Introduction to distributed mode loudspeakers (DML) with first-order behavioural modelling

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Abstract: A simple equivalent circuit of a distributed mode loudspeaker (DML) is described, which is accessible to engineers not specialised in acoustics. The DML is an acoustic radiator, the electrical, mechanical and acoustical properties of which differ radically from conventional moving coil transducers. DML radiation results from uniformly distributed, free vibration in a stiff, light panel and not pistonic motion. To enable acoustic engineers to use existing software programs to model their application of DML technology, an efficient equivalent circuit is developed. Within the constraints of the model, velocities and displacements of the various elements can be calculated, and the radiated acoustic power and pressure predicted.

List of symbols

Panel parameters:

E = Young's modulus, Pa
ρ = mass density, kg/m^3
ν = Poisson's ratio
η_{mech} = mechanical loss factor
η_{rad} = radiation loss factor each side of panel
γ = power ratio (radiation loss/total loss)
h = thickness of panel, m
B = bending rigidity of panel, Nm
μ = mass per unit area of panel, kg/m^2
$v(\omega)$ = velocity of travelling wave in panel (tangential to panel)
$k_p(\omega)$ = wave number of travelling wave in panel (tangential to panel)
Z_p = mechanical impedance of panel, kg/s

Note that, for an isotropic panel $B = h^3/12\ E/(1 - \nu^2)$ and $\mu = h\rho$

Acoustic parameters:

P = acoustic power, W
U = velocity, m/s
R_r = acoustic radiation resistance, kg/s
X_r = acoustic radiation reactance, kg/s
Z_r = acoustic radiation impedance, kg/s
ρ_0 = density of air, kg/m^3
c = speed of sound in air, m/s
k = acoustic wave number = ω/c, m^{-1}
a = radius of circular piston, m

Mechanical parameters (normal to panel):

x, x_p, x_m = displacements, m
u, u_p, u_m = velocities, m/s
F = force, N
Y_p = specific velocity (mobility) = u_p/F, m/Ns

General:

ω = angular frequency, rad/s
bar over variable = RMS value (e.g. $\bar{u}$)
single underline = vector (e.g. $\underline{x}$)
double underline = matrix (e.g. $\underline{\underline{M}}$)

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1 Introduction

The aim of this paper is to introduce a simple yet efficient equivalent circuit of a distributed mode loudspeaker (DML), which is accessible to engineers not specialised in acoustics. (DML is a term coined by New Transducers Limited (NXT) to describe a loudspeaker working according to their teaching. NXT is a registered trademark of New Transducers Ltd, a subsidiary of NXT plc.) Using this equivalent circuit, the velocities and displacements of the various elements can be modelled and the radiated acoustic power predicted.

In the 1920s the ideal loudspeaker was conceived to operate as a rigid piston with all points on the radiating surface moving in phase. This design aim applied irrespective of whether the loudspeaker diaphragm was driven from a moving armature, or later, from a moving coil. Operating a diaphragm in this manner imposes two fundamental requirements to maintain an even frequency response. First, the diaphragm has to be sufficiently small, compared with the wavelength of sound in air, to approximate to a point source and secondly, the whole diaphragm must move with the same acceleration so that, because of pistonic behaviour, it can be considered as a lumped moving mass. There have been a few notable exceptions to this design philosophy. For example, Bertagni (Sound Advance Systems Inc., California, USA) envisaged a 'timpanic' diaphragm with controlled break-up and Manger (Manger-Schallwandler, Mellrichstadt, Germany) exploited damped bending waves.

Contrary to normal pistonic diaphragm behaviour, a DML is an acoustic radiator, the electrical, mechanical and acoustical properties of which differ radically from conventional moving-coil transducers such that a new loudspeaker class is defined. A DML is distinguished by acoustic radiation that emanates from uniformly distributed, free bending wave vibration induced in a stiff, light panel and not from pistonic motion. Optimisation techniques have been developed and are the subject of many international patents by New Transducers Ltd (e.g. PCT/GB96/02145, September 1996). These solve complicated differential equations defining motion in such structures and enable stiff, light panels to be designed that exhibit an extremely uniform modal density, the prerequisite for distributed mode behaviour. The two fundamental requirements for a pistonic radiator do not apply to a DML, the latter being both arbitrarily scalable in size and the antithesis of a rigid body.

To understand and describe DML operation, a simple mechanical model is first derived, from which an equivalent circuit is developed. Using this equivalent circuit, the velocities and displacements of the various elements can be determined. Also, within the constraints imposed by the model parameters, the radiated acoustic power can be estimated from the panel mean velocity.

Currently, there is no technical terminology that adequately describes the acoustic radiation of a DML. In some respects a DML is a better approximation to a point source than a piston, but it also exhibits both temporal and spatial decorrelation, which are unusual yet distinctive DML traits. An open-baffled DML could be described as a 'diffuse dipole', although it is neither completely diffuse nor completely dipolar. Likewise, a DML within an infinite baffle could be described as a 'diffuse monopole'. A brief description of the DML is presented as background, although it is not proposed to discuss in depth the acoustics of this class of radiator. More complete descriptions of the acoustical properties of the DML are to be found elsewhere, for example [1–4] for physical acoustics and [5, 6] for psychoacoustics.

2 Piston loudspeakers

Initially, we review briefly the core theory that describes the conditions for a pistonic acoustic radiator to yield a flat power response, from which the corresponding results for a DML can be extrapolated.

For a pistonic loudspeaker to achieve frequency independent power transfer, the diaphragm velocity must be inversely proportional to frequency, or in other words, the acceleration must be constant. This is achieved when the loudspeaker operates as a mass-controlled device, such that a constant force produces a constant acceleration, a condition normally achieved above the fundamental resonant frequency. This requirement for a constant acceleration is a direct result of the frequency dependent real part of the radiation impedance of the piston [7]. The complex radiation impedance Z_r of a vibrating piston is given as

$$Z_r = \pi a^2 \rho_0 c \{R_1(2ka) + jX_1(2ka)\} \tag{1}$$

where

$$R_1(x) = 1 - 2J_1(x)/x \tag{2}$$

$$X_1(x) = (4/\pi)[x/3 - \text{higher terms}] \tag{3}$$

a = radius of piston (m) ρ_0 = density of air (kg/m^3)
c = velocity of sound in air (m/s) k = wave number

Defining U as the diaphragm velocity, the excitation force F and forward acoustic radiation power P are related by

$$F = Z_r U \quad \text{and} \quad P = \frac{1}{2} U^2 R_r \tag{4}$$

Consequently, observing the expression for power, a flat power response is achieved where the diaphragm velocity is inversely proportional to frequency and where the real part of the radiation resistance is proportional to frequency squared, a condition met when $ka < 1$ and when the loudspeaker operates above its fundamental resonance. For $ka > 1$, the radiation impedance becomes approximately constant and equal to $\pi a^2 \rho_0 c$, as can be seen from eqn. 2. Since the impedance characteristic changes from a 12dB/octave slope to a constant slope around $ka = 1$, while the mass controlled operation is retained assuming an ideal piston, a second-order high-frequency roll-off in the power response occurs.

A small acoustic radiator approximates a point source, which produces spherically symmetric acoustic radiation. At higher frequencies, when the source size becomes large with respect to the wavelength of sound in air, the radiation beams in the forward direction. This beaming maintains an on-axis pressure proportional to the acceleration and accounts for the loss of acoustic power. Consequently, the pressure response of a typical cone loudspeaker is limited at high frequency as illustrated in Fig. 1.

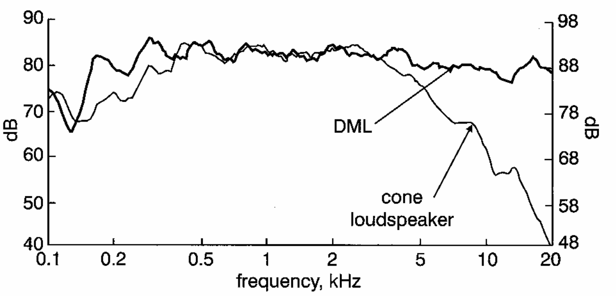

Fig. 1 *Measured on-axis pressure responses of DML and 140mm cone in baffle of identical size*
Levels adjusted for equality in pass-band of piston

3 Distributed mode loudspeakers

The radiation impedance of a typical pistonic loudspeaker is normally small compared with the mass of the cone and therefore has little effect on its motion. However, it dominates the loudspeaker's performance at the mechanical-to-acoustical interface, where the frequency dependence of the radiation resistance defines the required mechanical performance (i.e. a frequency independent acceleration).

The radiation impedance is also small compared with the impedance of a typical DML. However, in contrast to the pistonic loudspeaker, its effect on the acoustical performance is also small. The mechanical losses in a DML constructed from stiff but light panel material are typically very low, where the principal damping mechanism is attributable to acoustic radiation. The power ratio γ defined as

$$\gamma = \frac{2\eta_{rad}}{\eta_{mech} + 2\eta_{rad}}$$

can be as high as 98%, and is often as high as 90%. Therefore, a good approximation when modelling the mechanical behaviour of the DML is to assume that all the mechanical input power is radiated. This approximation holds for frequencies at which the loudspeaker is truly modal and where the panel is sufficiently large to be self-baffling. If the panel is mounted in a baffle, then this additional constraint applies to the total baffle size. At lower frequencies, a simple diffraction model can provide a fairly accurate extension to the high-frequency case.

Acoustic radiation from each element of a DML results from surface motion normal to the plane of the panel that is induced by bending waves that propagate across the surface. Because bending waves are dispersive (i.e. the wave velocity is a function of frequency, as shown in eqn. 5) [8], a good approximation is to consider the panel as a randomly vibrating area. The radiation intensity from such an area is shown in [9] to depend on the square of the mean velocity, hence the requirement is for constant velocity excitation. To achieve constant velocity with constant force, the mechanical impedance must be resistive. An infinite panel operating in a bending wave mode meets this criterion [10], where expressions for bending wave velocity $v(\omega)$, wave number $kp(\omega)$ and mechanical impedance Z_p are quoted below. Note that $v(\omega)$ is the in-plane velocity, and should not be confused with panel velocity u_p, normal to the panel surface.

$$v(\omega) = \omega^{0.5}\left(\frac{B}{\mu}\right)^{0.25} \tag{5}$$

$$k_p(\omega) = \omega^{0.5}\left(\frac{\mu}{B}\right)^{0.25} \tag{6}$$

$$Z_p = 8\sqrt{B\mu} \tag{7}$$

4 Mechanical model

To model a physical system, assumptions are required. Because the DML is considered to be in a state of random vibration, any existing panel motion is uncorrelated to any new applied input, and therefore appears as an infinite plate (see [11] for a definitive justification of a statistical approach to mechanical impedance). Additionally, because the panel has low mechanical loss, all the energy supplied to the panel is assumed dissipated by acoustic radiation. These assumptions have been shown to give useful results and measurements confirm that to calculate the radiated acoustic power, only the mechanical power delivered to the panel need be calculated [12]. If the impedance analogue is used, where voltage is analogous to force and velocity is analogous to current, the radiated pressure is proportional to the mean velocity in the panel.

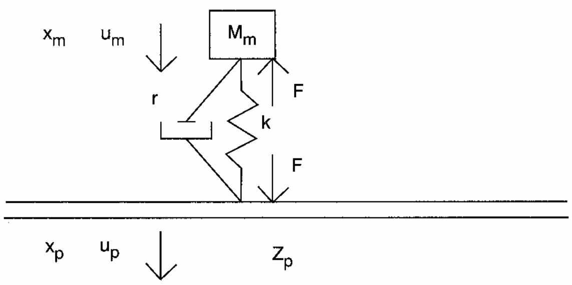

Fig. 2 *Mechanical elements and forces for panel driven by damped mass-spring oscillator*

Given that the DML is a resistance-controlled device and that the acoustic radiation need not be considered in detail, the equivalent circuit can be developed from Fig. 2. This structure represents a simplified version of the 'inertial magnet driver' application [13]. The coupled equations of motion are given in eqns. 8 and 9:

$$M_m\frac{d^2x_m}{dt^2} + r\left(\frac{dx_m}{dt} - \frac{dx_p}{dt}\right) + k\left(x_m - x_p\right) - F = 0 \tag{8}$$

$$Z_p\frac{dx_m}{dt} + r\left(\frac{dx_p}{dt} - \frac{dx_m}{dt}\right) + k(x_p - x_m) + F = 0 \tag{9}$$

If the driving force is assumed to be sinusoidal with angular frequency, ω, and using the same symbols to refer to the peak values of variables:

$$F(t) \equiv Fe^{j\omega t} \quad \text{(and similarly for } x_m \text{ and } x_p\text{)}$$

$$\omega^2 M_m x_m - j\omega r(x_m - x_p) - k(x_m - x_p) - F = 0 \tag{10}$$

$$j\omega Z_m x_p - j\omega r(x_m - x_p) - k(x_m - x_p) - F = 0 \tag{11}$$

or in matrix form, separating the stiffness, mass and resistance matrices:

$$\left(\underline{\underline{K}} - \omega^2\underline{\underline{M}} + j\omega\underline{\underline{R}}\right)\underline{x} - \underline{F} = 0$$

or

$$\underline{x} = \left(\underline{\underline{K}} - \omega^2\underline{\underline{M}} + j\omega\underline{\underline{R}}\right)^{-1}\underline{F} \tag{12}$$

where

$$\underline{\underline{M}} = \begin{pmatrix} M_m & 0 \\ 0 & 0 \end{pmatrix} \qquad \underline{\underline{K}} = \begin{pmatrix} k & -k \\ -k & k \end{pmatrix}$$

$$\underline{\underline{R}} = \begin{pmatrix} r & -r \\ -r & r \end{pmatrix} + \begin{pmatrix} 0 & 0 \\ 0 & Z_p \end{pmatrix}$$

$$\underline{x} = \begin{pmatrix} x_m \\ x_p \end{pmatrix} \qquad \underline{F} = F\begin{pmatrix} -1 \\ 1 \end{pmatrix}$$

So the specific velocity, or mobility Y_p in the panel is given by:

$$Y_p = \frac{u_p}{F} = \frac{j\omega x_p}{F} \tag{13}$$

$$Y_p = j\omega(0 \quad 1) \times \begin{pmatrix} k - \omega^2 M_m + j\omega r & -k - j\omega r \\ -k - j\omega r & k + j\omega r + j\omega Z_p \end{pmatrix}^{-1}\begin{pmatrix} -1 \\ 1 \end{pmatrix} \tag{14}$$

$$Y_p = \frac{\omega^2 M_m}{(\omega^2 M_m(Z_p + r) - Z_p k) - j\omega(kM_m + rZ_p)} \tag{15}$$

By inspection, and noting that the velocity in the spring and damper is the difference between the velocities in the mass and panel, the equivalent circuit using the impedance analogue can be drawn as in Fig. 3. It is then a relatively straightforward task to verify that the ratio of panel velocity u_p to force F matches that given by the reciprocal of eqn. 15, i.e.

$$Zm_{eff} = Z_p\left(1 - \frac{k}{\omega^2 M_m}\right) + r + \frac{1}{j\omega}\left(k + \frac{rZ_p}{M_m}\right) \tag{16}$$

5 Practical implementation of the equivalent circuit for a moving-coil motor

Figs. 2 and 3 represent a DML panel driven by an idealised point source. If the motor system is considered to be a moving coil, M_m represents the mass of the magnet, cup and pole piece. The spring and damper assembly represent a means of attachment of the motor to the panel. To account for the effect of the coil, a mechanical mass M_c is added in series with Z_p that is equal to the mass of the

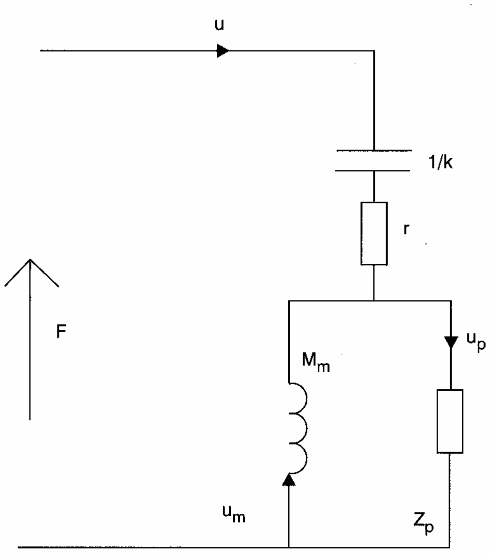

Fig. 3 *Impedance analogue model of DML panel*
$Zm_{eff} = F/u_p$
$u = u_p - u_m$

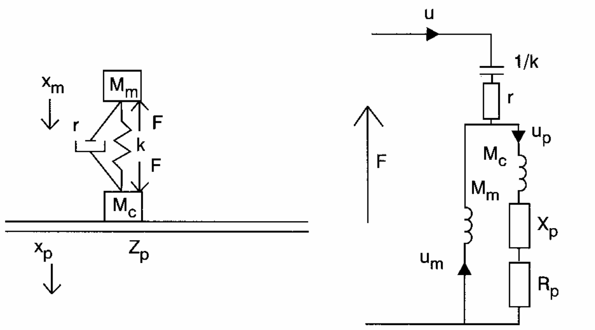

Fig. 4 *Impedance analogue model of DML panel with moving-coil motor system*
$Z_p = R_p + jX_p$

voice coil. Additionally, the impedance Z_p is only real for a point source and generally becomes complex for a finite diameter voice coil. The reactive component X_p is small except at high frequencies, where it should be recalled that pseudorandom fluctuations in X_p due to modal behaviour are not considered in this simple model. The high-frequency component of X_p is systematic and therefore of importance. Fig. 4 shows a model of such a system together with its equivalent circuit.

The effective mechanical impedance relating u_p to F for the model in Fig. 4 is

$$Zm_{eff} = Z_p'\left(1 - \frac{k}{\omega^2 M_m}\right) + r + \frac{1}{j\omega}\left(k + \frac{rZ_p'}{M_m}\right) \tag{17}$$

where

$$Z_p' = R_p + jX_p + j\omega M_c$$

However, at high frequencies eqn. 17 can be simplified, where if all terms involving negative powers of ω are ignored, $Zm_{eff} = Z_p' + r$, i.e.

$$Zm_{eff} \approx (R_p + r) + j(X_p + \omega M_c) \approx R_p + j\omega M_c \tag{18}$$

which gives the high-frequency limit f_{max} for the DML as

$$f_{max} \approx \frac{R_p}{2\pi M_c} \tag{19}$$

A similar simplification yields the low-frequency limit f_{min}, where, ignoring k,

$$\frac{1}{Zm_{eff}} \approx \frac{1}{R_p} + \frac{1}{j\omega M_m} \quad \text{so} \quad f_{min} \approx \frac{R_p}{2\pi M_m} \tag{20}$$

Alternatively, if the influence of M_m is small and stiffness dominates:

$$f_{min} \approx \frac{k}{2\pi R_p} \tag{21}$$

6 Modelling results

To complete the model, a gyrator and coil impedance must be included as in Fig. 5. To evaluate the model variables in the analysis, the complete electromechanical circuit is coded into a commercially available electroacoustic simulator (e.g. AkAbak™, formerly Panzer & Partner, now supported by New Transducers Limited, Huntingdon, UK). The electrical and mechanical domains are constructed from the mechanical equivalent circuit shown earlier, with the addition of the transfer characteristics of the moving-coil exciter. These parameters include the magnet moving mass, compliance, shove factor (usually written as '*BL*', where *B* is the magnet strength, and *L* is the conductor length. The authors have not used this in order to avoid confusion with the panel parameter, B) and voice-coil DC resistance, also the component values R_s and C_s are r and $1/k$, respectively.

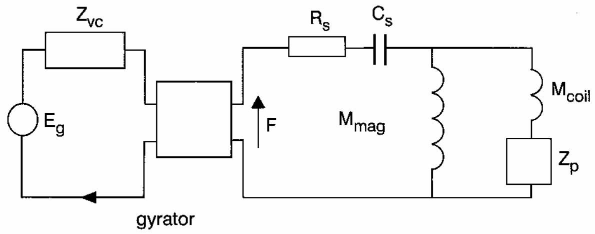

Fig. 5 *Complete electromechanical schematic for DML panel and exciter*

Solving the above circuit enables the mean driving-point velocity, and hence the sound pressure to be evaluated. The following results were obtained from such a model. A panel of relatively low mechanical impedance (~10kg/s) was used.

6.1 Terminal impedance

The reactive nature of the traditional moving mass loudspeaker is reflected in the terminal impedance, giving a classical low-frequency electrical resonance. Since the DML panel is approximately resistive, its terminal impedance is substantially flat. The two classes of loudspeaker are compared in Fig. 6. Notice the evidence of modal activity apparent in the measurement of the DML, which indicates the degree of approximation made in the model. Practical experience shows that these modes are less evident on higher impedance panels, or at higher frequencies. Interestingly, the cone loudspeaker example also shows evidence of modal activity at about 500Hz, which is not untypical, so is not entirely pistonic in its behaviour.

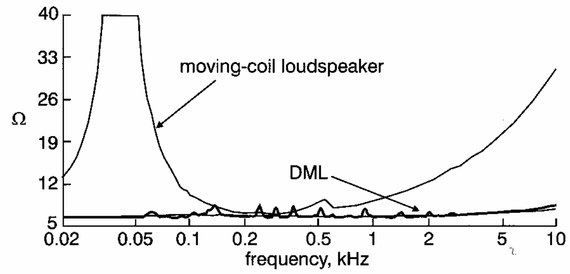

Fig. 6 *Terminal impedance of DML panel with exciter from model compared with measurement and cone drive unit of same nominal impedance*

6.2 Velocities

Fig. 7 shows plots of panel velocity overlaid with measured pressure, taken at 1m. The panel is truly modal from about

140Hz (the lowest mode is at about 45Hz), and self-baffling from about 250Hz. The high-frequency extension of this particular panel is actually better than predicted by the model and results from additional compliance between the voice coil and the panel. The model can readily be extended to include this effect. Ripples in the pressure response and the progressive attenuation below 250Hz are primarily due to diffraction. Some improvement in low-frequency accuracy can be obtained by modelling the lowest resonance of the panel, but the resulting increase in model complexity makes this unattractive.

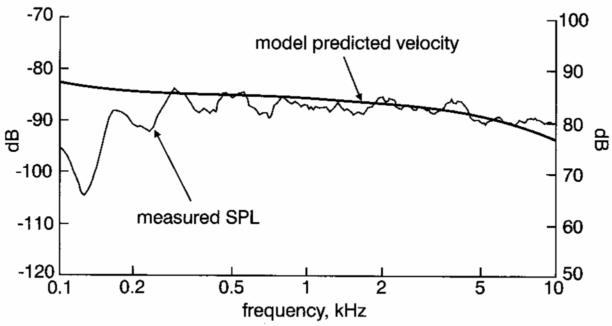

Fig.7 *Velocity of DML panel and exciter from model, with measured SPL at 1m*

7 Conclusions

An electromechanical model has been presented which enables engineers to use existing software programs to investigate the application of DML technology to their acoustic problems. Although more advanced models have been developed [3, 14], this simple model offers an efficient alternative while retaining good accuracy. Given that a stiff, light panel can be designed to have an optimal modal distribution, together with low loss, it has been shown that the simplified model can accurately predict acoustic pressure and acoustic power, where it is necessary to calculate only the mean velocity in the panel.

The bandwidth of the DML is seen from eqns. 18–20 to depend only on the ratio of magnet mass, coil mass, and suspension stiffness to the panel mechanical impedance. The panel properties affect the sensitivity and frequency limits only via the mechanical impedance. It is possible to design a single DML to be substantially flat in pressure and power response over a wide audio bandwidth (exceeding 100Hz to 20kHz) without any electrical filters, a task impossible to achieve with conventional loudspeaker technology.

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