between points 1 and 3 it may be weak with a phase of $-\pi$. Hence the uncoupled estimate in which the overall coupling is assumed to be zero is a more accurate approximation.

If an accurate estimate of the total active power is to be achieved accurate estimates of the complex force ratios are therefore needed.

The simplest estimate of the force ratios is obtained by assuming that the force at each contact point is dominated by the point mobility and the free velocity at that point. The force at point n is then given by;

$$F^n = \frac{V^n_{sf}}{Ys^n_{11}} \qquad (6)$$

The resulting 'point mobility, free velocity' normalised total active power estimate is shown in figure 7.

Compared to the 'unit magnitude, zero phase' estimate improvement is seen. When compared with the 'uncoupled' estimate however any improvement is not so clear and discrepancies in excess of 10dB are still seen. This suggests therefore that estimates of the force distribution need to be very accurate if accurate predictions of the total active power are to be achieved.

7. CONCLUDING REMARKS

A case study on the implementation of the effective point mobility concept has been investigated. It has been seen that the region of response as indicated by the form of the mobility functions is an important parameter. Three regions have been exemplified; mass, stiffness and resonant. Though the frequency ranges over which they extend will differ it can be expected that many structures will exhibit these three regions too.

In the regions where coupling between contact points exists accurate estimates of the total active power can only be achieved with accurate force ratio estimates. It is suggested therefore that future work should begin by concentrating on the viability of devising procedures to obtain more accurate estimates of force ratios.

8. ACKNOWLEDGEMENTS

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9. REFERENCES

[1]. Cremer, Heckl, Ungar. Structure-Borne Sound. Springler Verlag, 1973.
[2]. Petersson, Plunt. Journal of Sound and Vibration 82(4). 'On effective mobilities in the prediction of structure-borne sound transmission between a source and a receiving structure, Part 1: Theoretical background and basic experimental studies'.

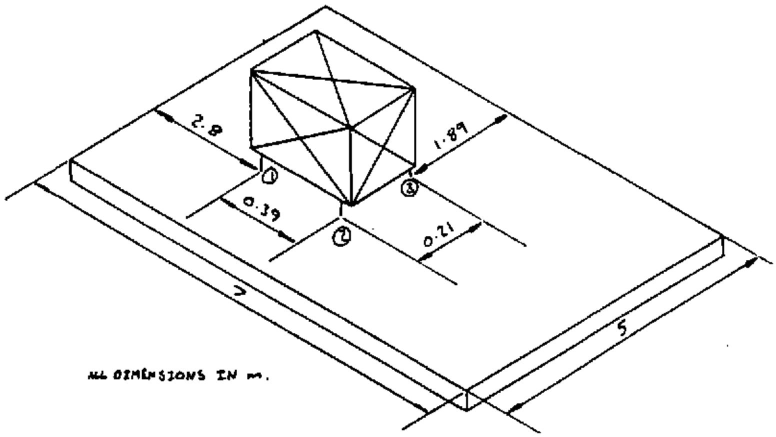

Figure 1: The fan mounted upon the floor.

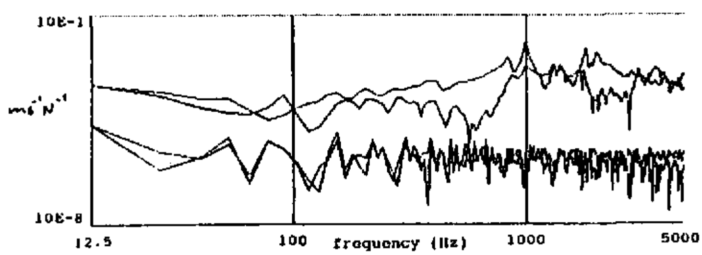

Figure 2: Typical point and transfer mobility magnitudes for both the fan and the floor.

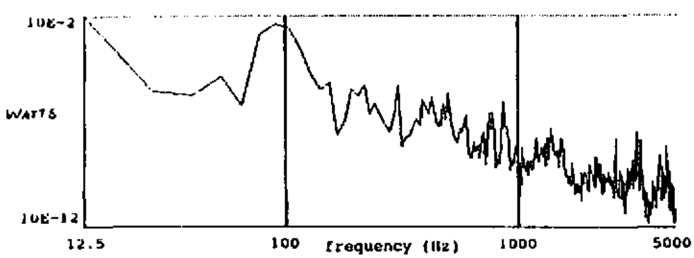

Figure 3: The exact total active power.

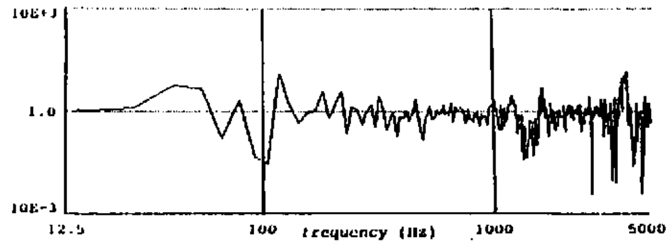

Figure 4: The normalised 'uncoupled' estimate of the total active power.

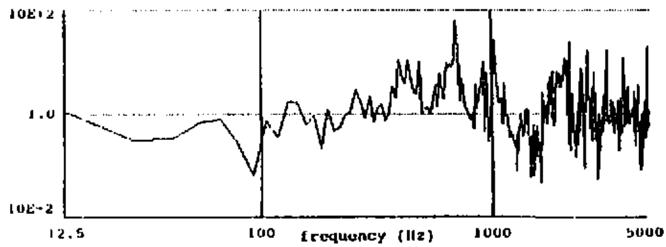

Figure 5: A typical force ratio magnitude.

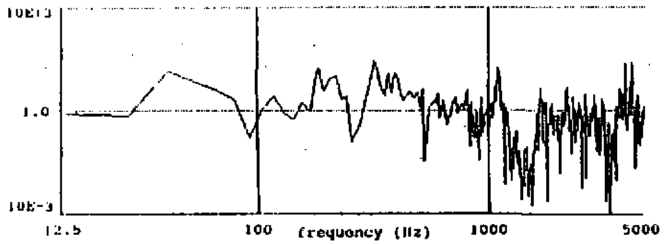

Figure 6: The normalised 'unit magnitude, zero phase' estimate of the total active power.

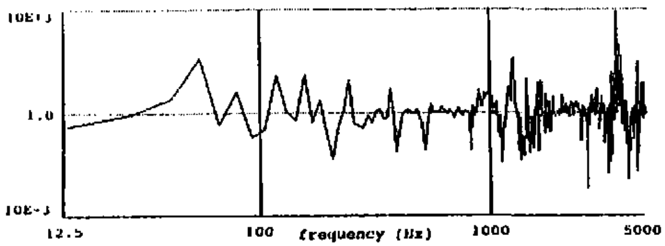

Figure 7: The normalised 'point mobility, free velocity' estimate of the total active power.