**Uncertainties in predicting structure-borne sound power input into buildings**

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There has been a steady development of methods of measurement and prediction of structure-borne noise in buildings, particularly over the last two decades. In proposing and evaluating these methods, a major consideration has been the likely trade-off between accuracy and simplicity. Structure-borne sound transmission is a more complicated process than airborne sound transmission, but practitioners seek methods of prediction for the former, which are as straightforward as for the latter. In this paper a description is given of a study of multi-contact sources in buildings. The study concentrates on measurement and calculation procedures for sources and calculation procedures for receiver structures, particularly lightweight building elements. Although the study is not exhaustive, the findings point to the limitations of simplified methods, specifically the uncertainties likely as a result of reducing the data sets and computational effort, and the discrepancies resulting from simplifying assumptions.

1. **INTRODUCTION**

Building engineers and consultants are able to address airborne noise problems more effectively and confidently than structure-borne noise problems. There are publications on the theory1 and application2 in vibro-acoustics, which provide useful information. However, there remains a shortfall in methods of predicting noise from vibrating machines in contact with building elements, which transmit vibrations and radiate sound, often at distance from the source.

Mechanical and water services become structure-borne sources, when in contact with the building structure, through supporting mounts, services runs and structural bracing. Multiple contacts generate multiple vibration transmission paths and, indeed, the machine should be treated as multiple sources. However, manufacturers consider their products as single entities and seek a single frequency dependent value of structure-borne sound source strength.

This is often (but not always) possible when dealing with airborne sound source strength, the airborne sound power. A washing machine, compressor, fan unit, etc., can be rated with a single frequency dependant value, which provides input to a source-path-receiver model. The path is generally well understood and the sound pressure level at distance from a source of known airborne sound power can be calculated, for example, for open spaces3 and enclosed spaces4, for comparison with appropriate receiver (listener) criteria. There is a menu of standard recommended laboratory methods of obtaining the airborne sound power: by source substitution, in reverberant conditions, in anechoic conditions and by intensity methods. A significant simplification is possible because, in most cases, the airborne sound power of a source is independent of its surroundings. Surface vibrations, which cause acoustic radiation, are not affected by fluid (e.g. air) loading. Further, the mobility of the receiving system, or conventionally the impedance of the air, c, does not vary significantly with location. This independence cannot be assumed for machines in tightly fitting enclosures. However, in most cases, the source obeys the classical velocity-source model.

1. **VIBRATION ACTIVITY, MOBILITY AND STRUCTURE-BORNE POWER**

For structure-borne sound, the source-path-receiver model of airborne sound is replaced with a source-transmission-path-receiver model, where the transmission occurs at the contacts between the sources and receiver structures5. In the same way that the airborne source is given as a power, the structure-borne source strength and the transmission also should be given as powers. This statement has yet to receive general acceptance. It is sometimes assumed that knowledge of the force or the velocity, at the contact between machine and supporting/connected structure, is sufficient to describe the vibration transmission. However, for an inert supporting structure, the force is the maximum deliverable by the machine, but the velocity of the supporting structure is zero. The power (the product of contact force and contact velocity) therefore is zero and also the resultant radiated sound. Conversely, if the supporting structure is highly compliant, the contact velocity approaches the maximum possible, but now the contact force and likewise the power approach zero.

In Fig. 1 is shown the interaction between a vibrating source and a receiving structure as represented by an inverse electric circuit analogy6. The vibrating source is represented by the free source velocity ***VSf****,* i.e. the velocity of the source when freely suspended but otherwise operating under normal conditions, and the source mobility, i.e. the complex ratio of response velocity to a force applied to the source, again when freely suspended. The receiving structure is represented by the receiver mobility.

The power at the contact is the product of the contact force and contact velocity,

 (1)

The two contact terms each relate to the two source quantities and receiver quantity, to give1

 (2)

The power *W* in Eq. (2) is complex, but of interest is the real part, *P = Re(W)*, which corresponds to the flow of energy into the receiving system and ultimately to the radiated sound. Two source quantities () therefore are required, which when combined with one receiver quantity (), give the installed power. The installed power provides the input to noise prediction models, for example, for buildings4.

Of relevance is the concept of the characteristic mobility1. The characteristic mobility of a thin plate-like receiver structure is that of an infinite plate of the same material and thickness and is given by:

  (3)

is the mass per unit area and is the bending stiffness for modulus of elasticity *E*, Poisson’s ratio ** and platethickness *h*. The characteristic mobility is independent of frequency and is real-valued. For ribbed or framed plate structures, the mobility at the reinforcing points can be approximated in terms of the complex characteristic beam mobility7,

 (4)

where bending wave speedand is the mass per unit length.

The source activity, free velocity or blocked force, is rarely predicted, although Breeuwer and Tukker proposed empirical estimates of (free) acceleration levels of resiliently mounted fan units, based on the eccentricity of the rotor and rotational speed8. The international Standard for measurement of resiliently mounted machines gives the free velocity9. Although the free velocity alone is not a sufficient measure of source strength, for resiliently mounted machines it can be used in combination with the spring stiffness of the mounts to give the contact forces. The installed power at a contact then is expressed as10:

  (5)

*Z*12 is the blocked transfer impedance of the resilient mount11.

It is difficult to correctly replicate normal operating conditions in the laboratory, e.g. the operational loads through drive shafts or belts, and a separate measure of an engine, which links to the power train through gears, is problematical. An alternative measure of source activity, the blocked force, can be obtained directly by inserting force transducers between the machine under test and an inert receiving structure12. This poses practical problems of access to the mount locations in situ and the resultant alteration of the operational mount conditions. Indirect methods of obtaining the contact forces are in development13, but are not yet available as standards. For this paper, measured free velocity will form the basis of discussion.

Even with the source activity available in measured form and source and receiver mobility available in measured or calculated form, the data acquisition and processing is computationally intensive. A question remains on if and how the data can be reduced in a useful way and the likely discrepancies, which will result from such reductions. The approach to this question is by case study, using measured source data (free velocity and source mobility) and the mobility of idealised but representative receivers.

**III. REDUCING SOURCE DATA THROUGH CASE STUDIES**

The approach was to investigate the effect of various forms of data reduction and the discrepancies which result. The effect is quantified by comparing the approximate powers with benchmark (simulated total power) values, obtained with full complex mobility data sets. The measured source data was used for several sources ‘connected’ to idealised but representative receivers, described by the characteristic plate mobility (Eq.(3)).

The free velocity was recorded at the resiliently supported contacts, while the selected machines were operating under typical conditions. Likewise, the point and transfer mobilities were recorded for the resiliently suspended condition.

The data reductions and simplifications are considered systematically, by: using magnitudes rather than complex values; reducing data into single equivalent values; neglecting transfer terms and using point mobility only. In addition, consideration is given to simple calculations of source and receiver point mobility. The benchmark powers were for three fan units, each on four mounts of: a flange base, a plate base and a frame base. The free velocity at each support point of each source was recorded while the fans were operating, while resiliently suspended. A single sample of velocity, for each source contact, only was required, since the comparison of approximate powers with the benchmark powers cancels the velocity. Therefore, repeatability and the effects of operational drift are not considered. However, it will be demonstrated that where the spectrum of the free velocity, and thus the transmitted power, indicate tonal components, then phase effects can assume importance at the associated frequencies.

In Fig. 2 is shown the point and transfer mobilities of the three sources. The mobilities were measured, using shakers and in-line force transducers, and matched accelerometer pairs. The accuracy of the measurements was not precisely determined but comparisons of reciprocal measurements of transfer mobility between contact points, e.g.compared with, gave agreement within +/-1 dB, in one-third octave bands, and within +/-3 dB at the lowest and highest frequency bands. The frequency resolution of the narrow-band data is 12.5 Hz. The acquired mobilities are complex but are presented in Fig. 2 as magnitudes. Point mobilities are shown as solid lines; transfer mobilities, between the contacts, are shown as dashed lines.

There is a wide variation in dynamic behaviour, even in this small sample, but some general observations are possible. In the rigid body region, such as for the flange base below 100 Hz (Fig. 2(a)), the transfer and point mobilities are of the same order of magnitude. In the stiffness controlled region, again as for the flange base, between 100 Hz and 1000 Hz, the transfer mobilities are significantly lower than the point mobilities. At the onset of the resonance controlled region, see all three sources, the transfer mobilities are of the same magnitude as the point mobilities, but eventually decrease with increased frequency, relative to the point mobilities.

Infinite plates were considered as idealised receivers, with real valued frequency invariant characteristic mobility. A high mobility receiver condition was assigned the value of characteristic mobility of 10-3 m/sN (see Fig. 3, which also shows the point mobility at four mount points of the flange base of Fig. 2(a)). This corresponds to contact or support points on boarding (bays) between timber joists or frames. A mid mobility receiver condition was assigned the characteristic mobility of 10-4 m/sN. This corresponds to contact or support points over ribs (e.g. timber joists or framing) at low frequencies. A low mobility receiver condition was assigned the value of characteristic mobility of 10-5 m/sN. This represents an upper limit for heavyweight concrete floors and masonry walls.

The benchmark total power (simulated for forces normal to the receiver surface) was obtained by reference to the concept of the effective mobility14. The effective mobility is based on the premise that the transmitted power can be obtained for each contact between the source and receiving structure, but where the influence of all other contacts is included. The total power from a source  to a receiver  then is the sum of the powers through  contacts:

 : (6)

The superscript denotes *effective point mobility*, so that contributions from all other contacts are taken into account. The effective point mobility at the *ith* contact is:

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|  |  |  (7) |

The first term on the right hand side of Eq. (7) is the point mobility at the *ith* contact of interest. The second term is the sum of the transfer terms (i.e. the contributions to the velocity at the contact of interest from the forces at the other contacts). The formulation gives the total power if the complex contact force ratios are known. The contact force is approximated by the free velocity and source and receiver point mobilities at the same contact:

 (8)

In Fig. 4 is shown the powers for the three source-receiver conditions of Fig. 3, again for source (a) in Fig. 2. The frequency range shown is 10 Hz – 5 kHz, but the power above 1 kHz is more than 50 dB (10log) below the peak at 100 Hz. The other sources gave similar decreases with frequency and so the audible frequency bandwidth was set at 50 Hz – 2 kHz in subsequent discussions. The powers are not only a function of the real part of the effective receiver mobility of Eq. (6), but also of the interaction between the complex source and receiver effective mobilities (see denominator in Eq. (6)). For example, the mid mobility receiver power at 100 Hz is the same as the high mobility receiver power. This is because there is mobility matching in the former and not in the latter.

This highlights the importance of establishing the relationship between source and receiver mobility and its effect on the installed power. If the receiver mobility is significantly higher than the source mobility (), Eq. (6) reduces to the approximation:

 (9)

All that is required of the source is a measure of its free velocity and the source is termed a velocity source.

If the receiver mobility is significantly lower than the source mobility (), Eq. (6) reduces to the approximation:

 (10)

This is a special but common situation in heavyweight buildings. All that is required of the source is a measure of its blocked force and the source is termed a force source.

When do these simplifying assumptions apply? In Fig. 5 are shown assumed velocity source (Eq.(9)) and force source (Eq.(10)) powers, normalised with respect to the benchmark power, for source (a) in Fig. 2, into a high mobility receiver (left) and a mid mobility receiver (right). Both source assumptions give significant discrepancies, depending on the source-receiver mobility ratio and neither applies over the whole frequency range of interest or for the two installations. Again, the source and receiver mobility must be known in an appropriate form.

At each contact, up to six components of excitation and response are possible (three translations and three rotations)and their relative contributions to the power transmitted into the total structure (e.g. building) are not easily assessed15. Translational forces parallel to the receiver surface and torsion about the axis perpendicular to the receiver surface generate in-plane fields, which transform into bending fields at junctions with connected plates. In addition, forces parallel to the surface of thick plates, and therefore distant from the neutral axes, generate rotational components. These components are neglected in this study because of the very small values of receiver mobility associated with them1. Moments can assume importance when sources are located close to structural discontinuities15. Machines are often located near to reinforcing ribs and edge junctions, for increased support, particularly in lightweight structures, and moments cannot be neglected a priori. In heavyweight buildings, moment induced power increases in relative importance, with decreased distance from plate edges and with increased frequency, but force induced power generally is dominant16. For lightweight receiver structures, consider the example in Fig. 6. Measured source and receiver data has been assembled for a medium size fan on a timber joist floor17. This has allowed the calculation of powers from moments, about axes perpendicular to the joists (Mx) and also parallel to the joists (My), normalised with respect to powers from perpendicular forces. Whilst moments can assume importance at some frequencies, such as in the range 630 – 1 kHz, for the example shown, overall, the total power is satisfactorily approximated by the force induced power. Despite the above comments and caveats, perpendicular forces are assumed dominant and are considered only in this study.

1. **Approximate power using magnitudes**

So far, the powers presented have been calculated using complex values of source mobility (the receiver mobility has been the pure real characteristic value). If band average values are to be used, this raises the issue of how and if phase can be spectrally averaged in a meaningful way. This can be circumvented by expressing the mobilities as magnitudes and also the real part of the receiver mobility. The free velocity squared also is real valued. Eq. (6) becomes:

 (11)

Eq. (11) approximates Eq. (6) only if the source mobility and receiver mobility are significantly different, i.e. a level difference of 10 dB or more. There are maxima in power when the source and receiver mobilities are complex conjugate18,19. In Fig. 7 is shown the total powers, using magnitudes (Eq. (11)), normalised with respect to the powers calculated from Eq. (6), for the sources shown in Fig. 2. Narrow band values are indicated for the frequency range 50 – 400 Hz and one third octave values are indicated for 200 – 2kHz.The approximations are within +/- 2 dB of the benchmark value, except at frequencies where the source activity (free velocity) has a strong tonal component and/or where there is source-receiver mobility matching. This is exemplified by the flange base (source (a) in Fig. 2). The dip in normalised power at 90 Hz corresponds to mobility matching with the mid mobility receiver (see Fig. 3) and a tonal peak in the power spectrum (see Fig. 4), also in the free velocity spectrum, not shown. Likewise, the dip at 175 Hz corresponds to mobility matching, again indicated in Fig. 4, and the harmonic peak in the free velocity spectrum. In these situations, phase effects assume importance but are not included in the approximate method, which therefore gives underestimates of the order of 5-10 dB. Despite these discrepancies, calculations with magnitudes warrant further consideration, since their use points to data acquisition and computation in octave and one-third octave bands favoured by practitioners. The Standard for measurement of the free velocity of resiliently mounted machines yields one third octave data9. A recent standard for obtaining source power by a reception plate method20, for incorporation into prediction models4, also deals with one third octave values.

1. **Approximate power using equivalent single values**

 Can multiple connected sources be treated as single sources? In order to describe the source before installation, the contact force ratio is required (see Eq. (7)). However, the forces are only manifest in the installed condition. For the source, before installation, some assumption of the force distribution is necessary21. The contact forces can be assumed to be of equal magnitude, to give a unit force ratio. The phase difference between forces depends on the vibration behaviour of the source. If the source has a rigid body behaviour and is in the bouncing mode, then a zero phase difference is assumed. Eq. (7) becomes14:

 (12)

If a random phase difference between contact forces is assumed, the real part of the effective mobility is approximated by the real part of the associated point mobility:

 (13)

The magnitude of the effective mobility is given by a sum of squares:

 (14)

The advantage of the random phase assumption is that it yields real valued quantities and therefore is amenable to band averaging. The random phase assumption is used in the following discussion.

In seeking equivalent single values of multi-contact sources, reference is made to the reception plate method of indirectly measuring structure-borne power. With the operating machine attached to an isolated plate, the total transmitted power equals the energy loss of the plate and the latter is obtained from the product of the spatial average of the mean square plate velocity, the plate mass and total loss factor22. If the source is attached to a high mobility plate, it is shown that the sum of the square free velocities, over the contacts is obtained indirectly23. Therefore, the equivalent single source velocity is given as a squared value. Further, if the source is attached to a low mobility plate, the data obtained, in combination with the equivalent single source velocity, yields the equivalent single value of source or receiver mobility as the average of the effective mobilities, over the contacts24, . Eq. (2) for the total power is now expressed as:

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| --- | --- | --- |
|  |  | (15) |

In Fig. 8 are shown the powers from the three sources, each for three characteristic receiver mobilities, where the approximate values from Eq. (15) are normalised with respect to the powers calculated from Eq. (6).

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| The discrepancies resulting from this second step in data reduction are greater than for the first step, i.e. use of magnitudes. The approximation generally is within +/-5 dB of the benchmark value, but with large negative discrepancies at low frequencies, corresponding to source tonal behaviour and/or matched source-receiver mobility conditions, now combined with the loss of detail on the complex interaction between contact forces. 1. **Using point mobilities only**

If the effective point mobility of Eq. (7) is replaced by the point mobility, this is equivalent to treating the four contacts of the considered fan units as independent sources. In Fig. 9 are shown the resultant normalised approximate powers from the three sources. The discrepancies resulting from this third step in data reduction also are within +5 dB, but again with significant discrepancies at low frequencies for reasons given earlier. In a recent study, the same source data of the fan flange base (Fig. 2(a)) has been combined with measured receiver data of a timber joist floor17. The total power through the four contacts was calculated for ten locations, including between joists, on joists and intermediate positions. In Fig. 10 are shown the approximate powers, again based on equivalent single values and with point mobilities only, normalised with respect to the exact values. Results are presented as one third octave values. The floor generally behaves as a high mobility receiver and the approximate power gives on average an overestimate similar to that observed in the present study (Fig. 9). Likewise, the range of normalised powers, over the ten locations, is of the same order as the present study. These results confirm the findings of Ohlrich25.1. **Discussion**

Although the reductive approach adopted in this investigation is systematic, it has not been exhaustive, being based on relatively few case studies. Measured source free velocity and mobility have been used but the receiver conditions have been idealised in terms of characteristic mobilities. Neither has this paper provided full physical explanations of the causes of the uncertainties resulting from each step in the data reduction, although causes have been indicated, such as due to matching mobility, tonal characteristics and lack of phase information. Practitioners, seeking accuracies of the order of +/-3 dB may be disappointed in the findings, but accuracies of the order of +/- 10 dB (as one third octave values) may be acceptable, in the absence of an accepted method of measurement and calculation. However, again larger discrepancies may be encountered at low frequencies, when expressed as narrow band values.1. **SIMPLIFICATION BY CALCULATING RECEIVER AND SOURCE MOBILITY**

As an alternative to measurement of mobility, consideration is given to calculating values. Again, it is assumed that measured free velocity data would be available as octave or one third octave values. The calculation methods proposed are relatively simple and are to establish the source-receiver mobility ratio and thus the appropriate data reduction(s).  1. **Receivers**

Receiver structures in buildings generally are supporting plates, e.g. walls and floors. Plate structures can be further categorised as homogeneous (heavyweight or lightweight), framed or ribbed (usually lightweight). In Fig. 11 is shown the measured point mobility of a thick homogeneous plate (resiliently supported 100 mm concrete plate of dimensions 2.8 m by 2 m) along with calculation, based on modal summation and beam functions6. The response is modal with peaks and dips in receiver mobility, converging with increase in frequency to the characteristic value. Analytical and numerical models, which give the resonant peaks and dips in mobility and the frequencies at which they occur, require detailed information on the edge conditions, which is seldom available for buildings. However, it is possible to establish upper and lower limits to the mobility26, where:  (16)*M* is the plate mass and ** is the total loss factor. The limits are shown in Fig.11. The associated upper and lower limits in installed power, from an attached source, are then obtained26. In general, the receiver mobility of homogeneous floors and walls can be predicted from thickness and relatively easily obtained material properties, both for heavyweight and lightweight structures. In the latter case, the thin plates behave as infinite plates over most of the frequency range of interest and the characteristic mobility applies. Examples of ribbed plates and framed plates are timber-joist floors and timber or metal studding partitions. The point mobility will vary with location, depending on if it is at a frame or rib, or in bays between the reinforcement. In Figure 12 is shown the measured point mobility of a range of timber floors and light-weight cavity walls7. The point mobility was measured at and near to ribs and framework, and in the bays between ribs and frames. Apparently different lightweight buildingconstructions have similar dynamic behaviour. For many locations and constructions, the behaviour can be described by the characteristic mobility of the covering board (plasterboard, floor boarding, etc.). Further, the point mobility of these different lightweight constructions converges with frequency, towards a characteristic value of 10-3 m/sN, which may serve in preliminary calculations. This assumption is as useful as that for heavyweight structures, where an initial estimate of point mobility of 10-5 m/sN is often used. However, not all locations on ribbed plates behave as an infinite plate, particularly at low frequencies. For example, see the point mobility at floor joist in Fig. 12 (a). Such locations can be described in terms of the characteristic beam mobility, also shown, the magnitude of which has a value of about 10-4 m/sN over the low frequency range of interest. For locations, at distance from the beams and frames, there is a monotonic transition between beam behaviour and plate behaviour. This is more clearly observed in Fig. 13, for three lightweight structures. The measured point mobility, at different locations, is shown, normalised with respect to the characteristic plate mobility of the covering plate, and plotted against distance relative to the governing bending wavelength of the plate. In general, ribbed/framed plates behave as beams at locations less than 0.1 bending wavelengths from a rib, and behave as plates at locations more than 0.25 wavelengths from a rib, with a straight-line transition between these two regions (see dotted lines in Fig. 13)..1. **Sources**

Mechanical installations are generally more complicated than building elements and provide a wide range of structural types. However, the mobility at the contact points is largely dictated by the material and geometry of the machine base around the contact. On this basis, it is possible to categorise machines bases as: compact sources, flange bases, plate bases, frames. In Fig. 14 is shown the mobility of a compact source, a domestic heating circulation pump of mass 2.65 Kg. In this case, rigid body (RB) behaviour is indicated over most of the frequency range of interest26. The source may move in a rocking, as well as a bouncing mode, and the expression for mobility will depend on geometric factors. The expression for rigid body mobilityis27:  (17)where *x,y* are the distance of the contact point of interest from the centre of gravity of the source and andis the moment of inertias about the x and y axis, respectively. If the source is modelled as a solid of height *H,* with a square base *L* x *L*, then;  (18)The point mobility of a flange base is exemplified in Fig. 3. Again, rigid body behaviour is evident below 80 Hz. Above 80 Hz, the mounts begin to flex, to give the stiffness controlled (SC) region, up to 1 kHz. In this region, the mobility increases 6 dB per octave. Above this region, the mounts behave as a resonant plate, the resonance controlled (RC) region. The mobility then converges to the characteristic plate mobility at high frequencies, which is 1.2 x 10-3 m/sN for the 3 mm thickness steel, which forms the flange base. The onset of resonance controlled behaviour is at the first resonant frequency, which depends on the plate size, thickness, material and edge conditions30,31. This suggests a procedure for constructing a source mobility ‘trend’ curve:1. Calculate the characteristic mobility of the base plate at the mount/connection;
2. Calculate the first resonance frequency of the mount flange or plate
3. Interpolate from the first resonant frequency, with reducing frequency at -6dB per octave, to form the stiffness controlled region
4. Calculate the rigid body mobility from the machine mass and geometry and intersect with the stiffness line.

In Fig. 3, the mobility trend is shown (dashed line) for a 3 mm flange base of resonance frequency 1400 Hz. This approach is less comprehensive than that of Petersson and Plunt31, since resonant behaviour (peaks and dips) is not indicated, but provides information to establish the source-receiver mobility ratio, when the receiver mobility is available, of course. There is a similar mobility curve for a fan plate base (Fig. 2(b)) in Fig. 15 but where the transition from the stiffness controlled to resonance controlled regions occurs at a lower frequency. Of special interest is the transition between the rigid body and stiffness controlled regions, the anti-resonance, since the source mobility can be significantly lower than the receiver mobility in lightweight building structures.In Fig. 16 is shown the mobility at eight mount points of a supporting frame of a whirlpool bath32. The mobility varies significantly with mount point, particularly below 100 Hz, since contact geometries (distances from free ends to frame junctions, overlapping framing, etc.) differ. In general, it is not possible to construct an approximate mobility curve for frame bases. However, for the example shown, the measured values converge to the characteristic beam mobility of the rectangular section frame.1. **Predicted powers from calculated source and receiver mobility**

It remains to assemble the equivalent single values of source and receive mobility, based on calculated point mobilities. The source mobility was estimated according to the procedure outlined in the previous section. The receiver mobility was estimated according to that outlined for ribbed/framed plates. From these estimated mobilities, the upper and lower limit to the installed power is predicted. The upper limit corresponds to source locations in bays, between joists. The lower limit corresponds to locations on joists. The limits are shown in Fig. 17, along with the powers at ten locations, obtained from measured complex source and receiver point and transfer mobilities. Values of the exact power exceed the upper limit and are below the lower limit, the latter particularly at low frequency. However, the approximation provides a working estimate of the spatial range.1. **CONCLUDING REMARKS**
2. Measurement procedures and calculations, which yield data as band–average magnitudes, can be used for preliminary predictions of structure-borne sound from multiple-contact sources in buildings. From the cases considered, accuracies of the order of +/- 3 dB are attainable but are of the order of +/- 10 dB in one third octave bands, where there is source-receiver mobility matching and/or where source activity has tonal components.
3. Multiple contact sources have been considered as equivalent single sources, based on the sum of squared free velocities, over the contacts, for the source activity, and average values of the effective mobility, over the contacts, for the source and receiver. When combined with the first step in data reduction (the use of magnitudes), accuracies of the order of +/- 5 dB result, but again with large negative discrepancies in narrow frequency bands, due to some or all of the following: source tonal behaviour, matched source-receiver mobility conditions, complex interaction between contact forces.
4. Assuming the contacts act as independent sources, i.e. by neglecting transfer terms between contacts, reduces the data acquisition and computational effort significantly. However, for the cases considered and incorporating the two previous steps in data reduction, the discrepancies are positively biased for high mobility receivers, but otherwise display a negative bias. However, the discrepancies are not significantly greater than for the previous reductive steps.
5. Although source activity (free velocity or blocked force) must measured, source and receiver mobility can be calculated by reference to characteristic dynamic behaviour. The receiver point mobility, for plate, ribbed plate and framed plate structures, can be estimated, based on infinite plate and infinite beam behaviour. For plates displaying modal behaviour, upper and lower limits to the receiver mobility can be calculated and thence upper and lower limits to the predicted installed power.
6. For sources, the mobility is characterised in terms of rigid body (RB), stiffness controlled (SC) and resonance controlled (RC) motion. Simple skeleton mobility curves can be constructed based on these behaviours.
7. A practical approach to the prediction of structure-borne sound power of mechanical installations in lightweight buildings is described, which is based on calculated mobilities. The case study of a fan unit on a timber joist floor has applications to wall-mount installations, which are usually attached to the frame or ribs. This could provide separately the power into the structural frame and into the cladding, as a first step in the prediction of structure-borne sound propagation in lightweight buildings.
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**Figure legends**

FIG. 1: Inverse analogous electric circuit, to give contact force and contact velocity.

FIG. 2: Magnitude of point and transfer mobility at four mount points of: (a) flange base, (b) plate base, (c) frame base

FIG. 3: Magnitude of point mobility at four mount points of fan unit on a flange base and three receiver conditions, represented by values of characteristic mobility.

FIG. 4: Total structure-borne power through four mount points of a flange base on high (solid line), mid (dashed) and low (dotted) mobility plate receivers

FIG. 5: Normalised power for force source (solid line) and velocity source assumption (dashed): left: into a high mobility receiver; right: into mid mobility receiver.

FIG. 6: Moment induced powers from a fan unit on a timber-joist floor, normalised with respect to the perpendicular force induced power17.

FIG. 7: Level differences between approximate powers, based on magnitudes, and benchmark powers for three sources into high (solid line), mid (dashed) and low (dotted) mobility receivers.

FIG. 8: Level difference between approximate powers, based on equivalent single values, with magnitudes, and exact powers.

FIG. 9: Level difference between approximate powers, based on equivalent single values and point mobilities, and exact powers.

FIG. 10: Normalised approximate powers, based on equivalent single values and point mobilities, for a flange base at ten locations on a timber joist floor17.

FIG. 11: Point mobility of resiliently supported concrete plate with free edges22.

FIG. 12: Measured magnitudes of point mobility of lightweight structures7.

FIG. 13: Normalised point mobility of: (a) 12.5 mm plasterboard wall with U-section metal frame, (b) 9.5 mm plasterboard 12.5 chipboard wall with 120x60mm timber frame, (c) 21 mm chipboard with 192x96mm timber joists

FIG. 14: Measured and calculated magnitude of point mobility of a domestic circulation pump26.

FIG. 15: Magnitude of point mobility at four support points of a fan plate base. Shown (dotted) is the approximate curve.

FIG. 16: Magnitude of point mobility at eight support points of a frame base30. Shown (dotted line) is the characteristic beam mobility.

FIG. 17: Total structure-borne powers at ten locations of a fan unit on a timber joist floor, with predicted upper (dashed) and lower (dotted) limits.