New developments in loudspeaker system design – the B&W 801 monitor

Presented by G. J. Adams
at the Consumer Electronics Show
Chicago, June 1979

Cover: Model 801 on test under free-field conditions
in Europe's largest anechoic chamber at
the Building Research Establishment, Watford.

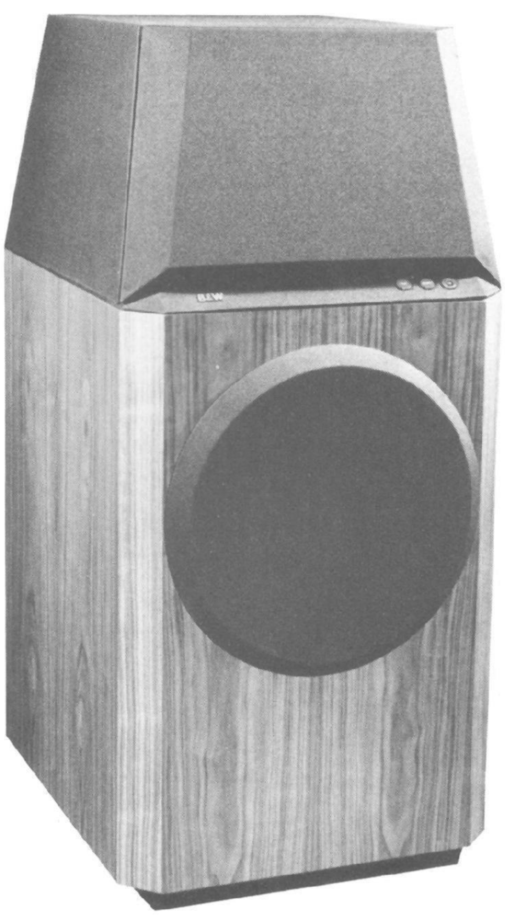
Foreword

by **John Bowers**
Managing Director
of B&W Loudspeakers

At the commencement of any new B&W loudspeaker design project, a considerable time is normally spent preparing the design brief against the often differing requirements of acoustics and engineering on the one hand, and the costs of production and marketing on the other. Physical size and production costs are two obvious areas where compromises are usually necessary. However, with the design of Model 801 our approach was different. My proposition was that the Design Team should be given complete freedom to design the loudspeaker without recourse to compromise so that a shared ambition to lay down a very demanding specification, and to meet or exceed this in the final design, could be fulfilled.

The detailed interpretation of a simple statement is often complicated, and this certainly proved to be the case with the 801. Three design criteria emerged which at first sight seemed contradictory. Firstly, the loudspeaker should be as neutral and as accurate as possible, with minimum colouration and low distortion. Secondly, the 801 must be capable of safely delivering the very high acoustical output powers required for professional monitoring and large environment situations. Thirdly, whilst the shape and configuration of the system must completely satisfy the acoustical requirements, the final design must be accepted as being a good piece of furniture.

The following paper discusses the avenues of research and development which have led to these conflicting requirements being resolved, and we offer the final product confident that we have met our original design aims.



Introduction

The design programme for any commercial product must by necessity be limited due to considerations of time and cost. These constraints unfortunately result in some elements of compromise being made in the final product. While the B&W development team naturally try to keep compromise to a minimum in each new model of the DM range, we felt the need to establish a new range to run in parallel with the existing DM range. Described as the Series 80, this is to be a no-compromise range of loudspeakers designed for the professional or discerning home user. Each new Series 80 model will reflect the latest construction and design techniques developed by the B&W research team. Whilst the research investment required for these developments, coupled with the desire to produce a professional-grade product, must inevitably result in a relatively expensive range of loudspeakers, some of these new developments will obviously be passed down to future models of the lower cost DM Series.

Although designed principally for studio monitoring applications, the 801 has been styled and finished so as to be equally at home in domestic surroundings. Studio monitor loudspeakers are widely used at the sound mixing stage in the production of disc recordings. The final tonal balance obtained on these recordings is therefore largely determined by the accuracy of the loudspeaker systems used for monitoring. Thus the first requirement of a studio monitor loudspeaker is that it should have a flat amplitude/frequency response over the frequency range of normal music programme material. A second requirement is one of acoustic power output capability. In many monitoring situations, the loudspeaker system will be required to reproduce sound-pressure levels which approach or equal those of the live performance. The maximum acoustic output power required from a studio monitor is thus considerably larger than that required from a normal domestic-type loudspeaker system.

The above performance requirements formed the basis of the design concepts for Model 801. In satisfying these requirements and the others laid down in Section 1, several new design and construction techniques were developed.

Section 1 **Design Concepts for Model 801**

1.1 Performance Requirements

The main performance requirements laid down for Model 801 were as follows:

(i) Low-frequency cut-off (-3 dB) $= 40$ Hz.

(ii) Amplitude/frequency response $= \pm 2$ dB, 50 Hz to 20 kHz.

(iii) Acoustic power output capability in passband $= 1$ W.

Requirement (iii) is equivalent to a sound-pressure level of 112 dB re. 20 μPa at 1 metre in a 2π sr field.

In addition to these, the amplitude/frequency response was required to remain linear for listening positions up to $\pm 30°$ off-axis in a horizontal plane and $\pm 5°$ off-axis in a vertical plane. This ensures that positioning of the loudspeakers and/or the listener is not critical when reproducing stereo programme material.

1.2 Visual Design

Because of the requirement for a high acoustic power output capability, studio monitor loudspeakers are generally larger than loudspeaker systems designed for domestic use. While no specification of the enclosure volume was laid down in Section 1.1, if the 801 was to be suitable for use in domestic surroundings then its dimensions would have to be limited. An enclosure of 100 litres in volume was thought to be the largest that could be accommodated in an average domestic listening room. Limitation of the enclosure dimensions is in itself not sufficient to ensure acceptance by the domestic user; the visual appearance and styling of the system must be such that it generates excitement on initial viewing, reflects the technical innovation of the product, and yet still fulfills the function of a piece of domestic furniture. To help the B&W development team achieve this goal we were fortunate to have the services of Kenneth Grange and his design consultancy, Pentagram.

Section 2 **Driver Design**

The 801 uses three moving-coil direct-radiator loudspeaker drivers. These are mounted in closed-box type enclosures and positioned so that their effective acoustic centres are approximately equidistant from the listener. Although vented-box loading offers higher efficiency for the same enclosure volume and low-frequency cut-off, this loading method was rejected in favour of the closed-box system because the latter has a smaller effective reverberation time (ref. 1).

A brief description of the design and construction of each of the drivers is given below.

2.1 Low-frequency Driver

The displacement-limited acoustic output power rating of the low-frequency system is proportional to the square of the peak displacement volume V_D of the driver diaphragm. V_D is given by (ref. 2)

$$V_D = S_D \, x_{max} \, , \qquad (1)$$

where S_D is the effective surface area of the driver diaphragm and x_{max} is its peak linear displacement. The value of x_{max} required to meet the specified acoustic power output capability is minimised by making S_D as large as possible. In view of the 100 litre volume limit specified for the 801 a 300 mm diameter driver was chosen. Taking the effective diaphragm diameter of this driver to be 270 mm, the value of x_{max} computed to satisfy the power output capability of 1 W was found to be 5.4 mm (ref.3). This value was calculated taking into account the peak voltage spectrum of a piece of recorded organ music, (ref.3, fig.9), and it should therefore represent the maximum displacement required for most types of music programme material. To ensure that distortion is not introduced for displacements up to 6 mm peak, the BW300 low-frequency driver designed for the 801 uses long-throw suspensions and a voice-coil overhang of 6 mm on both sides of the magnet pole piece. For $x_{max} = 6$ mm the displacement-limited electrical input power rating of the system for the organ music piece is 230 W. When the system is used with an amplifier of this rating the displacements of the driver diaphragm caused by fault conditions in the amplifier may be considerably larger than 6 mm. To prevent the voice coil being damaged under these conditions, a non-standard magnet assembly has been used so that the voice-coil displacement is limited by the restraining action of the inner suspension before the voice coil can come into contact with the magnet back plate. Fig.1 illustrates the magnet assembly used in the low-frequency driver.

The diaphragm assembly used in the BW300 driver is of conventional construction employing a Bextrene cone coated with a PVA damping compound. The long-throw outer suspension is formed from plasticised PVC. Fig.2 shows the diaphragm assembly of a

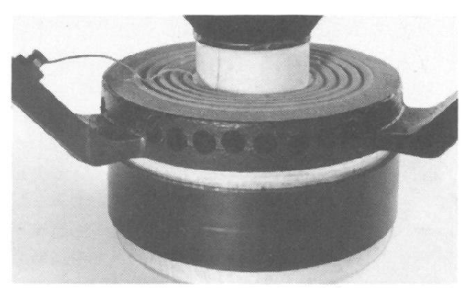


Fig. 1 **Rear view of BW300 low-frequency driver showing non-standard magnet assembly.**

Fig. 2 **Front view of BW300 low-frequency driver mounted in enclosure.**

A prototype 801 low-frequency driver under construction.

completed low-frequency driver. The BW300 voice coil is of 50 mm diameter and is wound on a Nomex former to increase its maximum working temperature.

2.2 Mid-frequency Driver

While attention must still be paid to the displacement requirements of the mid-frequency driver, these are much less of a problem at mid and high frequencies. The choice of the diameter of the mid-frequency driver is largely determined by the frequency response and directional characteristics required at high frequencies, and the thermal power rating required of the voice coil. In a 3-driver system, the mid-frequency driver is normally used for reproducing frequencies up to 4 or 5 kHz at most.

An approximately uniform frequency response up to these frequencies can be comfortably achieved using an effective diaphragm diameter of 100 mm. Although a smaller diameter driver would give an improved high-frequency performance, it would be difficult to obtain the low-frequency cut-off and the thermal power rating required of the mid-frequency system using such a driver. A 100 mm diameter driver with a 25 mm diameter voice coil was therefore chosen for the 801.

Good reproduction of mid-range frequencies is an essential requirement of any high-quality loudspeaker system. While accurate reproduction of low and high frequencies is also important, the mid-frequency system will be required to reproduce the greater part of the music signal spectrum (see for example the music spectrum shown in ref.3). This is of course the reason why a 3-driver system is preferred to a 2-driver one; the important mid-frequency range is reproduced by only one driver.

Because of the special requirements of the mid-frequency system, particular attention has been paid to the design of the 801 mid-frequency driver. The cone of the MK100 driver is formed from resin-bonded Kevlar material using a special process for which B&W hold the Patents. The Kevlar material is a woven matrix of aromatic polyamide fibres which when resin-bonded and heat-treated produce a cone of low mass and high strength. Comparative tests between cones of Kevlar and Bextrene have shown that the transient performance of the Kevlar cone is superior to that of the Bextrene cone. Because the frequency responses of the cones were similar, this suggests that the Kevlar material 'stores less energy' than the Bextrene material. Although the mechanism of this energy storage is not yet well understood, the good performance of the Kevlar cone led to its obvious choice for use in the mid-frequency driver.

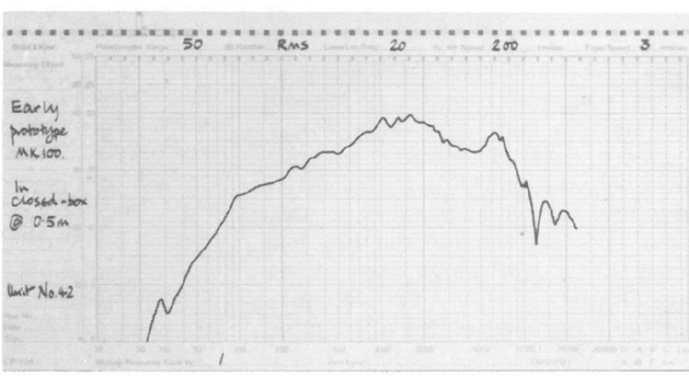
The measured frequency response of an early prototype MK100 is given in fig.3. This shows a relatively smooth response up to 6 kHz. However, the peak in the response around 6.5 kHz would make it difficult to obtain a smooth roll-off at high frequencies when the mid-frequency system is driven via its crossover. Even if the response peak was well outside the passband of the crossover network, its presence would still be undesirable if the harmonic distortion of signals reproduced in the passband was to be kept to a minimum.

From consideration of the geometrical and material parameters of the cone, it seemed likely that the response peak around 6.5 kHz was due to the presence of axi-symmetric bending waves on the cone (ref. 4). The facility to measure and identify these and other types of vibrational modes of the diaphragm assembly has recently been installed in the B&W laboratories. Measurement of the velocity on the diaphragm surface is achieved by directing a fixed-frequency laser beam (spot diameter = 0.3 mm) at the point of interest. A part of this incident beam is also converted to a reference beam by shifting its frequency by 5 MHz. The beam reflected from the diaphragm surface is collected by an optical lens system and combined with the reference beam. The combined beams are then directed into a photo-multiplier to obtain an electrical signal. This signal contains a 5 MHz difference frequency which is frequency modulated due to the motion of the reflecting surface. Conversion of this signal to a voltage proportional to the difference frequency enables the diaphragm velocity at the point of interest to be displayed on an oscilloscope. Because the output signal is dependent on the velocity of the vibrating surface, the instrumentation is relatively insensitive to the low-frequency structural vibrations present in a normal laboratory-type environment.

The velocity versus frequency at each point on the diaphragm surface can be investigated by applying a variable-frequency sine-wave voltage to the driver voice coil. However, even if velocity measurements are restricted to points along only one radius of the diaphragm, the number of measurements required can be quite large. Data collected in this way is also difficult to analyse.

The versatility of the laser interferometry equipment has been extended considerably by interfacing it to a PDP 11 computer. The computer has been provided with facilities for converting analogue signals into digital form for storage and processing. By exciting the driver with a narrow rectangular voltage pulse, the velocity impulse at each point on the diaphragm surface can be stored in the computer, and subsequently converted to frequency-response data using the Fast Fourier Transform. If a train of input pulses is used, the measured velocity impulses can be averaged to give an improved signal/noise ratio. Using signal averaging reliable velocity data

Fig. 3 Amplitude/ frequency response of early prototype mid-frequency driver.



up to 20 kHz can be obtained without driving the loudspeaker into non-linear operation. Measurement of the impulse responses at points on the diaphragm surface enables determination of both the amplitude and phase of the velocity at these points. Because the sound-pressure/frequency response of the system is a function of the complex transverse velocity of all of those points on the diaphragm surface which are in motion (ref. 4), the storage of this data in the computer enables calculation of the contribution of the motion of all or part of the diaphragm surface to the sound-pressure output. Fig. 4 shows the Laser Interferometry system being used for the investigation of the prototype MK100 driver.

For the investigation of axi-symmetric vibrational modes, it is sufficient to measure the velocity at a number of points along one radius of the diaphragm. This data is interpreted most easily by displaying it as a 3-dimensional graph where the three axes are amplitude of acceleration, frequency, and position along the diaphragm radius.

Acceleration rather than velocity is displayed because a completely rigid diaphragm would have constant acceleration at all points for frequencies above the fundamental system resonance. Fig. 5 shows the data measured at 14 points along the diaphragm radius of the prototype mid-frequency driver. Points on the dust cap and on the outer suspension are included because these parts contribute considerably to the sound-pressure output. Two areas of particular interest are indicated in fig. 5. Area A shows an increase in the diaphragm motion on the outer edge of the cone and on the suspension for frequencies around 1.0 to 1.5 kHz. This is typical of the behaviour which occurs at the 'surround resonance' where the cone and outer suspension vibrate out of phase. Because of this phase difference the sound-pressure response does not always exhibit a noticeable peak in this region. However, the increased motion of the outer suspension at these frequencies may increase the distortion arising from non-linear motion, particularly at high input levels. Investigation of the cone and suspension motion using sine-wave drive showed that waveform distortion was greatest at the outer suspension end of the diaphragm for frequencies in the region of the surround resonance. This discovery led to the use of a narrower surround (compare figs. 7 & 8) in the final MK100 prototype so that distortion resulting from non-linear motion of the suspension surface is minimised.

Area B in fig. 5 shows the phenomena responsible for the peak around 6.5 kHz in the sound-pressure/frequency response shown in fig. 3. The amplitude of motion is greater in the central region of the cone than it is at the inner and outer edges. Although the frequency at

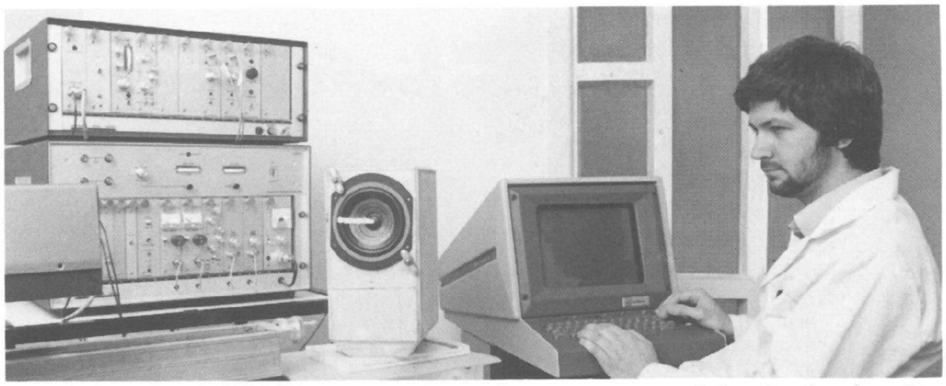


Fig. 4 **Laser interferometer equipment in use for measurement of the vibrational modes of the diaphragm of the MK100 driver.**

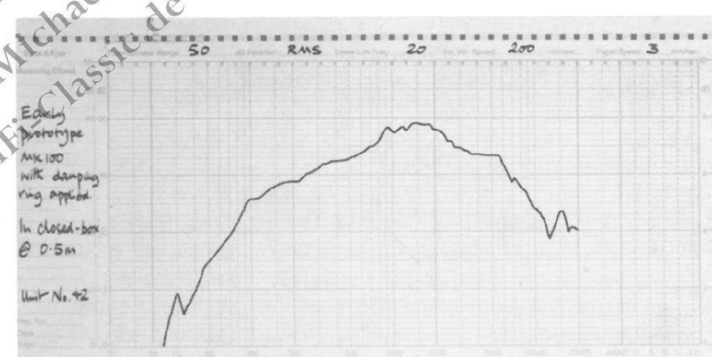


Fig. 6 **Amplitude/frequency response of early prototype mid-frequency driver measured after applying a ring of damping compound to the central area of the cone.**

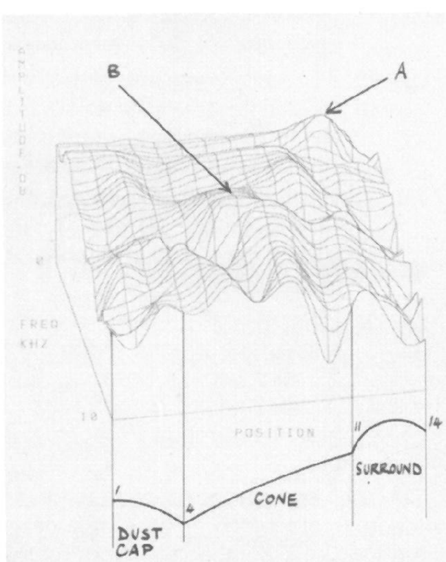


Fig. 5 **Three-dimensional graph showing the variation with frequency of the amplitude of acceleration measured along the radius of the prototype MK100 diaphragm.**

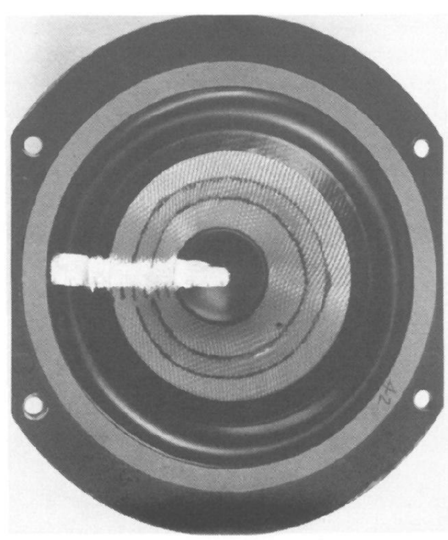


Fig. 7 **Front view of early prototype MK100 driver used for the investigation of the vibrational modes of the mid-frequency diaphragm.**

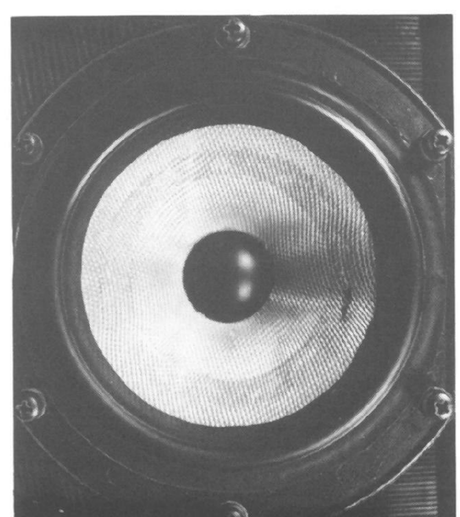


Fig. 8 **Front view of final prototype MK100 driver showing modified outer suspension.**

which this resonance occurs could be changed by modifying the material and geometrical parameters of the cone, the resonance is best reduced by increasing the internal loss factor of the cone material (ref. 4).

The application of a damping compound to the whole of the cone surface in an attempt to increase the loss factor of the Kevlar proved only partially successful; the peak in the sound-pressure response was reduced by about 1 dB. As the motion around 6.5 kHz is greatest in the central region of the cone, it was decided to try damping only this part of the cone. To our surprise, the application of a 1-cm wide ring of damping compound to the central region caused almost complete removal of the response peak. The sound-pressure/frequency response of the driver with the damping ring applied is shown in fig.6 (compare with fig.3, the sound-pressure response of the same driver without the damping ring). Damping only a selected area of the cone surface has the further advantage of minimising the increase in diaphragm mass which results from applying the damping compound.

The diaphragm of the early prototype MK100 driver showing the 14 measurement points and the applied damping ring is illustrated in fig. 7. The final prototype mid-frequency driver shown in fig.8 uses a narrower surround and thus the meridional length of the cone is greater. As one might expect, the response peak in the sound-pressure response of this unit occurred at a lower frequency than that of the early prototype. However, application of a damping ring at the antinode of the cone resonance again proved to be successful in removing the response peak.

2.3 High-frequency Driver

The TS26S high-frequency driver used in the Model 801 is a development of the tweeter designed for our DM7 loudspeaker system. Closed-box type loading is provided by the air space enclosed between the diaphragm assembly and the magnet structure. By employing a high-energy centre-pole magnet made from nickel cobalt the dimensions of the high-frequency system have been kept to a minimum so that good directional characteristics are obtained at high frequencies. The dome-shaped diaphragm which is constructed from woven polyester filaments is driven from its outer edge by a 26 mm diameter voice coil.

The thermal power rating of this driver has proved to be more than adequate for the 801. A front view of the TS26S driver is shown in fig. 9.

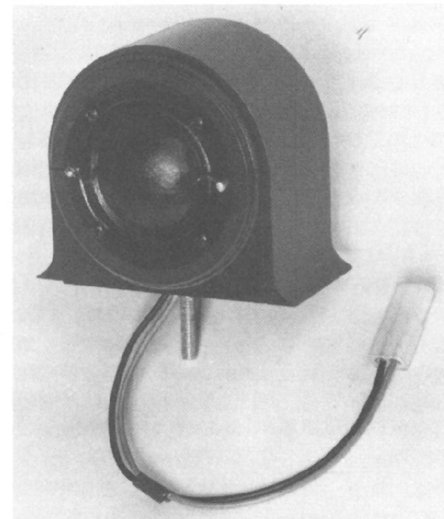


Fig. 9 **View of the TS26S high-frequency driver.**

Section 3 **Enclosure Design**

3.1 Low-frequency System

The parameters of the BW300 low-frequency driver were optimised to obtain a maximally-flat sound-pressure/frequency response and a cut-off (−3 dB) frequency of approximately 40 Hz when the driver was mounted in a 100 litre enclosure. The response of the BW300 driver mounted in a test enclosure of this volume was measured using the nearfield microphone technique (ref.5) and is shown in fig.10.

The choice of the type of materials and construction used for the low-frequency enclosure was made on the evidence of vibration measurements taken at a number of points on the enclosure surface. For least colouration of the sound output of the system, the vibration of the enclosure panels should be minimised. For enclosures constructed from particle board, the panel vibrations excited by the pressure variations existing inside the enclosure can often be reduced by bracing the structure with timber braces and/or by gluing sound-deadening panels to the internal enclosure surfaces. Past experience has shown that these measures are not always effective; thus several different types of enclosure construction were investigated during the development of the 801 low-frequency enclosure. The following constructions were tested:

- Type 1. 25 mm panels with timber cross braces.
- Type 2. 31 mm panels made from a sandwich of 19 mm board and 12 mm sound-deadening pads. Cross braces as type 1.
- Type 3. 25 mm panels made from a sandwich of 19 mm board and 6 mm sound-deadening pads. Cross braces as type 1.
- Type 4. 25 mm panels with picture-frame type braces made from 25 mm particle board.
- Type 5. 25 mm panels with braces as type 4 but made from 25 mm ply.

All the particle board panels were veneered on both sides. The panel vibrations at a number of points on the external surfaces of each construction for a sine-wave voltage applied to the low-frequency driver were measured using an accelerometer. The largest

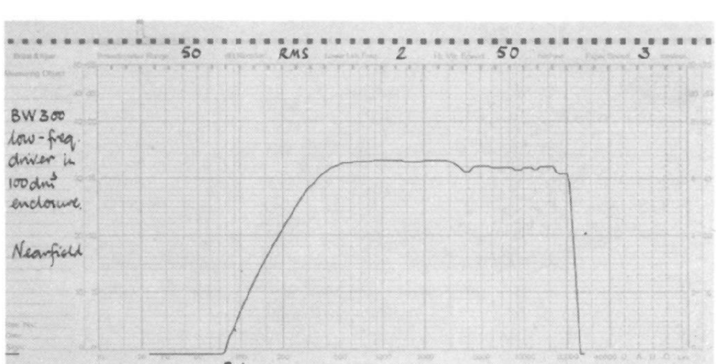


Fig. 10 **Amplitude/frequency response of low-frequency system measured using the nearfield microphone technique.**

amplitudes of panel acceleration occurred in the region of 150 to 500 Hz for all five types of construction. Fig.11 shows the maximum amplitude of acceleration measured in this frequency range for each of the enclosure types. The construction which showed the lowest panel vibration was type 5, and this was therefore chosen for the 801. An interior view of the enclosure illustrating the picture-frame braces is given in fig.12.

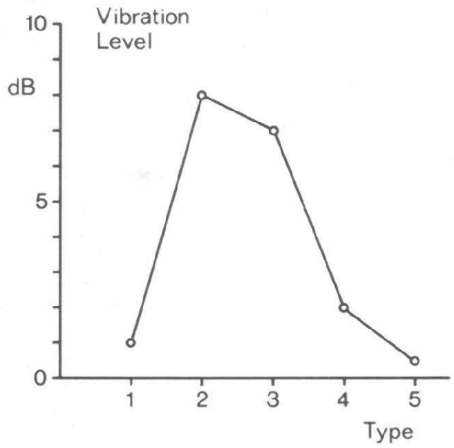


Fig. 11 **Maximum amplitudes of acceleration measured on the surface of the low-frequency enclosure for several different types of enclosure construction.**

During the investigation of the panel vibration at different points on the enclosure surface we found that the maximum amplitude of acceleration was in general greater at points near to the low-frequency driver. This suggested that a significant part of the measured panel vibration was due to transmission of driver vibration through the structure. To reduce any transmitted vibration the driver was mounted into the enclosure against a thick rubber gasket using screws which were isolated from the driver chassis. This modification produced a welcome reduction in the measured panel acceleration of some 12 to 14 dB. Fig. 13 illustrates the isolation mounting used on the prototype 801. In the production version this feature is an integral part of the driver chassis.

Fig. 12 **Interior view of the low-frequency enclosure construction finally adopted.**

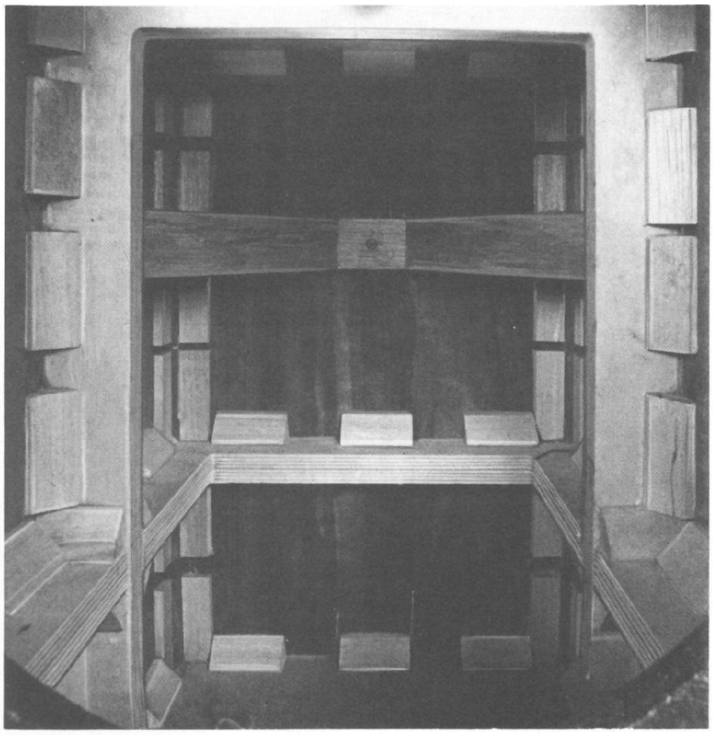
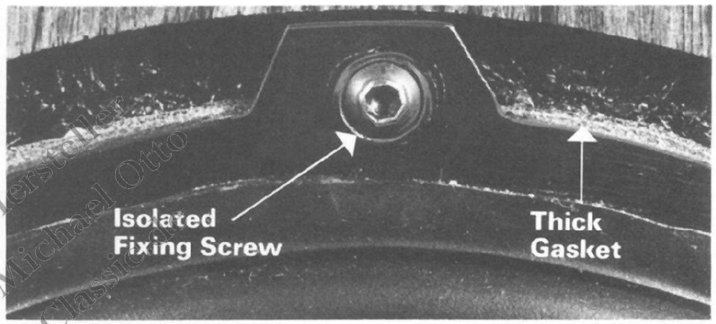


Fig. 13 **Close-up view of low-frequency driver showing the isolated mountings used to reduce transmission of driver vibration into the enclosure.**



3.2 Mid- and High-frequency Systems

The dimensions of the mid-frequency enclosure have been kept to the same order as the diameter of the mid-frequency driver so that good directional characteristics are obtained at upper mid-range frequencies. The prototype enclosure was constructed from particle board, and in this case the bonding of sound-deadening panels to the inside surfaces was found to be beneficial. Following the observations made during the design of the low-frequency enclosure, isolated driver mountings were also employed in the mid-frequency system.

The mid-frequency system is attached to the top surface of the low-frequency enclosure by means of a vertical clamping screw passing through the centre of the mid-frequency enclosure. This allows the mid- and high-frequency systems to be rotated relative to the low-frequency enclosure if desired. The base of the mid-frequency enclosure is seated on a circular rubber mounting to increase its isolation from the vibrations of the low-frequency enclosure.

The high-frequency system is positioned on the top of the mid-frequency enclosure. Measurement of the energy versus time response of this system when driven by a rectangular voltage impulse showed that a considerable part of the radiated energy was reflected off the top surface of the low-frequency enclosure. To reduce the amount of this reflected energy, the top surface of the low-frequency enclosure is covered with an acoustic absorbent foam. Fig.14 shows the prototype mid- and high-frequency systems mounted on the low-frequency enclosure.

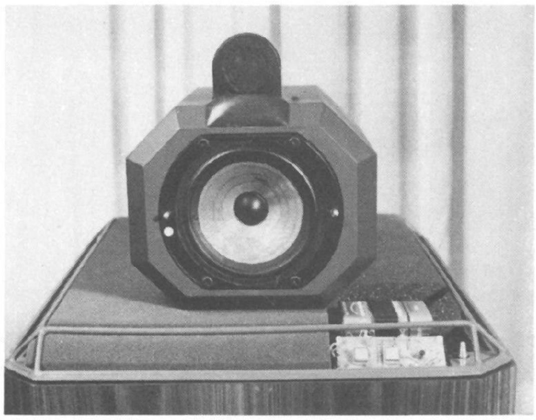


Fig. 14 **Mid- and high-frequency enclosures mounted on the top surface of the low-frequency enclosure. The acoustic absorbent foam positioned below the mid-frequency enclosure reduces the amount of high-frequency energy reflected off the top of the low-frequency enclosure.**

Section 4 **Crossover Network**

4.1 Choice of type of Crossover

After examination of the sound-pressure/frequency responses of the three component systems, the crossover frequencies were chosen to be approximately 400 Hz (low/mid) and 3.5 kHz (mid/high). While the attenuation rate of 3rd-order networks would have been sufficient for good integration of the three component systems, 4th-order networks were chosen in preference because of the more symmetrical vertical polar response obtained by using even-order networks (ref. 6).

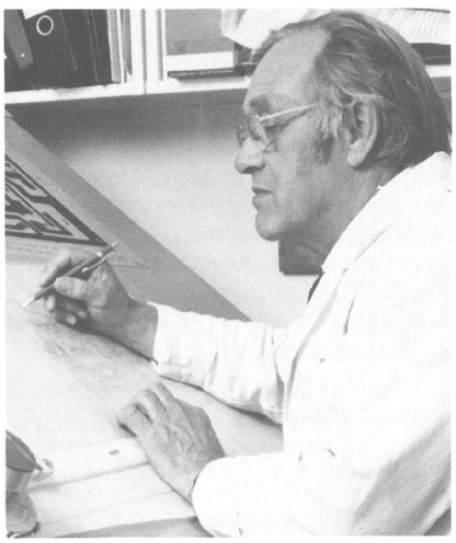
4.2 Synthesis by Optimisation

The design of a passive loudspeaker filter network is made difficult for the following reasons:

(i) The load impedance presented to the network by the driver voice coil is complex, and varies with frequency.

(ii) The sound-pressure/frequency response of the loudspeaker system is rarely uniform in the filter passband.

The complex nature of the filter load impedance means that the network cannot be designed accurately using simple analytical filter theory where the load is assumed to be a constant



Preparing artwork for the 801 crossover network.

resistance. Because of (ii) the frequency response required of the filter will not be the same as the sound-pressure/frequency response required of the loudspeaker system and crossover network combined.

As a result of these difficulties, crossover networks are normally designed using a mixture of simple analytical filter theory and trial and error. This procedure is quite adequate for designing crossover networks which employ only two or three components per section. However, for more complicated networks determination of the optimum values of the network components in this way proves to be a very arduous and time consuming job. In addition to the large number of adjustments required, it is unlikely that design by trial and error will achieve the best possible solution using the minimum number of components.

To overcome the limitations of the trial and error technique, the B&W research team have developed a computer-aided design technique for the synthesis of crossover networks.

The values of the crossover components are chosen using a numerical optimisation method so that the error between the desired response and the actual system response is minimised. This new technique is a development of a method for the computer-aided design of the basic parameters of a direct-radiator loudspeaker system given by the author in an earlier paper (ref. 7).

Consider the design of the crossover network for one component system of a multiple-driver loudspeaker system. Let the transfer function between the input voltage $V_{in}(s)$ to the crossover network and the sound-pressure output $P(s,\underline{x})$ of the loudspeaker system fed from this network be defined by:

$$H(s,\underline{x}) = \frac{1}{P_o}\left[\frac{P(s,\underline{x})}{V_{in}(s)}\right] \qquad (2)$$

where $\underline{x} = (x_1,\ldots,x_p)$ are the parameters of the crossover network, s is the Laplace transform or complex-frequency variable, and P_o is a real coefficient. If G(s) is the desired sound-pressure/frequency response of the loudspeaker system and crossover network combined, then one can form a functional given by:

$$\text{Error}(\underline{x}) = \sum_{i=1}^{r} W_i\left[\left|G(j2\pi f_i)\right| - \left|H(j2\pi f_i,\underline{x})\right|\right]^2, \qquad (3)$$

where f_i $(i=1,\ldots,r)$ are r discrete frequencies in the frequency range of interest, $W_i(f_i)$ are real weighting factors, and $j=\sqrt{-1}$. Error($\underline{x}$) is the weighted sum of squares of the difference between the amplitudes of the desired response and the actual response over a finite number of discrete frequencies.

To synthesize the best approximation to any desired amplitude/frequency response, the values of the parameters $\underline{x}$ are chosen using a numerical optimisation technique (ref. 7) such that the value of Error ($\underline{x}$) is minimised.

Because the functional is determined numerically, the weighting factors W_i can be given as non-linear functions of the parameters $\underline{x}$ so that a solution is obtained within any specified constraints on the values of these parameters.

The actual response $H(s,\underline{x})$ is computed from the measured sound-pressure/frequency response $H_L(s)$ of the component loudspeaker system and the calculated transfer function $H_C(s,\underline{x})$ of the crossover network, i.e:

$$H(s,\underline{x}) = H_L(s)\,.\,H_C(s,\underline{x})\,, \qquad (4)$$

$$\text{where } H_L(s) = \frac{1}{P_o}\left[\frac{P(s)}{V_g(s)}\right], \qquad (5)$$

$$\text{and } H_C(s,\underline{x}) = \left[\frac{V_{out}(s,\underline{x})}{V_{in}(s)}\right]. \qquad (6)$$

$V_g(s)$ is the input voltage to the loudspeaker voice coil, and $V_{out}(s,\underline{x})$ is the output voltage of the crossover network. $H_L(s)$ is determined from the loudspeaker system's impulse response which is measured and stored in the computer prior to the synthesis procedure. Calculation of $H_C(s,\underline{x})$ requires knowledge of the load impedance presented to the network. Thus the measured modulus and phase angle versus frequency of the input impedance of the loudspeaker system is also stored in the computer.

A block diagram illustrating the design of a filter network using the numerical optimisation technique is shown in fig.15.

4.3 Design Example

As an example of the optimisation technique, consider the design of the crossover network for the 801 mid-frequency system. The desired response for this system was specified by:

$$\left|G(j2\pi f_i)\right| = \frac{1}{1+\left[\dfrac{f_i}{f_u}\right]^4}\cdot\frac{1}{1+\left[\dfrac{f_l}{f_i}\right]^4}, \qquad (7)$$

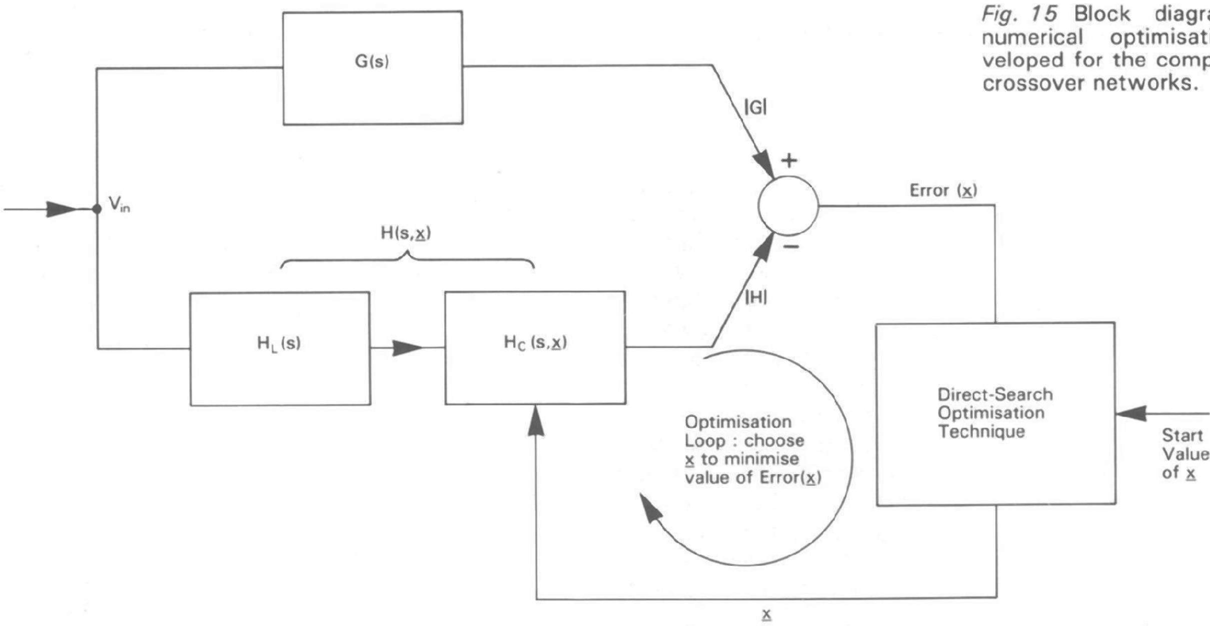


Fig. 15 Block diagram illustrating the numerical optimisation technique developed for the computer-aided design of crossover networks.

where f_l and f_u are the lower- and upper-crossover frequencies respectively. For $f_l = 400$ Hz and $f_u = 3500$ Hz the amplitude of the desired response is 6 dB down at the crossover frequencies.

The crossover network used for the optimisation was a standard 4th-order bandpass configuration consisting of eight components. A resistor was also included in series with this network to enable adjustment of the passband efficiency. The desired passband efficiency defines the value of P_0 used in equations 2 and 5. Including the attenuating resistor there were nine component values to be determined ($p = 9$).

The results of the numerical optimisation procedure obtained using fifty discrete frequencies ($r = 50$) are given in fig. 16. This shows the desired response G(s) and the actual response H(s,$\underline{x}$) calculated from equations 5 and 6 for the optimum set of component values $\underline{x}$ selected by the computer. The measured sound-pressure/frequency response of the mid-range system driven via a crossover constructed using these optimum component values is given in fig. 17. The good agreement shown between this response and the desired response is typical of the results obtained for several other design examples carried out using the optimisation technique.

The complete crossover network for

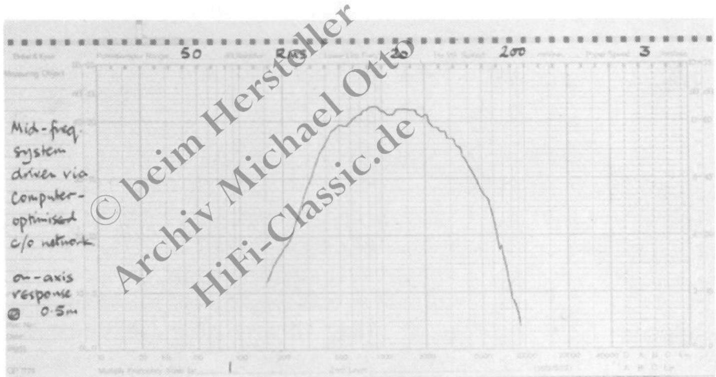
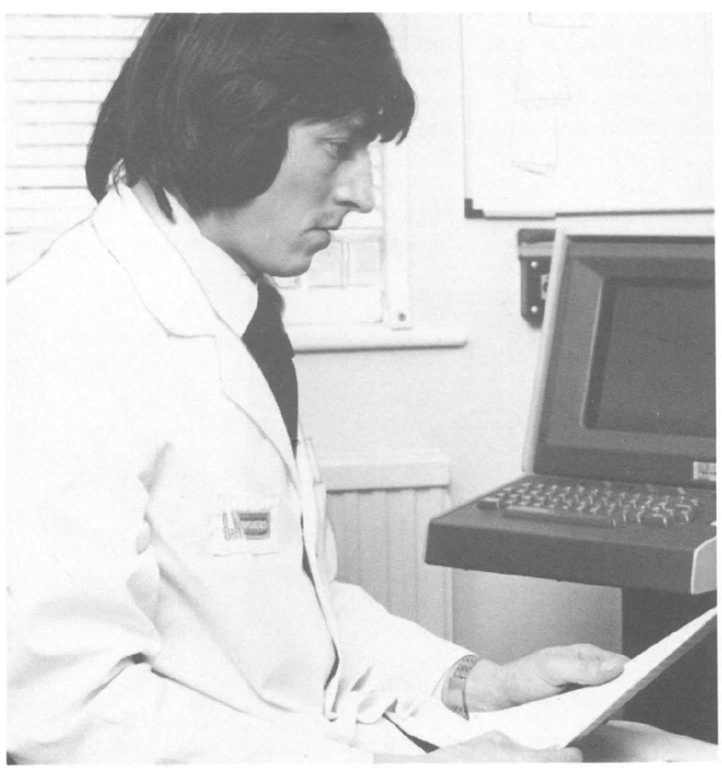


Fig. 17 Measured amplitude/frequency response of mid-frequency system driven via a crossover network constructed using computer-optimised component values.



Examining computer print-out of crossover network optimisation programme.

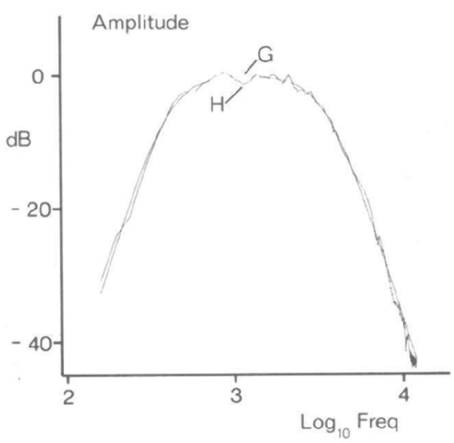


Fig. 16 Amplitude of the desired response specified for the optimisation of the mid-frequency system and crossover network. Also shown is the amplitude of the actual response calculated using the computer-optimised values of the crossover network components.

Model 801 constructed using computer-optimised component values is shown in fig. 18.

4.4 Environmental Controls

Subjective tests carried out using the prototype 801 in several different listening rooms showed that the balance between the low-, mid- and high-frequency ranges was often noticeably affected by the listening environment. To enable the correct balance to be obtained when using an amplifier which lacks comprehensive tone controls, the 801 has been provided with controls for adjusting the levels of the mid- and high-frequency ranges. The controls are built into the rear of the mid-frequency enclosure as shown in fig. 19.

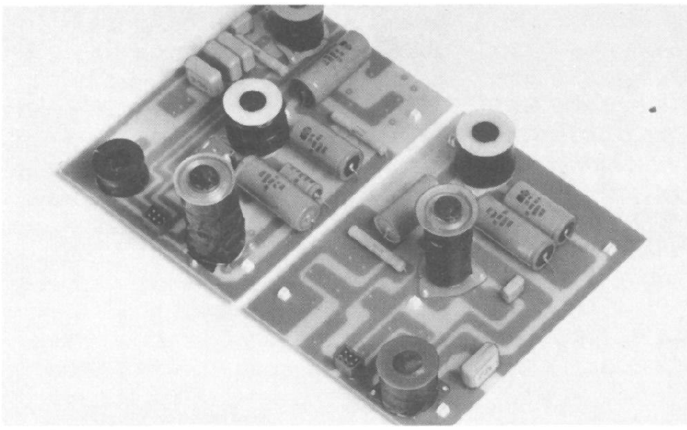


Fig. 18 The complete crossover network constructed for the prototype 801.

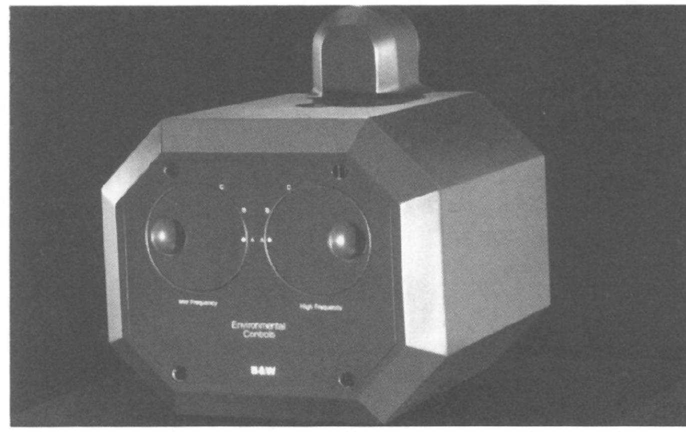


Fig. 19 Rear view of the mid-frequency enclosure showing the environmetal controls.

Section 5 **Overload Protection Device**

To obtain the maximum acoustic output power (about 1W) from Model 801 requires the system to be used with an amplifier of power rating greater than 180 W. Input power levels exceeding 230 W are likely to cause thermal damage to the loudspeaker drivers. Thus it was thought essential to provide some form of overload protection.

Inclusion of a fuse in the input to the loudspeaker system affords some degree of protection against overload. However, fuses are neither convenient or reliable. To overcome these limitations, the B&W research team have developed a unique electronic protection device (Patents applied for).

The device consists of an electronic circuit which senses the voltages applied to the voice-coil terminals of each of the three drivers. These voltages are compared to threshold voltage levels which are specified for each driver. If any of these threshold levels are exceeded for some given period, the electronic circuit operates a relay which removes the input drive to the loudspeaker system. The threshold levels are chosen so that the input drive is removed before the thermal power ratings of the drivers can be exceeded. Once the overload condition has been removed, normal operation of the system can be restored by operating a 'reset' button. A 'set' button is also provided for testing the system. When the protection device is triggered by an overload condition an indicator mounted on the top of the low-frequency enclosure is illuminated. Fig. 20 shows the control buttons and the indicator.

The protect circuitry is powered by a battery located beneath the top cover of the low-frequency enclosure (fig. 14). The current consumption of the electronic circuit when in its quiescent (reset) state is negligible, and thus the power supply can be left permanently connected.

The prototype and final versions of the protection device are illustrated in fig. 21.

Fig. 20 View of operating buttons and indicator of the overload protection system.

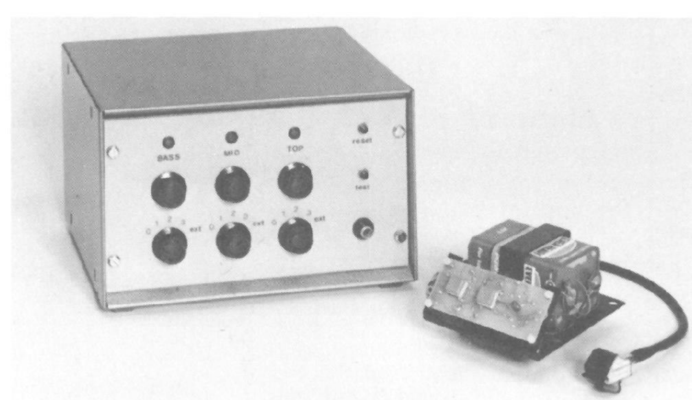


Fig. 21 Equipment constructed during the development of the protection system (left), and final protection device used in the prototype.

Section 6 **Subjective Assessment of Low-frequency Response**

The parameters of the low-frequency system were optimised to obtain a maximally-flat sound-pressure/frequency response. Although this is theoretically the best response, we decided to carry out a brief investigation to determine the optimum response judged on a subjective basis.

Fig. 22 shows an active equaliser constructed for this purpose. The equaliser consists of two active filter stages in cascade. The first stage has a frequency response which is the inverse of the frequency response of the low-frequency loudspeaker system at low frequencies and is flat at high frequencies. The second stage is a 2nd-order high-pass filter with adjustable resonance frequency and Q-factor. Six values of resonance frequency between 20 Hz and 100 Hz, and five values of Q-factor between 0.5 and 2.0 can be selected giving 30 possible response shapes in all. By inserting the equaliser between the pre-amplifier and power-amplifier stages of the driving amplifier, the effect of changing the system resonance frequency and total Q-factor of the loudspeaker system can be simulated. Fig. 23 shows the measured sound-pressure/frequency responses of the low-frequency system (with crossover) when fed via the equaliser for the four extreme settings of the resonance frequency and Q-factor controls.

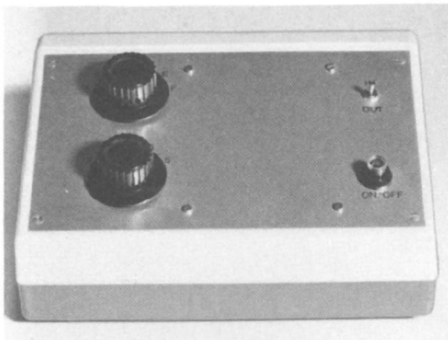


Fig. 22 **Active equaliser constructed to enable instant comparison of different values of the system resonance frequency and total Q-factor of the low-frequency system.**

The main conclusions drawn from subjective tests using the equaliser were:

(i) Variations of the resonance frequency and Q-factor had less effect on the subjective performance than expected.

(ii) Variation of the Q-factor was generally much more noticeable than variation of the resonance frequency.

(iii) Q-factors exceeding 1.0 caused the reproduction to be 'boomy', but much less so for a low resonance frequency than a high one.

(iv) No single response shape stood out as being superior to the rest; however, a low Q-factor is preferred providing that the resonance frequency is also low.

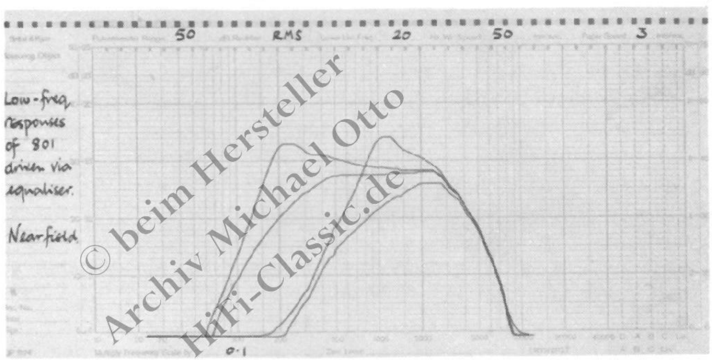
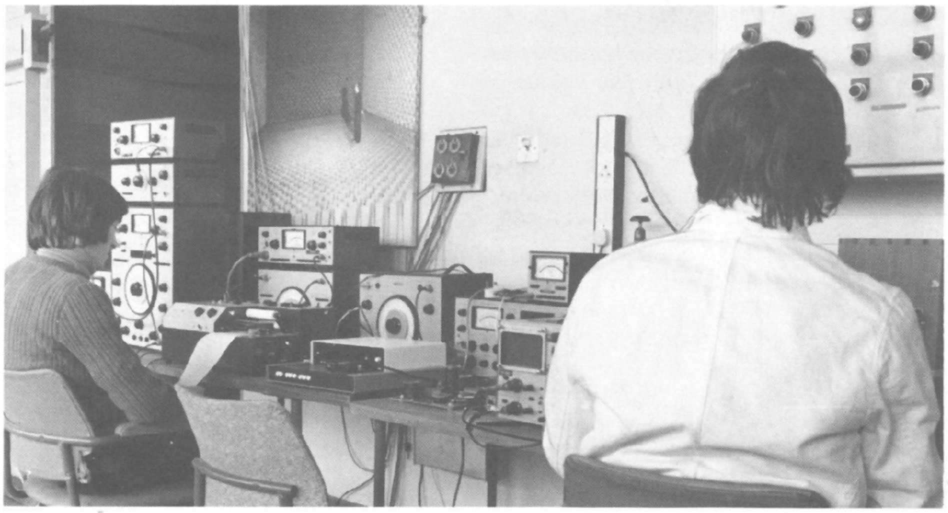


Fig. 23 **Four of the thirty possible frequency responses obtainable from the low-frequency system (with crossover) when driven via the active equaliser.**

Section 7 **Measured Performance Data**



Measurements in progress at the Building Research Establishment's anechoic chamber.

The free-field sound-pressure/frequency responses of the prototype 801 were measured using the large anechoic chamber at the Building Research Establishment in Watford, Hertfordshire. The on-axis response for a measuring distance of 2 m is shown in fig. 24. The responses for off-axis microphone positions are given in figs. 25, 26 & 27. The environmental controls were set in their flat positions for these measurements.

The transient performance of the system was assessed by computing the energy versus time response from the pressure response measured at the microphone for a 25 μs-wide rectangular input voltage pulse. Fig. 28 shows the measured energy response of Model 801 for a microphone distance of 2 m.

Measurements of the harmonic distortion of the sound-pressure output for a sine-wave input voltage were made using the anechoic chamber in the B&W laboratories. The amplitudes of the 2nd, 3rd, 4th and 5th harmonics versus frequency measured for an rms input voltage of 9.5 V are shown in figs. 29 & 30.

The measured modulus and phase angle of the input impedance are given in figs. 31 & 32 respectively.

The sensitivity of the prototype 801 measured for a microphone distance of 1 m using a 300 Hz sine-wave rms input voltage of 2.83 V was 85 dB. This input voltage corresponds to a power of 1W into an 8Ω resistive load.

Fig. 24 On-axis free-field response of the prototype 801 measured at 2 m.

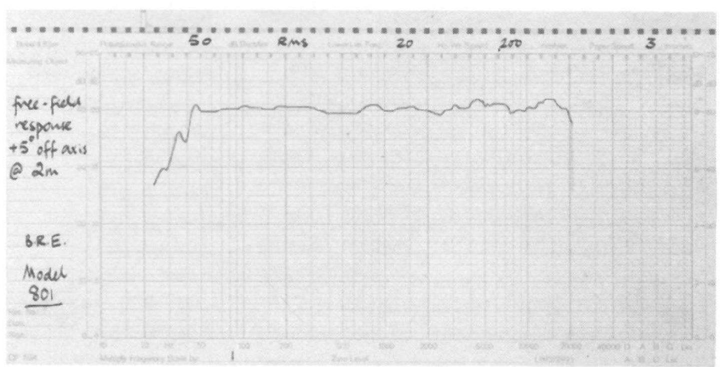


Fig. 25 Free-field response measured at 2 m for + 5° off-axis in a vertical direction.

Fig. 26 Free-field response measured at 2 m for − 5° off-axis in a vertical direction.

Measuring energy versus time response of the 801 using a B&K 2031 spectrum analyser interfaced to HP 9825A desk-top computer.

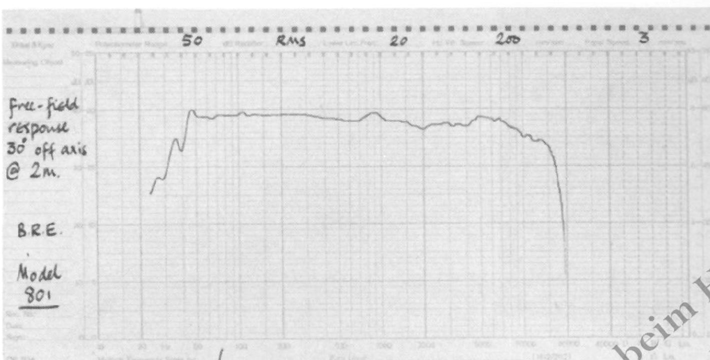


Fig. 27 Free-field response measured at 2 m for 30° off-axis in a horizontal direction.

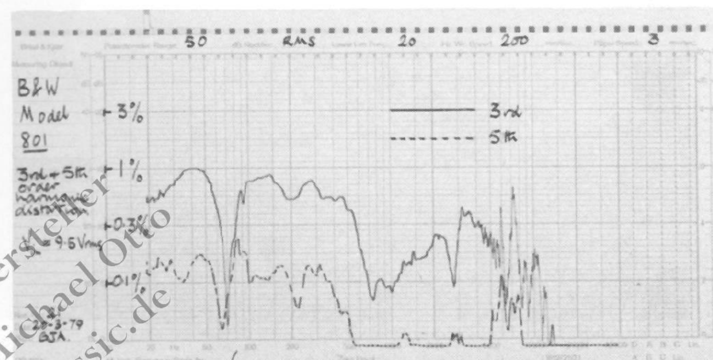


Fig. 30 Amplitudes of 3rd- and 5th-order harmonic distortions for a sine-wave rms input voltage of 9.5 V.

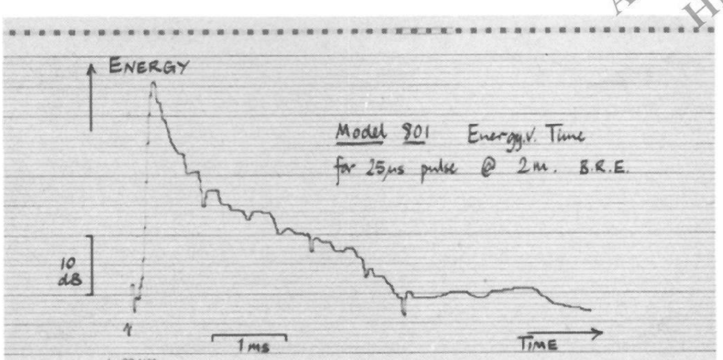


Fig. 28 Energy versus time response measured at 2 m for a 25μs-wide rectangular input voltage pulse.

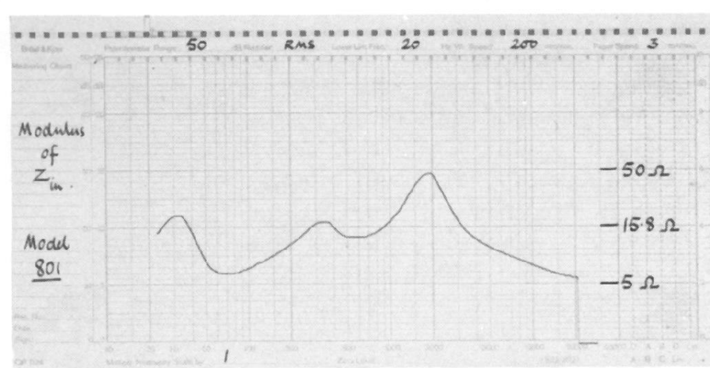


Fig.31 Measured modulus of input terminal impedance.

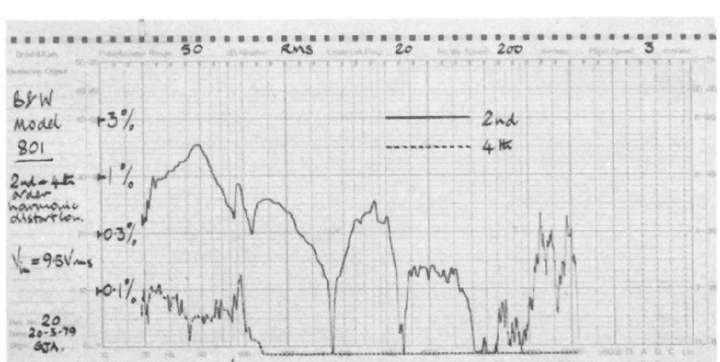


Fig. 29 Amplitudes of 2nd- and 4th-order harmonic distortions for a sine-wave rms input voltage of 9.5 V.

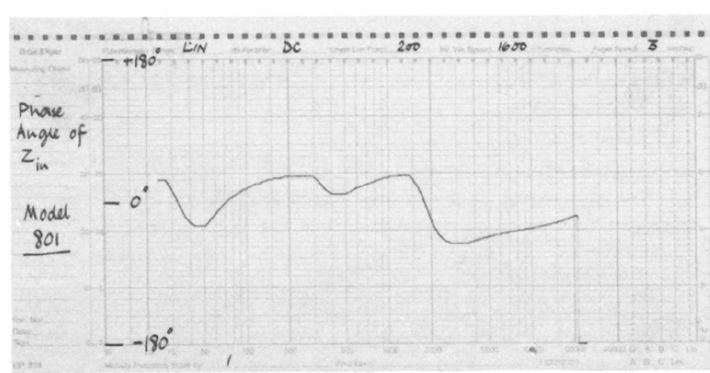


Fig. 32 Measured phase angle of input terminal impedance.

Live versus Recorded Tests

As part of the subjective tests carried out on Model 801, we tape recorded a number of musical instruments so that the reproduced sound could be compared directly with the live sound. Ideally the instruments should be recorded in anechoic conditions to ensure that the recorded sounds are free from reverberation. A wide variety of percussive and woodwind instruments were recorded in the anechoic chambers of the Building Research Establishment and B&W laboratories. Recordings of male speech were found to be particularly useful when compared with the real thing.

Good subjective tests still convey information which is not easily deduced from the measured performance of the system. The B&W Model 801 is, we believe, an outstanding performer which owes its success to the use of both high technology—and well-trained ears.

Recording musical instruments in an anechoic environment for live versus recorded tests of the 801.

Acknowledgements
We would like to thank the Material Physics Division, A.E.R.E. Harwell for supplying us with the Laser Interferometry equipment and for their advice on its use.

We are also indebted to the Building Research Establishment, Watford, for the use of their anechoic chamber and to Pentagram Design for their enthusiasm and interest in this project.

Acknowledgements by John Bowers
I should like to acknowledge the dedication and effort of the design team responsible for Model 801:

Dennis Ward
Steven Roe
Glyn Adams
Ray Ellaby
Andrew Monteith
Ray Greenwood
Kenneth Grange
(Pentagram Design Partnership)

Their enthusiasm and perseverance in working to a most ambitious specification has in my opinion been totally justified in the final product.

References

1. **A. N. Thiele,** *''Loudspeakers in Vented Boxes,''* J. Audio Eng. Soc., vol.19, pp.382-392 (May 1971) and pp.471-483 (June 1971).
2. **R. H. Small,** *''Closed-Box Loudspeaker Systems, Part 1: Analysis,''* J. Audio Eng. Soc., vol.20, pp.798-808 (December 1972).
3. **G. J. Adams,** *''Computer-Aided Loudspeaker System Design, Part 2: Determination of System Power Ratings and Some Design Examples Using Optimisation Techniques,''* J. Audio Eng. Soc., vol.26, pp.922-929 (December 1978).
4. **F. J. M. Frankort,** *''Vibration and Sound Radiation of Loudspeaker Cones,''* Philips Res. Repts. Suppl., No.2 (1975).
5. **D. B. Keele, Jr.,** *''Low-Frequency Loudspeaker Assessment by Nearfield Sound-Pressure Measurement,''* J. Audio Eng. Soc., vol.22, pp.154-162 (April 1974).
6. **S. H. Linkwitz,** *''Active Crossover Networks for Noncoincident Drivers,''* J. Audio Eng. Soc., vol.24, pp.2-8 (January/February 1976).
7. **G. J. Adams,** *''Computer-Aided Loudspeaker System Design, Part 1: Synthesis Using Optimisation Techniques,''* vol.26, pp.826-837 (November 1978).

B&W Loudspeakers

1973 1978

Meadow Road Worthing West Sussex BN11 2RX England
Telephone (0903) 205611 Telex 87342

Printed in England GSB 5814/79