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5 TITLE: ACOUSTIC DEVICE COMPRISING A PANEL MEMBER RELYING ON BENDING WAVE ACTION

10 FIELD OF THE INVENTION

This invention relates to acoustic devices capable of acoustic action involving bending waves.

BACKGROUND TO THE INVENTION

Co-pending International Patent Application PCT/GB96
15 /02145 (published W097/09842) includes various teaching as to nature, structure and configuration of acoustic panel members having capability to sustain and propagate input vibrational energy through bending waves in operative area(s) extending transversely of thickness usually (if not
20 necessarily) to edges of the member(s). Detail analyses are made of various specific panel member configurations, with or without directional anisotropy of bending stiffness across said area(s), so as to have resonant mode vibration

components distributed over said area(s) beneficially for acoustic coupling with ambient air. Analyses extend to predetermined preferential location(s) within said area(s) for transducer means, particularly operationally active or moving part(s) thereof effective in relation to acoustic vibrational activity in said area(s) and signals, usually electrical, corresponding to acoustic content of such vibrational activity. Uses are also envisaged in the above PCT application for such members as or in "passive" acoustic devices, i.e. without transducer means, such as for reverberation or for acoustic filtering or for acoustically "voicing" a space or room. Other "active" acoustic devices, i.e. with bending wave transducer means, include a remarkably wide range of loudspeakers as sources of sound when supplied with input signals to be converted to said sound, and also in such as microphones when exposed to sound to be converted into other signals.

Co-pending International Patent Application PCT/GB98/00621 concerns applying to panel member(s) distribution(s) of stiffness(es) and/or mass(es) not centred coincidentally with centre(s) of mass and/or geometrical centre(s). This is particularly (but not exclusively) useful to beneficially combining both pistononic acoustic action (as

for hitherto conventional, typically cone-type, loudspeakers) with bending wave acoustic action generally as in the above published PCT application. Specifically, location(s) of transducer means for both pistonic and bending wave actions can include at centre(s) of mass and/or geometrical centre(s) (as very much suits pistonic action), but still satisfy general desiderata for bending wave action.

This invention has arisen from intuitive feeling that various approaches of the above PCT applications to design and specification of acoustically useful bending wave action members reflect some other useful concept/methodology that should be capable of yielding as good or yet better and/or as practical or more practical design/specification criteria, perhaps including other useful configurations and transducer locations not before specified or otherwise appreciated. It has been an object of this invention to investigate, and arrive at such results.

SUMMARY OF THE INVENTION

According to first general method and device aspects of this invention, panel member parameters affecting bending wave action, such as particularly configuration/

geometry in relation to bending stiffness(es) and/or bending wave transducer location(s), is/are in accordance with desiderata applied to analysable characteristic(s) relevant to power transfer for the acoustic device
5 concerned, such desiderata usefully favouring acceptable distribution and/or density and/or evenness of excitation of acoustically relevant resonant modes of surface vibration involved in bending wave action.

It has been particularly established that desirably
10 effective resonant mode density/distribution correlates with a measure of smoothness of power transfer for the acoustic device concerned; and use and results of such correlation in terms of acoustic panel members involving bending wave action constitute various other aspects of
15 this invention.

Underlying inventive rationale or concept involved includes appreciation that, for active acoustic devices as sources of sound, satisfactory acoustic performance of panel members concerned is more dependent on smoothness of
20 power output than on hitherto conventionally esteemed flatness of output over whatever frequency range is concerned/desired. Deviation from flatness of output is actually readily compensated by suitable electronic signal

conditioning, specifically so long as the output deviations concerned are reasonably smooth.

Energy losses within panel members and transducer means of acoustic devices concerned tend to be both
5 relatively small and reasonably smooth in themselves. Accordingly, for the purposes hereof, effectiveness of device design and specification can be based on smoothness of input power transfer, including particularly as to geometry/configuration such as aspect ratios and as to
10 bending wave transducer location(s) such as in terms of proportionate co-ordinates.

Whatever particular characteristic(s) is/are involved in assessing smoothness of power transfer, conveniently and preferentially input power transfer, it is practical to be
15 concerned with deviation from some useful condition, state or value, whether of arbitrary or of relational nature. Thus, analysis relative to same or unity weighting of whatever resonant frequency modes are concerned has produced useful results, as has analysis relative to mean
20 value(s). However, selective adjustment of weighting etc is also seen as useful refinement, for example at least for end-most modal frequencies involved, particularly lowest; and feasibly more generally or otherwise.

The frequency modes concerned/involved in analytical assessment hereof can be as arise from making practically viable simplification, such as using analogies of one-dimensional nature, say to orthogonal beams notionally in directions parallel to pairs of opposite sides of substantially rectangular panel members. This simplification approach reflects success achieved in specific teaching of W097/09842, including first consideration relative to a number of resonant modes in each beam direction and directly related inter-active modes. Refinements of analyses relative to two-dimensional relationships should more closely reflect realities of panel members as such, including revealing and taking appropriate account of more inter-actively related resonant modal frequencies.

Preferred said characteristic(s) relevant to power transfer for the panel member include criteria for mechanical impedance, say as to standard deviation with application of a smoothing factor, say 10%.

In some particular inventive aspects hereof, criteria for mechanical impedance are used in assessing input power transfer, specifically in finding practical geometries and/or stiffness parameters/distributions of panel members

for acoustic action relying on distribution of resonant modes of bending wave action. It can be of high practical value first to investigate relative to known favourable transducer locations and to present results functionally, usefully graphically, relative to variant aspect ratios of general geometrical shape concerned in looking for minima of deviation.

In other particular inventive aspects hereof, criteria for mechanical impedance is/are used to find practical transducer locations for particular desired geometries/ configurations and/or stiffness distributions of panel members for acoustic action involving bending waves, specifically and advantageously without limitation to panel members having favourable geometry/configuration such as available from said some inventive aspects. It can be of high practical value to investigate variable one relative to fixed other of co-operative areal locators such as co-ordinates of transducer location and present results functionally, usefully graphically, in looking for minimum deviation of preferably smoothed mechanical impedance. It can also be of high practical value to present results of this investigation of panel members as areal distribution of mechanical impedance or deviation thereof, conveniently

in contoured manner to indicate extremes and gradations between, and for which it is a matter of choice whether to apply selected values and/or to normalise relative thereto, or to do no more than have relative step-wise gradations indicate at least best and worst locations, say within 10% or less steps.

In further aspects of invention hereof, geometries promising for acoustic action involving bending waves are investigated using a measure of mechanical impedance for promising transducer locations, and such promising geometries are further investigated in relation to use of such promising transducer locations, such investigations being capable of application cumulatively/successively/recursively for any desired degree of further refining of both of promising geometrical parameters and promising transducer location parameters.

For substantially rectangular panel members and methodology, simplification based analyses involving superposition of orthogonal beam-type functions, and with reference to 10% smoothness criteria for mechanical impedance, have confirmed and refined calculation for one known preferred aspect ratio, specifically 1:1.134 as taught in above published PCT application, to be at about

1.138:1; and refined proportionate co-ordinates for
transducer location $(4/9, 3/7)$ thereof to about
 $(0.440, 0.414)$. In addition, however, and starting from
substantially the same transducer location co-ordinates,
5 analyses hereof have revealed another promising aspect
ratio, specifically at about 1.41 to about 1.47. In
practice, particular investigation of 1.47 aspect ratio
with transducer locations substantially at proportionate
co-ordinate position(s) $(4/9, 4/9)$ led by cumulative
10 refinement to aspect ratio 1.41 and transducer co-ordinate
locations 0.455, 0.452; indeed, to appreciation that there
may be considerable inter-relationship between these 1.41
to 1.47 aspect ratios and variant transducer locations.

It is a particular inventive aspect hereof that a
15 substantially rectangular panel member (as or in an
acoustic device and relying on 'bending wave action) and
substantially isotropic as to its bending stiffness in at
least two directions has an aspect ratio of about 1.41:1 to
about 1.47:1; and another particular aspect of invention
20 that proportionate co-ordinate transducer location(s)
involve substantially 0.453 and/or substantially 0.447.

Moreover, two other reasonably promising aspect ratios
have also emerged from further development of simplified

beam type analyses, namely about 1.6 and about 1.2, together with viable transducer locations at (0.41, 0.44) and (0.403, 0.406), respectively; again with expectation of useful inter-relationships between particular aspect ratios and particular transducer locations.

It has further been established for the purposes of this invention that, perhaps particularly for panel members of favourable geometries/configurations, including such variations as known to arise from anisotropy of bending stiffness(es), the above attainable high specificity as to transducer locations amounts to refined determination within more extended areas that are generally favourable in terms of transducer locations. Indeed, there is strong correlation between size of such areas, particularly medial but off-centre for panel members with isotropy of bending stiffness, and favourability of geometry/configuration, thus between what might be termed truly significant high specificity and unfavourability of geometry/configuration. At least for the latter, it can be particularly valuable to utilise accompanying analyses by scrutiny of power output with frequency and/or finite element analysis (FEA) at least to assess low frequency modality, say as indicative of start positions for analysis of transducer location as

above (or below) and/or of overly intrusive resonant modes for useful correction by localised clamping/damping or for compensation by signal conditioning. Interestingly, for favourable substantially rectangular geometry/configuration
5 viable edge adjacent transducer locations are indicated on basis of mechanical impedance characteristics/desiderata.

The above-indicated alternative techniques utilising inherently two-dimensional analysis, also in terms of mechanical impedance, generally confirm efficacy of above
10 aspect ratios and transducer locations, including promising relatively discrete and extended areas, whether or not for hitherto favoured aspect ratios, thus efficacy of such methodology and results with manifest merit of a general nature even including converse approaches identifying
15 particularly poor areas to be avoided for transducer location and/or aspect ratios of low prospects (albeit then capable of indicating possibly or likely viable, or best attainable singly or in combination, transducer locations in unfavourable geometries).

20 It is of particular practical interest that hitherto known least promising or worst cases of most symmetrical geometries, such as isotropic as to bending stiffness within square or circular boundaries, and substantially

central locations of transducers, continue to be indicated as poor combinations, but that much more or most promising transducer locations can now be identified even to the point of viability at least for perhaps relatively limited frequency ranges and output responses.

Inventive methodology hereof and results obtainable can take account of boundary conditions ranging from free or only lightly damped to more strongly damped and constrained including clamped for which promise is, if anything, now highest (and practically highly beneficially so in relation to actual physical implementation and presentation of acoustic devices hereof, particularly in or as panel-form loudspeakers).

BRIEF DESCRIPTION OF THE DRAWINGS

Exemplary specific implementation of methodology embodying this invention, including results thereof, is now described and detailed with reference to the accompanying diagrammatic drawings, in which:

Figures 1 is an outline diagram indicating basis of specific implementation hereof;

Figure 2 indicates rationale(s) of analytical processing hereof;

Figures 3A and 3B are graphical representations of

mechanical impedance with frequency in substantially rectangular isotropic panels starting with selected aspect ratios;

Figures 4A, B and C are graphical illustrations of a
5 measure of smoothed mechanical impedance (deviation/
variation) for particular transducer locations to indicate
useful aspect ratios of rectangular panels;

Figures 5A - D are graphical illustrations for one
previously known particular panel aspect ratio and known
10 values of one transducer location co-ordinate to
investigate value of the other co-ordinate;

Figures 6A - D are graphical illustrations for another
previously unknown particular panel aspect ratio and known
values of one transducer location co-ordinate to
15 investigate values of the other co-ordinates;

Figures 7A and 7B are generally similar to Figure 3
but starting with other selected aspect ratios;

Figures 8A - D are generally similar to Figure 4
showing confirmation of aspect ratios previously indicated
20 as useful (Figures 8A, B) and also indicating further
promising aspect ratios;

Figures 9A - D are areal contour plots of mechanical
impedance demonstrating transducer location co-ordinate

determination for panels with aspect ratios indicated in previous Figures;

Figures 10A, B are quarter-panel areal contour plots for smoothness of mechanical impedance for the aspect ratios of Figures 6A - D;

Figures 11A, B and 12A, B and 13A, B are also generally similar to Figures 3A, B but for boundary conditions in which all panel edges are clamped;

Figures 14A - C are generally similar to Figure 4 but related to Figures 11, 12, 13 and location of promising aspect ratios;

Figure 15 is similar to Figures 10A - D relative to the aspect ratio of Figure 13A;

Figure 16 shows graphical comparison of the frequency responses of various aspect ratio panels, including those of Figures 11, 12 and 13;

Figures 17A - T are quarter-panel contour plots of mechanical impedance obtained by full two-dimensional analysis/methodology;

Figure 18 is a larger scale quarter-panel contour plot of mechanical impedance for longest known favourable aspect ratio 1.134; and

Figure 19 is a corresponding three-dimensional plot.

PARTICULAR EMBODIMENT(S) OF THE INVENTION

In Figure 1, an active acoustic device, specifically a distributed mode acoustic panel member complete with exciting transducer(s) is represented by block 10, basically as a "black box" with electrical input 11 shown from such an audio amplifier, acoustic output 13 shown in phantom for in-principle completeness in equivalent electrical terms as driving resistive impedance Z_{air} , and indication of intrinsic losses also in electrical terms as resistive leakage path 14 to ground.

By its nature as a structure sufficiently stiff to support bending wave action and afford useful acoustic coupling to air, a resonant mode acoustic panel component of "black box" 10 will have low loss. Also, bending wave transducers along with usual couplings to such panel generally have low losses; and overall loss represented by path 14 tends to be low, at least compared with input and output power at 11, 13 - which would be good for proposed analysis whether or not smooth, but does also tend to be reasonably smooth thus further beneficial.

Figure 2 is believed to be helpful for understanding basis of analytical assessment for which worked examples will be given relative to later Figures. Block 21

indicates a first useful exercise to some extent common to the above-mentioned published PCT application, specifically looking at spacings of resonant mode frequencies. Indeed, such inspection based on angled single dimensions relevant to fundamental frequencies, specifically as for notional orthogonal beams parallel to sides of a rectangular panel member, is indicated at 21A; and is, of course, inherently of a nature that is positionally one-dimensional though capable of limited two-dimensional application as to frequency. More complete two-dimensional treatment is indicated at 21B, essentially using inherently two-dimensional equations of vibration in plates.

The next indicated stage 22 represents investigation of modal distribution and mechanical impedance, on the one hand relative to assumed equal or unit excitement of each mode (22A), i.e. without application of any differential weighting; and on the other hand taking account of mean values (22B), preferably with further selective adjustment for end-most modal frequencies involved. A further stage of inter-active assessment of estimated mechanical impedance is indicated at 23, specifically as to aspect ratios relative to specific drive-coupling transducer positions (23A) and as to specific transducer positions

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relative to aspect ratios (23B).

More specifically, spread of frequencies of natural resonant modes for an acoustic panel member is readily investigated by use of central differencing analysis, viz:-

$$SEE(A) := \sqrt{\frac{1}{last(A) - 3} \sum_{n=1}^{last(A) - 1} (A_{n+1} + A_{n-1} - 2 \cdot A_n)^2}$$

where A_n are resonant mode frequencies (eigenvalues) in ascending order.

Appropriate refinement regarding investigating spread of resonant mode frequencies can include considering useful sub-groupings according to some characteristic, say of a nature involving symmetry. For example, for substantially rectangular acoustic panel members, and at least relative to orthogonal beam simplification, the SEE measure could be in relation to odd-odd, even-even, odd-even and even-odd sub-groups of resonant modes individually for such sub-groups and collectively by weighted summing, viz:-

$$F_{moo}(\alpha) := \begin{array}{|l} \text{for } p_o \in 1, 3 \dots P - 1 \\ \text{for } q_o \in 1, 3 \dots Q - 1 \\ A_{N_o \cdot \frac{p_o - 1}{2} + \frac{q_o - 1}{2}} \leftarrow f_m(\alpha, p_o, q_o) \\ \text{sort}(A) \end{array}$$

$$F_{mee}(\alpha) := \begin{array}{|l} \text{for } p_e \in 0, 2 \dots P - 1 \\ \text{for } q_e \in 0, 2 \dots Q - 1 \\ A_{N_o \cdot \frac{p_e}{2} + \frac{q_e}{2}} \leftarrow f_m(\alpha, p_e, q_e) \\ \text{sort}(A) \end{array}$$

$$F_{mix1}(\alpha) := \begin{array}{|l} \text{for } p_o \in 1, 3 \dots P - 1 \\ \text{for } q_e \in 0, 2 \dots Q - 1 \\ A_{N_o \cdot \frac{p_o - 1}{2} + \frac{q_e}{2}} \leftarrow f_m(\alpha, p_o, q_e) \\ \text{sort}(A) \end{array}$$

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$$\text{Fmix2}(\alpha) := \begin{array}{|l} \text{for } p \in 0, 2 \dots P - 1 \\ \quad \text{for } q \in 1, 3 \dots Q - 1 \\ \quad \quad B_{\frac{p}{2} + \frac{q-1}{2}} \leftarrow f_m(\alpha, p, q) \\ \text{sort}(B) \end{array}$$

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$$\text{SEW}(\alpha, a, b, c, d) := \frac{a \cdot \text{SEE}(\text{Fmoo}(\alpha)) + b \cdot \text{SEE}(\text{Fmee}(\alpha)) \dots + c \cdot \text{SEE}(\text{Fmix1}(\alpha)) + d \cdot \text{SEE}(\text{Fmix2}(\alpha))}{a + b + c + d}$$

The values of frequencies of natural resonant modes and their distribution or spread depend on materials/structure and geometry/configuration of panel members concerned; and indicate suitability for acoustic device application, for which evenness of spread/distribution is established as being particularly beneficial. There is, of course, no account taken of transducer location at this stage.

For known resonant mode frequencies and corresponding shape of bending wave vibration can also be modelled, the mechanical admittance can be investigated for any particular transducer location (p,q), viz:-

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$$Z_m = \frac{\mu \cdot m^2}{j \cdot \omega \cdot \left[\sum_p \sum_q \left(\frac{Z(p, \xi_0)}{C_p} \right)^2 \cdot \left(\frac{Z(q, \eta_0)}{C_q} \right)^2 \cdot \frac{1}{\omega_{pq}^2 - \omega^2} \right]}$$

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$$Y_m(\omega) = j \cdot \omega \cdot \left[\sum_p \sum_q \frac{Y_{p,q}}{(\omega_{p,q})^2 - \omega^2 + 2j \cdot \xi \cdot \omega_{p,q} \cdot \omega} \right]$$

where $Y_{p,q}$ is the square of the amplitude of the mode shape at the transducer location concerned, and ξ represents an amount of damping. Plotting a log-log graph can facilitate finding smoothest response, or the root mean square deviation can be investigated over a specified range, say for minima of

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$$\sigma(\alpha, \xi, \eta) := \sqrt{\frac{1}{sx} \left[\sum_i x_i \cdot (\log(|Ym(x_i, \alpha, \xi, \eta)|))^2 \right]}$$

or of

$$\sigma(\alpha, \xi, \eta) := \sqrt{\frac{1}{sx} \left[\sum_i x_i \cdot (W_i \cdot \log(|Ym(x_i, \alpha, \xi, \eta)|))^2 \right]}$$

representing application of a weighting function.

Where the resonant mode frequencies are known, but not the corresponding vibration shapes (or same not modelled and taken into account by choice), investigation of intrinsic mechanical impedance can be investigated using the formulae

$$Ym'(\omega) = j \cdot \omega \cdot \left[\sum_p \sum_q \frac{1}{\left[(\omega m_{p,q})^2 - \omega^2 \right] + 2j \cdot \zeta \cdot \omega m_{p,q} \cdot \omega} \right]$$

$$\sigma'(\alpha, \xi, \eta) := \sqrt{\frac{1}{sx} \left[\sum_i x_i \cdot (W_i \cdot \log(|Ym'(x_i, \alpha)|))^2 \right]}$$

which can be done without reference to any particular transducer location by setting $Y_{p,q}$ to unity. Results will not be as accurate as for mechanical admittance taking account of transducer location, and will be slower than above investigation of mechanical admittance.

Figure 3 is a graphical representation of variation of of mechanical impedance with frequency choosing rectangular panel aspect ratios expected to be above (1.527), below (0.838) and between (1.141) optimum for useful acoustic action substantially isometric panels. Figure 3B shows real and imaginary components of the mechanical impedance

for the intermediate aspect ratio (1.141). Generally smooth nature at higher frequencies is apparent, and importance of resonance modes at lower frequencies is implicit, as already well established from the above published PCT application, particularly distribution as evenly as practical.

Figure 4A plots a measure (SD) of standard deviation of mechanical impedance against aspect ratio for a substantially isotropic rectangular panel member with a preferred transducer location from the above published PCT application, specifically at proportionate length and width co-ordinates (0.444, 0.429), and subject to a smoothing factor of 10%. Expected optimum aspect ratio of 1.134:1 is substantially confirmed by one minimum of the plot. However, other minima appear, particularly one of promising depth and greater width, i.e. less sharply defined, specifically bottoming at about 1.47:1.

Further investigations of these aspect ratios for standard deviation of mechanical impedance against proportionate co-ordinate values for transducer locations have led to useful refinement of the latter. Thus, for the aspect ratio of 1.134:1 of the above published PCT application, plots of Figures 5A - D in turn set each of length and width proportionate transducer location co-ordinates to the established values of $3/7$ and $4/9$ and show

10% smoothed standard deviation of mechanical impedance for the other proportionate co-ordinate, i.e. of width and length, respectively. These investigations result in refinement of the 0.444 value to 0.441 and of the 0.429 value to 0.414; and results of listening tests have shown noticeably improved performance; both subjectively and objectively within constraints and limitations of such measurement exercises.

The plots of Figures 6A - D likewise investigate the unexpected aspect ratio possibility at its minimum value of about 1.47:1. The resulting values for length and width proportionate co-ordinates of transducer location are 0.453 and 0.447. Further listening tests have shown excellent promise for acoustic performance, and the lesser curvature of the minimum concerned in Figure 4A is believed to be particularly advantageous by reason including actual practical transducers inevitably having extent beyond their centring at particular prescribed positions.

The investigation represented by Figure 4A was then repeated for the transducer location co-ordinate values arising from Figures 5A - D and Figures 6A - D, and results shown in Figures 4B and 4C, respectively. Figure 4B shows that the minimum for the standard deviation of mechanical

impedances bottoming at the aspect ratio 1.134:1 is deepened and sharpened, whereas that at 1.47:1 is less deep and sharper. This, of course, correlates well with the greater changes of co-ordinate values arising from Figures 6A - D compared with Figures 5A - D. Figure 4C produces a refinement of the aspect ratio 1.47:1 to 1.41:1, including to a deeper minimum of standard deviation of mechanical impedance. The interestingly deep minimum at an aspect ratio of about 0.72:1 is, of course, close to reciprocal for 1.41:1, thus to be expected; and, for the indicated lesser minima at about 0.66:1 and 0.85:1 in Figure 4A, perhaps particularly in view of refining a little downwards in Figure 4B, there is closeness to the reciprocals for upper of the range 1.141/1.47:1 and lower of 1.134/1.138:1, respectively.

Indeed, much as these processes of refinement, including mutual refinement, can be of value in optimising for best available acoustic performance, they appear to be as valuably viewed in terms of indicating ranges of variation for viable acoustic operation. Particular merit arises in identifying areas of viable location for transducer means, perhaps especially for panel members with favourable geometry/bending resistances, and further for

optimisation of locations for two or more transducer means on the same panel member. However, at least equal merit arises in identifying best available locations for transducer means on panel members of unfavourable geometry/
5 bending stiffnesses. Much the same applies to identifying worst locations for transducer means, i.e. as to be avoided even where high acoustic performance is not deemed to be necessary. Accordingly it is found to be useful to present analytical results on a relative basis, effectively in
10 percentage terms, though any particular values could be applied, and normalisation may be seen as useful. It is the case that favourable geometry panel members show larger areas for likely viable-to-good/best locations for transducer means, and unfavourable geometry panel members
15 show smaller such areas; and that edge locations are confirmed as viable, though perhaps normally best used in pairs to ensure similar excitation of resonant modes that useful beam-based simplifications indicate as related to different geometrical axes.

20 Moreover, due account should be taken of available power output, whether as to low being acceptable for evenness of excitation of more resonant modes, or high being preferred even at cost of fewer modes excited and/or less

evenly excited. However, higher numbers and more evenness are usually associated with smoothness of power, and are most readily compensated towards flatness by suitable electronic input signal conditioning, at least where power efficiency is not necessarily of paramount importance.

Figures 7A, B indicate arriving at the aspect ratios 1.38 and 1.41, together with transducer location co-ordinates (0.44, 0.414) and (0.455, 0.452), respectively, see Figures 8A, B, by a route as above for Figures 3A, B etc, but starting from aspect ratios 1.149, 1.134 and 1.762. Interestingly, however, further indication arises other favourable aspect ratios at about 1.6 and 1.2, with transducer location co-ordinates (0.41, 0.44) and (0.403, 0.406), respectively, see Figures 8C, D. The mechanical impedance plots of Figures 9A -D are generally useful regarding the transducer location co-ordinates, as is evident by inspection for all of above aspect ratios, i.e. 1.138, 1.41, 1.6 (taken as refined to 1.62 or during refinement to 1.6) and 1.2 (taken as refined to 1.266 or during refinement).

Generality of such usefulness is manifest in self-evident identification of areas including precisely calculated locations. At least where such areas are larger

than transducer dimensions, good excitation coupling is to be expected along with tolerance of actual location without losing viability. Figures 10A, B are quarter panel contour plots of mechanical impedance deviation for the aspect ratios 1.41 and 1.47, respectively, and establish credence for such range affording good transducer locations, see substantial extents of areas of least/smoothest mechanical impedance location (cross hatched), albeit within which further precise calculation is available as desired/useful.

Indeed, this technique lends itself readily to extension for investigation of best available transducer locations even for panels other than identified as favourable. Identified such locations may well have more viable mechanical impedance than for better aspect ratio panels, but can be viable at least for somewhat lesser frequency ranges of operation.

It is also feasible to investigate virtually any boundary conditions for acoustic panels, ranging from substantially free or only lightly damped as specifically described in the above published PCT application to much more constrained, even clamped. Indeed, preferential co-ordinate positions have even been identified for a circular panel at (0.8, 0.6).

Investigation of aspect ratios for fully clamped panels, as highly suitable for practical loudspeaker equipment with preference for rigid or semi-rigid edge-mounting, has revealed precisely calculated favourable aspect ratios 1.160, 1.341 and 1.643 together with likewise
5 precisely calculated preferential transducer location coordinates (0.437, 0.414), (0.385, 0.387) and (0.409, 0.439), respectively. Figures 11A, B with Figure 14A, Figures 12A, B with Figure 14B and Figures 13A, B with
10 Figure 14C demonstrate application of analytical methodology as above for Figures 3A, B etc in confirmation of values just listed - see also the quarter-panel mechanical impedance plot for the aspect ratio 1.16 and substantial extent of areas promising for transducer location, even two
15 such separate areas, (cross hatched).

Indeed, much as for the aspect ratio 1.138 for free or near-free panel edge conditions, the actually quite close aspect ratio 1.160 for clamped edge panels appears to have significant extent(s) of at least viable transducer
20 locations - and is itself postulated as having substantial tolerance, at least with likely increasing particularity of transducer locations. Figure 16 gives revealing comparison of above preferential clamped edge aspect ratios and

transducer locations, including further for above aspect ratio 1.138.

Particular exemplification is now given of specific mathematics and calculation/computation supporting above

5 given results in terms row by row of

- eigenvalues corresponding to investigated resonant modes, and smoothing factor
 - useful angle definitions
 - specific panel parameters and related expressions
 - 10 - displacement functions for different (free/clamped) boundary conditions
 - length/width fractions for proportionate transducer location co-ordinates along with formula involving mechanical impedance
 - 15 - three mechanical impedance formulae
 - two ratios of infinite and finite panel impedances involving aspect ratios and transducer locations
- all intendedly without prejudice to generality implicit in approach hereof.

EXAMPLE I

calculate eigenvalues $p := 0..14$ $q := 0..14$ $\zeta := 10\%$

$$\lambda_p := (2 \cdot p + 1) \cdot \frac{\pi}{2} \quad \lambda_0 := 0 \quad q := 0 \quad \lambda_p := \text{root} \left(\tan \left(\frac{\lambda_p + q}{2} \right)^2 - \tanh \left(\frac{\lambda_p + q}{2} \right)^2, 0 \right) + \lambda_p$$

$$\text{Area} := 0.141093 \cdot \text{m}^2 \quad B := 8.82 \text{ N} \cdot \text{m} \quad \mu := 0.694 \text{ kg} \cdot \text{m}^{-2} \quad \omega_0 := \frac{1}{\text{Area}} \cdot \sqrt{\frac{B}{\mu}}$$

$$\omega_{p,q} := \sqrt{\frac{B}{\mu} \cdot \left[\left(\frac{\lambda_p}{L_x} \right)^2 + \left(\frac{\lambda_q}{L_y} \right)^2 \right]} \quad C_p := \frac{\cos(\lambda_p) - \cosh(\lambda_p)}{\sin(\lambda_p) + \sinh(\lambda_p)} \quad C_0 = 1$$

$$Z_{\text{free}}(p, \zeta) := \text{if}[p > 0, (\cos(\lambda_p \cdot \zeta) + \cosh(\lambda_p \cdot \zeta)) \cdot C_p + (\sin(\lambda_p \cdot \zeta) + \sinh(\lambda_p \cdot \zeta)), 1]$$

$$Z_{\text{fix}}(p, \zeta) := \text{if}[p > 0, (\cos(\lambda_p \cdot \zeta) - \cosh(\lambda_p \cdot \zeta)) \cdot C_p + (\sin(\lambda_p \cdot \zeta) - \sinh(\lambda_p \cdot \zeta)), 0]$$

$$\xi_0 := 0.441 \quad \eta_0 := 0.414 \quad Y_{p,q} := \left(\frac{Z_{\text{free}}(p, \xi_0)}{C_p} \right)^2 \cdot \left(\frac{Z_{\text{free}}(q, \eta_0)}{C_q} \right)^2$$

$$Z_m(\omega) := \frac{\mu \cdot L_x \cdot L_y}{1 + \omega \cdot \left[\sum_p \sum_q \frac{Y_{p,q}}{(\omega_{p,q})^2 - \omega^2} + 2j \cdot \zeta \cdot \omega_{p,q} \cdot \omega \right]}$$

$$Z_m(\omega, \alpha) := \frac{\mu \cdot \text{Area}}{1 + \omega \cdot \left[\sum_p \sum_q \frac{Y_{p,q}}{\left[\frac{B}{\mu} \left[\frac{(\lambda_p)^2}{\alpha \cdot \text{Area}} + \frac{(\lambda_q)^2}{\text{Area}} \right] - \omega^2} + 2j \cdot \zeta \cdot \omega^2 \right]} \right]} \quad \text{approx}$$

$$Z_m(\omega, \alpha) := \frac{\delta \cdot \sqrt{B \cdot \mu}}{1 + \frac{\omega}{\omega_0} \cdot \left[\sum_p \sum_q \frac{\delta \cdot Y_{p,q}}{\left[\frac{(\lambda_p)^2}{\alpha} + \alpha \cdot (\lambda_q)^2 \right] - (1 - 2j \cdot \zeta) \cdot \left(\frac{\omega}{\omega_0} \right)^2} \right]}$$

define

$$Y_m(x, \alpha) := \frac{\delta \cdot \sqrt{B \cdot \mu}}{Z_m(x \cdot \omega_0, \alpha)}$$

$$Y_m(x, \alpha) := 1 + x \cdot \left[\sum_p \sum_q \frac{\delta \cdot Y_{p,q}}{\left[\frac{(\lambda_p)^2}{\alpha} + \alpha \cdot (\lambda_q)^2 \right] - (1 - 2j \cdot \zeta) \cdot x^2} \right]$$

Turning to alternative analysis and design methodology specifically using inherently fully two-dimensional plate vibration equations, there is self-evident possibility of taking account of more up to all possible modes of bending wave related vibration in panels. This, of course raises the matter of assessing which up to given set of circumstances.

However, first application of such methodology gives rise to substantially free-edge rectangular panel aspect ratios precisely calculated at 1.134, 1.227, 1.320 and 1.442 together with likewise calculated "best" transducer location co-ordinates (0.359, 0.459), (0.414, 0.424), (0.381, 0.429) and (0.409, 0.459), respectively. For substantially rectangular clamped edge panels, precisely calculated aspect ratios (1.155, 1.299, 1.309, 1.5, 1.602 arise together with transducer location co-ordinates (0.446, 0.407), (0.391, 0.374), (0.281, 0.439), (0.347, 0.388) and (0.399, 0.488), respectively.

Both of closeness and differences as compared with above orthogonal two-beam simplified methodology are of interest and subject of further investigation.

Reverting to analysis of panels of any aspect ratios, fully two-dimensional analysis and methodology has been

applied over a wide range, specifically from 1.05 to 2.00 in steps of 0.05. Results are shown as quarter-panel plots of mechanical impedance in Figures 17A-T, in each case by proportionate contouring with worst and best indicated by hatching and cross-hatching, respectively, and with lightest coalesced from original 14-level scaling. Whilst this means that each plot is individual, it is found to be useful to know the darkest and near darkest locations in areal terms at about 7% intervals, though other presentation and analysis will be useful, whether as to levels and intervals as such or even as to relationships with minimum areas reasonably required for transducer coupling or with absolute levels related to transducer performance, etc.

A larger scale areal plot on a six-level grey scale contour basis is given in Figure 18 for one of the original preferential aspect ratios, specifically 1.134, and the distribution of worst locations (lightest) is interestingly mostly in accord with previous thinking, namely close to, but not actually at, each corner. However, possibility of true or near-true point energisation could well be attractive if precisely on a corner itself, perhaps even on a localised extension for practical sizes of transducer,

and if smoothness of power transfer out-weighed inevitable reduction of efficiency of power transfer. Extension of the worst locations in lobes away from the corner at quite acute angles to the sides is seen as noteworthy.

5 Concentration of lowest mechanical impedance (darkest) at long-known well in-board but eccentric locations is also of interest, including separation into discrete sub-areas, though perhaps particularly extent of next-darkest region to splitting intrusion from a virtually diagonal lobe of
10 more variable mechanical impedance from the worst near-corner location. Edge-adjacent location of strips of low to lowest mechanical impedance deviation is in accordance with what we had found empirically, namely including favouring positions correlating well with co-ordinates of in-board
15 sub-areas of least mechanical impedance deviation and longest known preferential location 25 for transducers.

Figure 19 is essentially another representation of what is shown in Figure 18, but usefully in effectively continuous three dimensional format in accordance with
20 mechanical impedance.

Example is now given of two-dimensional analysis and of methodology along the lines of the previous example for two-beam simplified techniques.

EXAMPLE II**Panel data:**

- E_x, ν_x Young's modulus and Poisson's ratio of panel material along the x-axis
 E_y, ν_y Young's modulus and Poisson's ratio of panel material along the y-axis
 5 G_{xy} In-plane shear modulus of panel material
 ρ Density of panel material
 L_x, L_y Panel length along the x- and y-axis directions respectively
 h Panel thickness

Constants :

$$\begin{aligned}
 10 \quad D_x &= \frac{E_x \cdot h^3}{12 \cdot (1 - \nu_x \cdot \nu_y)} & D_y &= \frac{E_y \cdot h^3}{12 \cdot (1 - \nu_x \cdot \nu_y)} & D_k &= \frac{G_{xy} \cdot h^3}{12} & D_{xy} &= D_x \cdot \nu_y + 2 \cdot D_k \\
 \mu &= \rho \cdot h & \alpha &= \frac{1}{L_x \cdot L_y} \cdot \sqrt{\frac{D_{xy}}{\mu}} & r &= \frac{L_x}{L_y}
 \end{aligned}$$

Modal frequency expression :

$$15 \quad f = \frac{\alpha}{2 \cdot \pi} \cdot \sqrt{D_x \cdot \left(\frac{\lambda_x}{\sqrt{r}} \right)^4 + D_y \cdot \left(\sqrt{r} \cdot \lambda_y \right)^4 + 2 \cdot D_{xy} \cdot \lambda_x \cdot \lambda_y \cdot (\lambda_x - \beta_x) \cdot (\lambda_y - \beta_y) \dots} \\
 \sqrt{+ 4 \cdot D_k \cdot \lambda_x \cdot \lambda_y \cdot [(\lambda_x - \beta_x) \cdot (\lambda_y - \beta_y) - (\lambda_x - \gamma_x) \cdot (\lambda_y - \gamma_y)]}$$

where λ_x, λ_y are the relevant (boundary-condition dependent) beam eigenvalues in the x- and y-directions respectively and $\beta_x, \beta_y, \gamma_x, \gamma_y$ are corresponding constants.

As an example, for a fully free panel :

$$20 \quad \beta_x = \beta_y = -6 \quad \gamma_x = \gamma_y = 2 \quad \lambda_x = \lambda_y = \lambda \quad \text{where} \quad \cosh(\lambda) \cdot \cos(\lambda) = 1$$

Mode shape expression :

$$\phi = c1 + c2 \cdot \zeta + c3 \cdot \cosh(\lambda \cdot \zeta) + c4 \cdot \sinh(\lambda \cdot \zeta) + c5 \cdot \cos(\lambda \cdot \zeta) + c6 \cdot \sin(\lambda \cdot \zeta)$$

where $c1 \dots c6$ are boundary-condition and mode-dependent beam function constants

As an example, for the 1st flexural mode of a fully free beam :

$$c1 = c2 = 0 \quad c3 = c5 = 1.0 \quad c4 = c6 = 0.982502215$$

Relative mobility expression :

The mobility of the finite panel relative to that of an infinite panel ($\delta \sqrt{D_{xy} \mu}$) at a specific point on the panel is given by

$$\psi = 1 - \frac{4 \cdot \alpha}{\pi} \cdot \sum_j \frac{\Phi_j}{\left[(f_j)^2 - F^2 \right] + 1 \cdot \left[\delta_s (f_j)^2 + 2 \cdot \delta_v (F \cdot f_j) \right]}$$

where F is the driving frequency and δ_s, δ_v are the structural and viscous damping factors for the panel material respectively and $\Phi = (\phi_x)^2 + (\phi_y)^2$

Being a function of driving frequency, the relative mobility for any point is sampled at 'j' discrete frequencies in the frequency range of interest, the mean of which is given by

$$\psi_{av} = \frac{1}{2 \cdot (F_{max} - F_{min})} \cdot \sum_j \Delta F_j \cdot (\psi_j + \psi_{j+1}) \quad \text{where} \quad \Delta F_j = F_{j+1} - F_j$$

Measure of goodness :

A logarithmic measure of the variation of relative mobility (with the mean removed) is used for optimisation purposes, i.e. :

$$\kappa = \log \left(\frac{\psi}{\psi_{av}} \right)$$

The standard deviation of this measure is used for identifying optimum drive locations

$$\sigma = \sqrt{\frac{1}{2(F_{max} - F_{min})} \sum_j \Delta F_j [(x_j)^2 + (x_{j+1})^2]}$$

Precision of values given above for aspect ratios and/or co-ordinate transducer locations is an inevitable result of calculation, and not necessarily indication of more than some point within a range of viability For

transducer locations areal plots are particularly promising, certainly affording deserving basis for investigation by experimentation both as to matching between results of analytical methodology as proposed
5 herein and as to actual acoustic performance for which number of resonant modes coupled is important as is reasonable evenness of couplings to as many modes as practical. Ready availability of analysis for any aspect ratios and refinement thereof relative to particular
10 transducer locations and own refinement capability can be useful in revealing greater generality of application of some especially favourable transducer locations/areas as well as particularity to aspect ratios of other transducer locations/areas.

15 It is believed to be of particularly high potential to have arrived at a single discipline or demonstrator of merit, termed herein measure of smoothness of mechanical impedance, that is equally capable of locating and specifying both valuable aspect ratios and transducer
20 locations, including evident capability for recursive refinement, i.e. essentially jointly choosing geometry and transducer location by similar procedures using essentially the same variable or parameter, or feasible variations

thereon.

CLAIMS

1. A method of making an acoustic device relying on bending wave action in an area of a panel member, the
5 method characterised by comprising

analytically assessing a power transfer related parameter of the panel member, as a function of shape parameters of said panel member, said shape parameters being selected from the configuration/geometry of said
10 area of the panel member, the bending stiffness of said area of the panel member, and the location of bending wave transducers in said area of said panel member; and

selecting said shape parameters in accordance with said analytical assessment of the power transfer related
15 parameter of said panel member, the acoustic device concerned and desiderata therefor correlating with achieving said acoustic device performance.

2. A method of making an acoustic device relying on bending wave action in an area of a panel member, the
20 method characterised by comprising

analytically assessing a power transfer related parameter of the panel member, as a function of shape parameters of said panel member, said shape parameters being selected from the configuration/geometry of said
25 area of the panel member, the bending stiffness of said area of the panel member, and the location of bending wave transducers in said area of said panel member,

identifying, as shape parameters are varied, the

minimum or minima of deviation of the said power transfer related parameter from a value or relation therefor; and

selecting said shape parameters in accordance with said identified minimum or minima of the power transfer
5 related parameter of said panel member, the acoustic device concerned and desiderata therefor correlating with achieving said acoustic device performance.

3. Method according to claim 1 or 2, including compensating for any deviation from flatness of output
10 power by correlated conditioning of the input to the acoustic device.

4. A method according to claim 1, 2 or 3 in which the power transfer related parameter is mechanical impedance.

5. Method according to claim 4, wherein the analytical
15 assessment includes determining the standard deviation of the mechanical impedance from some value or relation therefor.

6. Method according to claim 5, wherein the deviation is relative to same or unity weighting applied for resonant
20 frequency modes involved.

7. Method according to claim 5 or 6, wherein the deviation is relative to mean values for resonant frequency modes involved.

8. Method according to any of claims 5 to 7, wherein the
25 deviation is relative to selective weighting for resonant frequency modes involved.

9. Method according to claim 8, wherein the selective weighting is applied for end-most resonant frequency

mode(s) involved.

10. Method according to claim 9, wherein the end-most resonant frequency modes involved are lowest in frequency range of interest for desired or acceptable operation of 5 said acoustic device.

11. Method according to any preceding claim, wherein the resonant frequency modes involved in said analytical assessment arise from making simplification analogy of one-dimensional nature.

10 12. Method according to any preceding claim, wherein the panel member is or is considered as substantially rectangular, and resonant frequency modes involved in the analytical assessment arise from making simplifying analogy to orthogonal beams notionally in directions 15 parallel to pairs of opposite sides of said panel members.

13. Method according to any preceding claim, wherein resonant frequency modes involved in said analytical assessment arise relative to two-dimensional relationships of the panel member as such.

20 14. Method according to any preceding claim, wherein said selecting is as to proportions of shape of said panel member.

15. Method according to claim 14, wherein said selecting is assisted by presenting smoothed results of said 25 analytical assessment as mechanical impedance of said panel member against varying said shape proportions to show minima of deviation.

16. Method according to claim 14 or claim 15, wherein said

analytical assessment is for given transducer location(s).

17. Method according to any preceding claim, wherein said selecting is as to location of transducer means for a given said panel member.

5 18. Method according to claim 17, wherein said analytical assessment is for variable one relative to fixed other of co-operative areal locators such as co-ordinates of transducer location and results are presented functionally, usefully graphically, in looking for minimum
10 deviation of preferably smoothed mechanical impedance.

19. Method according to claim 18, wherein alternate said assessments involve alternation as to which areal locator is fixed and which variable.

20. Method according to any preceding claim, wherein said
15 analytical assessment includes distribution of mechanical impedance of said panel member on areal basis.

21. Method according to claim 20, wherein said selecting is assisted by presenting results of said analytical assessment as contour mapping of areal deviation/variation
20 of mechanical impedance.

22. Method according to claim 21, wherein said analytical assessment and contour mapping is of one only of two or more notional parts of the same or similar geometry that together substantially match area of said panel member.

25 23. Method according to claim 22, wherein said analytical assessment and contour mapping is of one quadrant for a substantially rectangular shape of said panel member.

24. Method according to any preceding claim including the

step of selecting panel member shape proportion and the step of selecting transducer location wherein one of the two said steps of selecting is done at least once after and using results of doing the other said step.

5 25. Acoustic device comprising a panel member having its geometry/configuration and/or location(s) for bending wave transducer means in accordance with application of method according to any preceding claim.

26. Acoustic device comprising a panel member relying on
10 bending wave action for its acoustic performance, the panel member being of substantially rectangular shape and isotropic as to its bending stiffnesses in directions parallel to its substantially orthogonal axes, the panel member having axial proportions of 1 : 1.41.

15 27. Acoustic device according to claim 26, comprising a transducer having co-ordinates, measured from one corner along the axes, of substantially 0.453 and/or substantially 0.447.times the length of the panel member measured along the corresponding axes.

20 28. Acoustic device comprising a panel member relying on bending wave action for its acoustic performance, the panel member being of substantially rectangular shape and anisotropic as to its bending stiffnesses in directions parallel to its substantially orthogonal axes, the panel
25 member having axial proportions affording axial bending stiffnesses in a ratio of 1 : 1.41.