CAD OF DEDICATED HIGH POWER MODULAR LOUDSPEAKER CABINETS FOR CONCERT SOUND REINFORCEMENT

PREPRINT N° 2085 (F8)

BY
E. VINCENOT
PH. ROBINEAU
NEXO
PARIS
FRANCE

**Presented at the 75th Convention
1984 March 27-30
Paris**

AN AUDIO ENGINEERING SOCIETY PREPRINT

C.A.D. OF HIGH POWER LOUDSPEAKER CABINETS FOR CONCERT SOUND REINFORCEMENT

Eric VINCENOT
NEXO
Paris, France.

Philippe ROBINEAU
Paris, France.

ABSTRACT

User requirements for modern modular high-power concert sound reinforcement systems dictate new advances in loudspeaker cabinet design . An elaborate C.A.D system is set up. The results lead to specific solutions used in prototypes, whose performance are measured and compared to simulations. The complex transferance and impedance data are then stored in the C.A.D. system for filter design and optimisation. Use and performance of new cabinets in high power sound reinforcement are discussed.

INTRODUCTION

ENVIRONMENT CHANGES IN THE ENTERTAINMENT INDUSTRY

During the last 15 years studio monitors and home listening cabinets have not seen much change in their basic requirements. Control rooms are still about the same volume, human ears have remained pretty much unchanged as well as the ability of neighbours to accept high SPL's in their living room.

On the contrary working environment has changed enormously in the entertainment industry, particularly for live concerts. A manufacturer of sound reinforcement loudspeaker cabinets must now be able to offer a limited number of models covering an extremely wide range of applications. Halls must have a sound system able to cope with the delicate reinforcement of a string quartet as well as with the very high sound pressure levels of a Rock band. Cinemas are faced with increasingly spectacular sound tracks and discotheques now have customers listening the same records on Compact Disc players at home.

The highest requirements are placed on the equipment used by sound equipment hire companies which are now operating in most concerts and large public events. In this case the equipment cannot be optimised for a given location or program type, with

the same cabinets a rental company can be within a few months faced with :
(all examples from real facts)

-Shipping a 15 kW system for a tour in Africa where the audience will range from 3,000 to 25,000 and where complete transport and set up of the sound system will happen between each concert sometimes thousands of miles distant from each other.

-Setting up a sound system for a classical music and theater festival spread over a whole town.

-Allowing audiences of 100,000 people to actually hear something in exceptionnaly large outdoor events in sports, politics or entertainment.

All these large scale set ups must be realised with the same two or three types of cabinets used by the company for day to day work in the usual medium and small sized halls.

The above introduction can be condensed in a simple statement: the average listener and the average customer are just asking now for the same sound quality they achieve at home on their Compact Disc hi-fi system . The fact that acoustic powers several thousand times higher are needed is generally considered by technically untrained people as easily bypassed with actual technology; but how has that technology evolved?

TECHNOLOGY EVOLUTION IN THE ENTERTAINMENT INDUSTRY

For most sound reinforcement equipments this evolution has not been a major problem : microphones, mixing consoles, sound effects, reverberation units, amplifiers are often of similar design to studio equipment. There is enough advance in state-of-the-art studio equipment to ensure that manufacturers of this type of products will not encounter major problems to follow users demand in the future.

Meanwhile for loudspeaker components basic designs remain the same as 20 years ago. Widely used 1" and 2" throat compression drivers, 15" bass drivers have kept the same basic architecture. Reasonable progress has been made on power handling through the use of new materials for adhesives and voice coil formers but as the gap and voice coil volumes have not changed in large proportions the DC resistance increase due to operating temperature tends to cancel any further improvements (ref 1). Ferrofluids may represent a quantum leap but only when these will be available for higher operating temperatures (ref 2 , 3).

Although "Voice of the theatre" type two way systems still represent a significant part of actually operating units for medium sized audiences, cabinet design has evolved in several directions.

The availability of constant directivity horns has widely influenced the design of fixed installations and array layout with these new components has been the subject of numerous papers (ref 4 , 5 , 6) . These horns are generally oriented towards 500-800 Hz crossover frequency two way systems and optimised for excellent directivity control but the physical space occupied compared to the amount of acoustic power that can be generated by a two inch throat compression driver makes them unsuitable for high power modular systems and applications involving portability.

The evolution of their market pushed hire companies since 1970 to use dedicated 3 or 4 way active modular systems incorporating narrow bandwidth high efficiency designs making best use of existing loudspeaker components. The ability to stack up large quantities of powerfull units was achieved but often at the expense of universality, ease of use and overall quality.These 3 or 4 way systems are basically designed unit by unit for each frequency band and filtered by active crossovers of standard theoric design neglecting the critical bandpass response of each unit and complex interactions within the system. A sound engineer is necessary to set up and adjust the system at each installation, even then proper in phase addition at crossover points is not always guaranteed. The freedom left to the user to lay out the system elements often results in improper arrangements. Most important is the fact that these systems are cost effective only for a limited range of applications and appear bulky, complicated and expensive in general sound reinforcement use (theaters, cinemas, discotheques ....).

Some manufacturers also attempted to apply modern high SPL studio monitor design techniques to sound reinforcement in realising full range cabinets of similar design (direct radiating units in the LF and Low Mid frequency range, compression drivers in the HF range) but with active filtering and higher power handling. This proved interesting on ergonomic grounds: relatively small self contained systems with reasonable efficiency (mutual coupling and different LF compromise from studio monitors) with a lot of the practical advantages of the old "Voice of the theatre". However the essentially non modular concept and medium efficiency are major drawbacks for high-power setups .Lobing and poor directivity control result from the large number of stacked full range enclosures as well as adverse effects from excessive mutual coupling of direct radiating loudspeakers (happening when proportional increase of the acoustic load with frequency is not respected within the operating range due to very large dimensions of the array compared to the wavelengths).

As no major technological breakthrough seems to be possible a different approach can be considered in trying to fulfill the following practical and technical criterions while using computer aided engineering techniques at most stages of the development. The goal is to try and prove if a complete system approach can result in a fully optimised product featuring:

- Integrated system design rather than separate components oriented design. This ensuring standard and predictable performance in the field and being the only way to apply global optimisation techniques (what would one think now of a state-of-the-art hi-fi enclosure user assembled and adjusted from brand x,y and z speakers and crossovers?).

- High-efficiency and modularity to allow use in high power applications in an easier way than large specialised systems with a minimum number of individual units and predictable behaviour under mutual loading conditions.

- Volume and weight of the basic units low enough to allow easy transport and installation. Quick and foolproof set-up.

The ergonomic advantages of "Voice-of-the-theatre" systems, large modular high-power components and direct radiation compact cabinets would thus be offered in the new system.

DESCRIPTION OF THE CAD SYSTEM

It is not our purpose to describe sophisticated numerical techniques neither to discuss the complex physical problems involved. For more detailed developpements on theoretical points one may refer to the quoted references. Furthermore we think that the most important feature of an efficient CAD system must be to allow synthetic evaluation on most aspects of product performance, rather than very accurate analysis on isolated points. As an example a recently published paper by K.O.BALLAGH (ref 7) may well prove that classical extensive analysis on direct radiator loudspeakers LF alignments ,neglecting real boundary effects, proves less efficient than numerical methods involving even simplified room load simulations.

To allow complete evaluation of each component and of the whole system, the CAD involves four calculation blocks used separately or interactively:

- acoustical load on the diaphram
- analog circuit computations
- directivity evaluation
- complete system simulation (includes filters)

The following overview is made with specific accent on important points for sound reinforcement system design.

ACOUSTICAL LOAD ON THE DIAPHRAGM

The calculation in case of a direct-radiating loudspeaker is often reduced to computing the radiation impedance of the cone by using the rigid circular piston model. The piston can be assumed as mounted in an infinite rigid wall or in the end of a long tube which is generally in good agreement with practical configurations (ref 8 , 15).

However this model is not appropriate for, frequently square, horn mouths, we then tried a rectangular piston model whose radiation impedance has been studied in both baffled and unbaffled conditions (ref 9 ,21). As for the circular piston, asymptotic analytical functions are used to approximate the effective load for different ranges of values of the variable.

The results from these different models were compared and usefull points for the accuracy of the simulation verifyed :

- the two boundary conditions (baffled or unbaffled piston) must be available in the program, this possibility is important for reinforcement systems (ex: "flown" cabinets in open air and recessed wall mounting)
- in the same configuration, the radiation impedance of a square piston is nearly equivalent to that of a circular one having an equivalent emissive area.
- for a rectangular piston the radiation impedance is greatly dependant on the width to height aspect ratio.

Taking into account the so-called acoustical mutual coupling is necessary to incorporate the effects of 2 important practical cases :

- stacking of several cabinets
- sources located in the neighbourhood of reflecting areas (i.e. ground)

Assuming that high-power sound reinforcement cabinets must fit in large systems, their design must include simulation of their performance when being stacked or placed side by side. We know that in the case of a number N of adjacent areas S radiating with equal phase and equal amplitude, the coupling effect is reduced to a fictive increase of the radiation area of each source : the radiation impedance of the sources becomes a function of $2\pi\sqrt{NS}/\lambda$ instead of $2\pi\sqrt{S}/\lambda$. We must however note that coupling also changes the aspect ratio, this effect must be taken into account.

The influence of reflecting areas may be analyzed by introducing mirror image sources. These effects can therefore be modelised in a similar way to the mutual coupling.

As a conclusion to this point, the calculation implies the following parameters :

- baffled or unbaffled boundary conditions
- radiating area S of a source
- number of sources N including mirror images
- overall aspect ratio including mirror images
- spacing of the sources including mirror images

Designing horn-loaded cabinets implies proper calculation tools to simulate the horn transfer of the mouth acoustical load towards the throat.

Although many publications have been made on this subject (ref 10 , 11)we will describe briefly a general method of calculation fitted to computer use.

Assuming plane wave propagation in a direction x, the general equation of horns may be formulated as :

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{1}{S}\frac{\partial S}{\partial x}\frac{\partial \phi}{\partial x} - \frac{1}{c^2}\frac{\partial^2 \phi}{\partial t^2} = 0$$

where $\phi(x)$ is the velocity potential
$S = S(x)$ is the section of the horn

Intoducing the wave number $k \,|\, k^2\phi = -\frac{1}{c^2}\frac{\partial^2 \phi}{\partial t^2}$ yields to a second form :

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{1}{S}\frac{\partial S}{\partial x}\frac{\partial \phi}{\partial x} + k^2\phi = 0 \qquad (1)$$

For certain forms of the functions S(x), this differential equation can be integrated and allows an analytical solution for (x), from which we can derive both acoustical pressure and velocity using the basic relations :

$$\begin{cases} p(x) = -\rho \dfrac{\partial \phi}{\partial t} & (2) \\ v(x) = \dfrac{\partial \phi}{\partial x} & (3) \end{cases}$$

As an example for the classic exponential horn where $S = So.e^{2ax}$,it becomes :

$$\text{for } k^2 \geqslant a^2 \qquad \phi(x) = e^{-ax}.\left(X_1 e^{-jbx} + X_2 e^{+jbx}\right)$$

$$\text{for } k^2 < a^2 \qquad \phi(x) = e^{-ax}.\left(X_1' \operatorname{ch} bx + X_2' \operatorname{sh} bx\right)$$

$$\text{where} \quad b = \sqrt{|k^2 - a^2|}$$

The two forms are respectively "propagative" and "evanescent". In a general way, the propagative solutions of (1) can be reduced to a sum of 2 terms :

$$\phi(x) = \phi_+(x) + \phi_-(x)$$

where $\phi_+(x)$and $\phi_-(x)$ correspond respectively to a propagation towards positive x (e^{-jbx} for the expo.) and negative x (e^{+jbx} for the expo.). From a physical point of view these 2 terms can be considered as direct and reflected waves, knowing that their amplitude ratio is only related to the boundary conditions.

Except for the ideal case of the infinite horn (no reflected wave meaning $\phi_-(x)$ is 0), determining the ratio p/v (acoustical impedance) at an arbitrary point x allows computation at any other point using the relations (2) and (3). The only basic condition of the calculation is the integration of equation (1).

Assuming this condition is achieved, an appropriate method is based on a quadripole description (quadripole is meant in an electrical sense).

A duct segment of variable section S(x), in which acoustical pressure p(x) and velocity v(x) are known and continuous functions of x upon an interval x1,x2, is considered as a quadripole whose transfer matrix [T] is defined with its 4 complex elements by the relation :

$$\begin{bmatrix} p1 \\ U1 \end{bmatrix} = \begin{bmatrix} t11 & t12 \\ t21 & t22 \end{bmatrix} \begin{bmatrix} p2 \\ U2 \end{bmatrix}$$

where $p1 = p(x1)$, $U1 = U(x1) = S(x1).v(x1)$
$p2 = p(x2)$, $U2 = U(x2) = S(x2).v(x2)$

If the system is linear, the 4 coefficients are not dependent on the input and so :

$$\frac{p1}{U1} = \frac{t11\ p2 + t12\ U2}{t21\ p2 + t22\ U2}$$

or, if $Z1 = p1/U1$ and $Z2 = p2/U2$

$$Z1 = \frac{t11\ Z2 + t12}{t21\ Z2 + t22}$$

The coefficients of T are derived after calculating the functions p(x) and U(x) at both x1 and x2 points. For instance in an exponential horn :

$$p2 = e^{-ax_2}\left(A\, e^{-jbx_2} + B\, e^{+jbx_2}\right)$$

$$U2 = e^{-ax_2}\left(A.(jb+a).e^{-jbx_2} - B.(jb-a).e^{+jbx_2}\right)$$

exchanging the variable pairs (p2,U2) and (A,B), then calculating p1 and U1 using A and B, gives p1 and U1 versus p2 and U2 and so finally the 4 coefficients t11,t12,t21,t22 by identification.

This method presents 3 main features :

- to allow computation of Z1 in two steps, first computing T (only dependent on the horn itself) then Z2, so that we can separately change the radiation impedance at the mouth (depending on boundary conditions and mutual coupling) without full recalculation.
- since we use transfer matrix, chaining several segments with different characteristics is equivalent to cascading several quadripoles and leads to multiplication of 2x2 matrixes. This can prove particularly interesting as in the case of these following examples :
- calculation of the load for a constant directivity horn featuring flanges and diffraction throat, including the influence of the tapered driver throat and phase plug (Fig 3).
- calculation of the load for a segmented bass horn accounting for the manufacturing approximations.
- analysis of the influence of discontinuities (ref 12).

- in some cases we are interested in data other than the throat impedance, for example the phase relation between Ut and Um (acoustic volume velocity respectively at the throat and at the mouth). This is obtained directly from the overall transfer matrix :

$$Ut = t21.pm + t22.Um$$

then since pm = Zm.Um

$$Um = \frac{1}{t21.Zm + t22} Ut$$

finally the phase relation between complex arguments becomes:

$$Arg(Um) = Arg(Ut) - Arg(t21.Zm+t22)$$

In practice, the computation program which has been developed allows to simulate a horn having 1 to 10 chained segments with different specifications among 3 standard types :

- constant section pipe
- parabolic horn
- expansion laws according to the following equation where T can vary from 0 to infinity representing most classical horns with the variation of this single parameter (ref 13):

$$S(x) = So.(chax + T\ shax)$$

As shown in reference, T = 0 implies hyperbolic expansion law, T = 1, exponential law , T = ∞ conical law.

ANALOG CIRCUIT COMPUTATIONS

Using computers may theoretically allow analog circuits to describe mechanical-acoustical systems including both "located constants" or "shared constants" phenomenons if the latter are unidirectionally propagated (the limits being reduced to pluridimensional modal effects, ie. diffraction or non rigid membranes). However bearing in mind that entering an increased number of parameters will subsequently increase the computation time and make the analysis difficult, it is convenient to reduce this number according to what accuracy is expected.

Thus, starting from the general model of an electrodynamic driver shown by fig. 1 (seen from the acoustical point of view), where Zr and Zf are the acoustic loads on respectively the rear and front side of the diaphragm, we can deduce over a reasonably wide frequency range most of the driver's practical performances with the circuit of fig. 2.

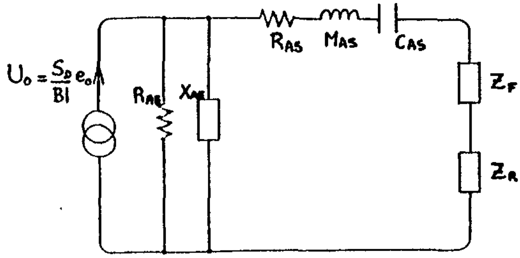


fig. 1

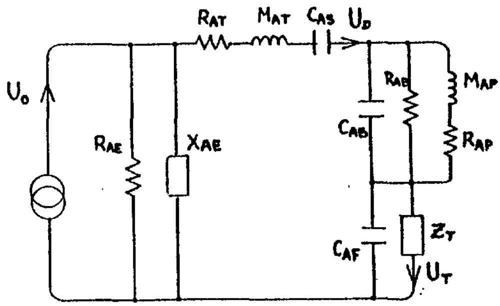


fig. 2

This circuit can be used with most classical cabinet types according to usually admitted simplifications: for closed boxes R_{AP} will be infinite and M_{AP} will be 0 , for a direct radiation loudspeaker C_{AF} (acoustic compliance of compression chamber) will 0.

No particular comment is needed for this circuit diagram except for :

- R_{AT} and M_{AT} may include the complex radiation impedance on the rear side of the diaphragm.

-

$$R_{AE} = \frac{(Bl)^2}{S_D^2} \cdot \frac{R_E}{R_E^2 + X_E^2}$$

$$X_{AE} = -\frac{(Bl)^2}{S_D^2} \cdot \frac{X_E}{R_E^2 + X_E^2}$$

where R_E and X_E are respectively the real and imaginary components of the non motional part of the electrical impedance.

- The external source e_o is represented as an acoustic current generator equal to the volume velocity U_o , rather than as an acoustic voltage generator : although both are equivalent as viewed by the load, only the first is also valid as viewed by the source allowing direct computation of the overall dissipated power.

Using a program based for most cases on that model, we derive from classical relations the following information as function of frequency:

- Power sensitivity
- Acoustic phase
- Efficiency
- Complex input impedance
- Diaphragm excursion
- Diaphragm acceleration
- Electrical power dissipated in the voice coil
- Volume velocity in the vents
- Phase relation between horn and vent

Power sensitivity S is defined as the ratio of the acoustic power radiated P_{AR} to a reference electric power inversely proportional to the DC resistance of the voice coil R

$$S = \frac{P_{AR}}{\frac{1}{2}\frac{e_0^2}{R_{cc}}}$$

Ideal impedance matching sets its theoretical maximum at -6 dB.

Efficiency is defined as the ratio of radiated acoustic power to the total electric power effectively dissipated. Infinite load impedance sets its theoretical maximum at 0 dB (100%).

Diaphragm excursion, diaphragm acceleration and electrical power dissipated in the voice coil are major results to evaluate the behaviour of sound reinforcement components under heavy use. These data are also necessary for correct design of passive and active filters, equalisers and protection networks.

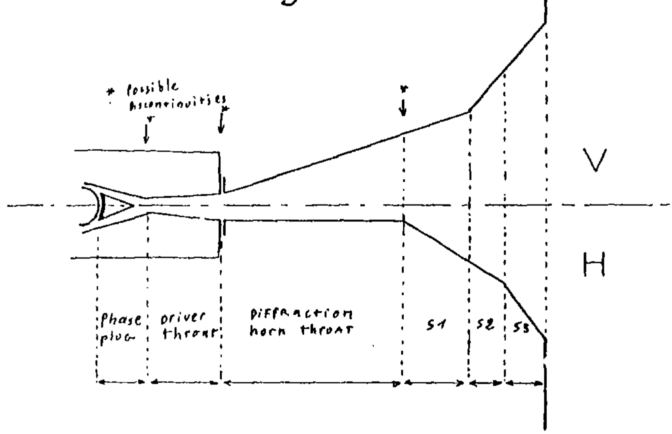
Acoustic phase response and complex input impedance are necessary to transfer matrix calculations implemented in the system simulation block.

DIRECTIVITY EVALUATION

Although exact prediction and calculation of the spatial radiation of a given horn is extremely complex, generally making necessary the use of sophisticated and heavy mathematical methods such as finite elements (ref 14). Experimental research has allowed the establishment of several simple but efficient algorithms for directivity calculation used in modern horns (ref 4 , 5). These, allied to simplified results on plane circular and rectangular surfaces (ref 15) can be used to roughly estimate directivity in both planes and Q.

Coupling the directivity and Q estimation routine with the acoustic impedance computation allows to simultaneously view the effects of horn shape modifications on power sensitivity, diaphragm displacement, axial response and spatial dispersion.

fig. 3



SYSTEM SIMULATION

Using computers is particularly interesting for filter network design. One basic reason is, in the case of passive components, the possibility to take into account the complex impedance of the transducers. As a general overview we can say that the computer allows to work on active or passive filtering and equalisation networks as components of a whole system, thus avoiding two mistakes:

- viewing the filter network only in terms of its own transfer function rather than as an element contributing to an overall system transfer function (frequent with large multiway systems involving general purpose active crossovers).
- not taking into account the effects of filtering upon spatial performance of a system. The differences between additions of electrical signals and additions of soundwaves in three dimensional spaces are too often overlooked in sound reinforcement system design

From this basis the C.A.D. program has to satisfy the following 2 categories of requirements :

* "system" oriented criterions:
 - possibility to view the overall response of each transducer including all components, therefore implying an interface with other computation routines.
 - simulation of the acoustic addition of several sources with crossover networks, this implies take into account the complete transfer function of each transducer including phase data.

* efficiency and accuracy requirements:
 - ability to simulate either active or passive components
 - calculations made with complex impedance data of the transducers.
 -possibility to work on either simulated or true (measured) transducers.
 - ability to apply time lags to the different sources .
 - ability to evaluate the effects of electric networks and geometrical positioning upon spatial performances
 - evaluation of the influence of networks on large signal behaviour (temperature, acceleration, displacement).

One important choice was to enable programmed s uctures for filters rather than a fixed sructure supposed to represent all usual configurations. The inconvenients of the latter are obvious:
 - choosing the appropriate structure involves delicate compromise between versatility and calculation speed.
 - in practice, considering the networks as diagrams with sometimes an unnecessary large number of components will tend to confuse user's view.

The principle of calculation is practically dictated by the fact that, in case of passive filters, we want to derive the input impedance of each network as well as its transfer function. We therefore have to deal with both voltage and current, a quadripolar model becomes then necessary. The choice of transfer matrix as a basic calculation element is also evident since it yields an efficient method to enable programmed structures.

According to that method, the electric diagram designed by the user is split into a cascade of elementary quadripoles whose types are indexed in a directory involving standard passive and active networks. For each of these elementary networks, 8 relations have been previously stored, yielding the 4 complex coefficients of its transfer matrix as functions of a stated number of specific parameters. Implementation of the filtering network is then achieved as follows:
 - specify the output under consideration (1 of 4 simultaneous channels)
 - set the number of cascaded quadripoles
 - specify the type of each quadripole
 - set the values of each specific parameter (automatically

selected by the program after the previous step).

After computation of the transfer matrix the computer has to store 8 real data (the 4 complex coefficients t11, t12, t21, t22) for each frequency point.
From the basic relation :

$$\begin{bmatrix} v1 \\ i1 \end{bmatrix} = \begin{bmatrix} t11 & t12 \\ t21 & t22 \end{bmatrix} \begin{bmatrix} v2 \\ i2 \end{bmatrix}$$

as an example we then derive :

$$F(j\omega) = \frac{v2}{v1} = \frac{1}{t12/Z2 + t11}$$

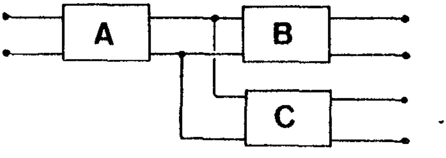
as complex electric transfer function (Z2 being the impedance of the transducer)

$$\frac{v1}{i1} = \frac{t11\,Z2 + t12}{t21\,Z2 + t22}$$

as input impedance

Note that for active quadripoles we assume that they are "impedance transparent" (without influence upon overall input impedance and not affected by their load impedance) which yields a diagonal form for their transfer matrix (t12 and t21 being 0) with t22= t11.
In a few cases, the computation has to be split in several parts, mostly when the filter networks of 2 different outputs involve a common portion (fig. 4). In a first step the complex transfert functions for both B and C networks are computed and stored as well as their common input impedance. These are then reentered as transducer data to compute the A network.

Fig 4



The entire block involves 4 main routines :
- Edition of electrical data (previously detailed)
- Edition of electroacoustical data
- Specific calculations
- Utilities

The electroacoustic data for section No 2 (complex impedance and complex transferance of transducers) are stored either from simulations or from measurements. Section No 3 contains all the computation routines whose practical purpose will be discussed later:

- Electrical T.F. (transfer function)
- Acoustical T.F.
- Sum of electical T.F.
- Sum of acoustical T.F.
- Moduli Sum and Subtraction
- Phase difference
- Input impedance
- Electric power in the voice coil
- Diaphragm excursion and acceleration
- Group delay

DESIGN USING THE CAD SYSTEM

The main drawback encountered in the use of highly detailed analytical methods on specific points in cabinet design is to draw the attention of the engineer towards a limited set of goals at the expense of a synthetic view on the finished product.

The complete CAD system enabled the designers to quickly grasp complex relationships between apparently unimportant modifications and the finished product, this resulted in constant feedback from the marketing department.

As the primary goal was to achieve a cabinet where the main advantages of high power modular systems would be offered in an easy to use optimised all-round format, the first runs on the computer were used to verify the feasibility of the project. We will not here develop that lengthty part of the design work as specific end results will be given further. Let it just be said that at that early stage of simulation it became possible to set up very precise limits on the technology, cost, volume and apect ratio of each individual radiating element as well as to determine the features of the finished product including purely ergonomic considerations. Thus each manufactured part could be individually optimised in relation to the other elements.

A few examples can be given, some of which may appear irrelevant to domestic or studio equipment specialists, all of them were nevertheless important and taken in consideration at the simulation stage (little "details" like "superb design but nowhere to ergonomically fit the handles, wheels, plugs, etc... without affecting performance" can be particularly infuriating when working on equipment where practical considerations will be the prime user's concern) :

ERGONOMICS

TOTAL VOLUME: 0.5 m3 (User requirement)
VOLUME OF BASS ELEMENT: 0.4 m3 (From first simulation run)
VOLUME OF MID-TREBLE ELEMENT: 0.133 m3
ASPECT OF ELEMENTS: rectangular for both elements (User requirement)
ASPECT RATIO FOR THE BASS ELEMENT: 3:2:X
ASPECT RATIO FOR THE MID-TREBLE ELEMENT: 1:2:X (same units as

above for modularity)
WEIGHT OF BASS ELEMENT: 75 Kgs MAX (two person transport)
WEIGHT OF MID-TREBLE ELEMENT: 35 Kgs MAX. (one person transport)

PERFORMANCES

MAXIMUM SPL: comparable to dedicated high power modular systems
EFFICIENCY: "" " "" "" "" ""
LF EXTENSION: at least equivalent to common sized modular LF horns
DIRECTIVITY CONTROL: optimum within the size limits, minimum influence of stacking by modular design
FILTERING AND EQ : the system should be able to operate either in passive, two way active and three way active configuration without modifications on the cabinet other than switch configurations. In each case the use of external adjustments as well as the necessity for additional equalisers is precluded except for large setups. Active and passive crossovers will be specifically designed with the system including for each configuration all necessary LF and HF protection, sensitivity alignments and equalisation (necessary if constant directivity horns are present). Proper behaviour of the system in the crossover areas must be insured in each operating mode without mechanical alignment of the relative position of the drivers, delay networks must thus be used when combined design of the individual horns proves insufficient.

SPECIFIC RESULTS

Although we will here describe specific results of some interest these must not be considered as independent of the general simulation on the overall design . Different system goals like non modular, larger or smaller cabinet size will lead to different conclusions particularly for the choice between horn and direct radiation.

BASS ELEMENT

Optimisation was here oriented towards best low frequency extension within a given volume with a minimum limit on efficiency. This had obviously been the subject of extensive well known work but not for the case of a horn system with such a wide range of variables (chained finite horn sections with variable expansion laws, possibility to accurately simulate the horn-vent combination if necessary, influence of mutual coupling). Although extremely lengthy due to the large number of variables, the simulation proved very encouraging. With regards to the limited volume available we had first thought that significant sacrifices would be necessary in the LF extension. On the contrary the "optimum electronic prototype" showed significantly better LF performance than our previous, much larger, modular design , along with similar sensitivity and efficiency. The earlier design had nevertheless been made with the help of a much simpler version of the present program, was thought of as near optimum and very well considered amongst our customers. Measured results agreed very well with the simulations, as an example simulated and measured impedance curves are shown in fig 9 and 10 .

For this given format an optimised horn loaded cabinet proved to be the best solution in terms of LF response, sensitivity, efficiency and diaphragm displacement. Excessive diaphragm displacements and adverse effects of mutual coupling in large systems were thus avoided. Simulations on the latter point, confirmed by available litterature, showed that beyond a point up to which mutual coupling proves usefull (ref 16), unacceptable influence on the power sensitivity response of direct radiator elements is present.

SPECIALISED MIDRANGE DRIVER:

This is certainly the most important result and the key element in the finished product design. We have been working on high-efficiency midrange drivers since 1979 and manufacturing these since early 1981 (French patent Jan 81 - US patent pending), this could be the subject of an entire paper but we will restrict ourselves to general considerations and practical conclusions and refer the reader to a Nov 81 publication (ref 17).

The use of 3 or 4 way systems with main crossover points loacated in the region of 250 Hz and 2 Khz is well supported by theory and experience in optimised extended bandwidth high power designs:

-The requirements for maximum output in the LF spectrum are usually fulfilled at the expense of midrange performance, folded bass horns representing an extreme example.

- High quality HF compression drivers prove cost effective only if used with high crest factor signals and far from their displacement limited range. In practice this dictates crossover frequencies higher than 1.5 Khz - In active crossover configurations using industry standard power amplifiers a reduction in crest factor of signals applied to the LF channel is necessary to avoid heavy clipping before reaching thermal and displacement limits on the low frequency drivers. In practice this dictates a crossover frequency lower than 400 Hz.

For full justification the last two points must be related to available statistics on music signals (ref 18). Conversely theory, practical experience and these statistics indicate that, as well as being the most critical on subjective grounds, considerable energy is present in the 200Hz-2 kHz band.

The feasibility study on the CAD system clearly indicated the possibility to achieve very high performance with reasonable physical parameters, but also that all known professional drivers were unsuitable for that use. It was then decided to design, test and manufacture such a driver. This component having been used successfully in our previous designs oriented towards large high-power sound reinforcement systems was since the early stages of the present development considered as mandatory .

One important question arose : could this driver offer good enough LF performance in the reduced space allocated knowing that the horns we had previously produced for it were much larger? Fine optimisation on the CAD system resulted in a horn of overall 31 cm depth , 46.5 cm width and 30.5 cm height (including mounting flanges). The simulated throat impedance curve is shown in fig 13 and the anechoic on axis response curve in fig 14 .

The sensitivity of the two-driver system being very high these are manufactured in 16 Ohms nominal impedance to avoid adverse effects of resistive attenuators on frequency response. The large total throat area allows a threefold reduction in air nonlinearity generated distorsions compared to standard 2" throat compression drivers if these were able to withstand identical acoustic power output.

TREBLE HORN

Similar fine optimisation on a compound horn was applied to the treble element resulting in a horn of well controlled vertical and horizontal polar behaviour and LF loading. The results shown in fig 15 can be related again to the overall dimensions of 25.5 cm depth , 18.5 cm width and 30.5 cm height (including mounting flanges). This and the midrange horn are flat fronted and designed for simultaneous use in the peculiar arrangement shown in fig 24 and justified further on.

SYSTEM DESIGN

Several basic features were predetermined as required from other important choices such as :

- enabling three fully compatible filter configurations (entirely passive , two way and three way active)
- yielding correct horizontal directivity while midrange and high frequency units were placed side by side to allow modularity.

Taking these into account, the structure first had to be of arborescent type and to include non interactive permanent and removable sections . Last the phase relation between units had to be adjusted when necessary (all-pass networks).

Reducing the whole filter to specific networks of five basic types (i.e. cutoff, protection, equalisation, attenuation, phase adjustment) gave a functional diagram shown in fig 5 (impedance compensation networks omitted).

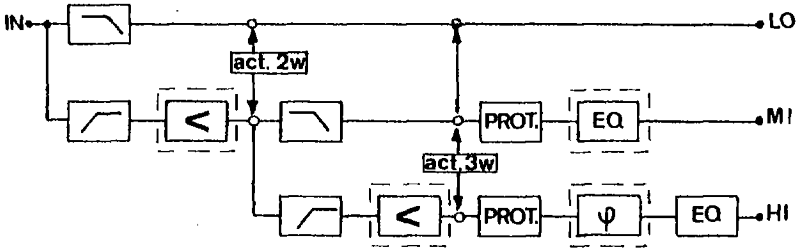


Fig. 5

As said previously the network was effectively optimised using true measured data for complex transfer function and complex impedance of each transducer. It must be emphasised that most particular care has been taken for phase response measurements as required for accurate prediction of acoustical recombinations. A first important result was the phase relation between midrange and treble units : since we had considered this feature as a major criterion for design of the horns, we could verify (in good agreement with predicted results) that no specific all-pass network was necessary to obtain correct acoustic recombination.

The electrical frequency response of the filters is only considered as usefull in terms of its influence on mechanical and thermal behaviour of the drivers;the following filter analysis is made on final bandpass filters obtained by combining the acoustic element and the filter.

The major point to be considered at is that,in large sound reinforcement arrays, the effect of filtering networks upon spatial behaviour is as important as on axis frequency response. As has been put in evidence in several reference papers (ref 19 , 20), the spatial effects (ie beaming and lobe shifting) which are relevant to the relative distances between the acoustic sources tend to become critical in arrays involving large interdriver spacing. From the fact that,in the farfield, the off axis combination of two noncoincident sources is equivalent to a difference in respective path length, we can simulate the performance on a single listening axis by incorporating proper time lags between the elements. However an efficient means of global evaluation is possible, based on simulation of what we have called "Moduli Sum and Subtraction" (sum and difference of moduli of two vectors representing the transfer functions). It is clear that the combined transfer function of the vectors of two overlapping bandpass units involving a variable time lag will in all cases vary between extreme limits set by the moduli sum (if $\Delta\varphi = n\ 360°$) and the moduli difference ($\Delta\varphi = n\ 180°$) ; increasing time lags will tend to produce a comb filter . Considering the special combination as to be analysed with respect to 2 main effects, the previous result yields basic criterions for the evaluation of filter performance:

- the lobe shifting will be viewed by comparing MS to vectorial sum curves and moreover MD to vectorial subtraction curves: an ideal performance will be obtained in case of perfect identity, proving on-axis stability of main lobe over the whole frequency range (ex: theoretical L4 and B4). This concept is in fact close to that of "Normalised Lobing Error" introduced in ref 20.

- "Mutual Interaction Bandwidth" (MIB) may be viewed as represented by the total area included within the two curves plotting MS and MD versus frequency. This feature effectively provides the bandwidth of interaction as a specific performance of crossovers .

CONCLUSION

Some basic features and final performances of the finished product are shown in fig 6 to 24. We are nevertheless deeply convinced that the techniques used for its conception are not truly reflected in these isolated results. System optimisation applied to all the components could be better put in evidence by analysing the performance flaws when these are predicted tradeoffs for ergonomy made at an early design stage in search for the optimum "product for the job". This goes beyond simple budget objectives and sketches given to the engineer by the marketing department as the use of complete system simulation can often put results in a different light than detailed theoretical analysis based on existing litterature. Specialised cabinets for sound reinforcement, specially if they have to be compatible with either fixed, mobile, single unit and high power modular installations, are working tools in an industrial sense while being simultaneously the "control room monitors" for the operating sound engineer and the "personal hi-fi system" for the entire audience. To meet all these requirements CAD is particularly important.Although numerous practical papers have been published on sound reinforcement cabinet design, most theoretical research is only directly applicable to "state of the art" home and studio systems.

The program shown is in constant evolution since 1979 and development is under way to directly interface it with measurement systems (analog recorders and DSP units). This will hopefully evolve into a complete research, development and production tool ranging from theoretical feasibility to production control through prototype design and measurement.

REFERENCES

1 J.KING , "Loudspeaker Voice Coils", J.A.E.S., 18, 34-43, 1970.

2 J.KING , "The Use of Magnetic Fluids in Loudspeaker design", A.E.S. Preprint 1246 (K-1), 1977.

3 W. BOTTENBERG, L. MELILLO, K.RAJ, " The Dependence of Loudspeaker Design Parameters on the Properties of Magnetic Fluids" Presented A.E.S. 61st Convention, 1978.

4 KEELE D.B. Jr, "What's so Sacred About Exponential Horns " , A.E.S. Preprint 1038

5 HENRICKSEN,C.A.;UREDA,M.S., "Blasphemy - The Manta Ray Horns !", A.E.S. Preprint 1288.

6 T.UZZLE , "Loudspeaker Coverage by Architectural Mapping", A.E.S. Preprint 1786, 1981

7 K.O.BALLAGH , "Optimum Loudspeaker Placement Near Reflecting Planes", J.A.E.S. ,12,931, 1983

8 H.LEVINE and J.SCHWINGER, "On the Radiation of Sound from an Unflanged Circular Pipe", Physical Review vol 73, No4, 1948.

9 H.G.DAVIES, "Acoustic Radiation from Fluid Loaded Rectangular Plates" M.I.T. TR 71476-1, 1969

10 A.G.WEBSTER, "Acoustical Impedance and the Thory of Horns and of the Phonograph", Procedings of the National Academy of Sciences,pp 275-282, 1919

11 O.K.MAWARDI, "Generalized Solutions of Webster's Horn Theory", J.A.S.A.,21,4,323-330,1949

12 R. SINCLAIR, "The Effects of Manufacturing Tolerances and Driver to Horn Mismatch on Exponential Horn Performance ", A.E.S. Preprint 1893, 1982

13 V.SALMON, "A New Family of Horns " J.A.S.A.,17,212-218, 1946

14 NOBORU KYOUNO, SHINICHI SAKAI, and SHIGERU MORITA "Acoustic Radiation of a Horn Loudspeaker by the Finite Element Method - Acoustic Characteristics of a Horn Loudspeaker with an Elastic Diaphragm-", A.E.S. Preprint 1756, 1981

15 H.F. OLSON, Elements of Acoustical Engineering, Van Nostrand, Princeton, N.J., 1947

16 D.B. KEELE Jr, "An Efficiency Constant Comparison between Low-Frequency Horns and Direct-Radiators", A.E.S. Preprint 1127 ,1976

17 B.HOWZE and C.HENRICKSEN "A High Efficiency, One-Decade Midrange Loudspeaker", A.E.S. Preprint 1848 ,NOV 1981.

18 G.PENKOV, "Power and Real Signals in an Audio System",J.A.E.S., 31:6,p 423 (1983).

19 S.H.LINKWITZ, "Active Crossover Networks for Non Coincident Drivers". J.A.E.S. jan/feb 1976.

20 LIPSHITZ S. P. and VANDERKOOY J.,"A Family of Linear-Phase Crossover Networks of High Slope Derived by Time Delay", J.A.E.S. 31:1/2, p.2 (1983).

21 P.M. MORSE and K.U.INGARD, "Theoretical Acoustics", Mc Graw-Hill, N.Y., 1968

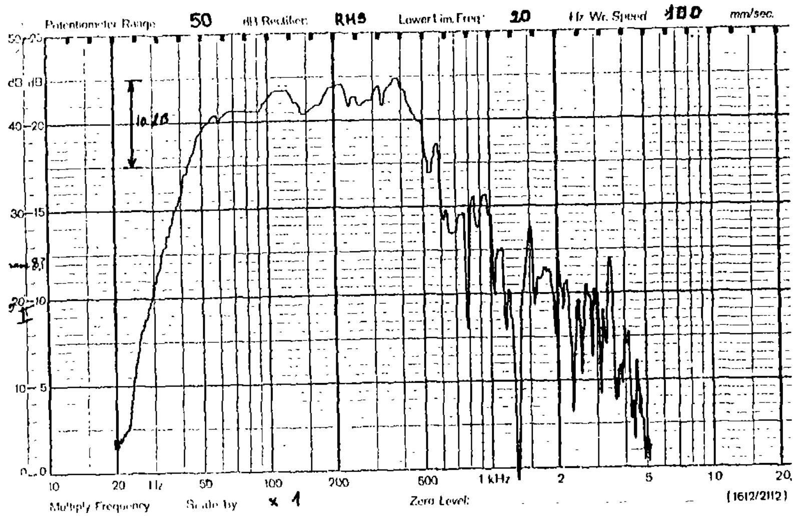


Fig 6 : nearfield response of a single bass element (unwarbled sinewave)

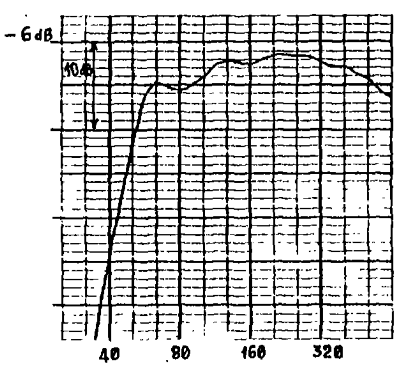


Fig 7

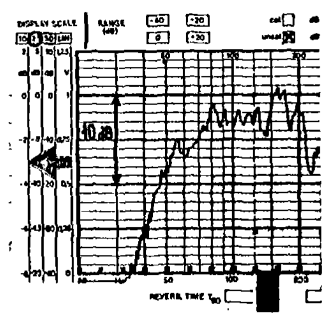


Fig 8

Fig 7: Computed power sensitivity, single bass element, half space

Fig 8: On axis farfield response, single bass element, half space, accidents over 150 Hz due to the reflecting surface.

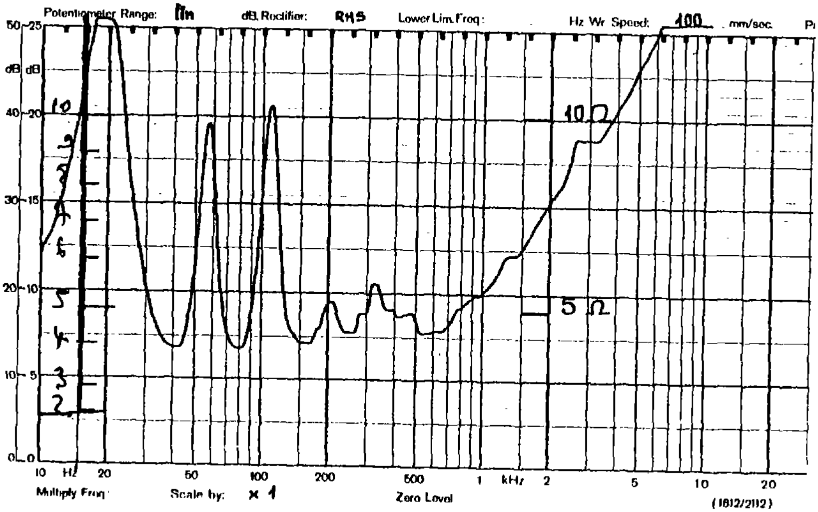


Fig 9 : Measured impedance curve, single bass element, full space (large anechoic room). Note agreement with Fig 8.

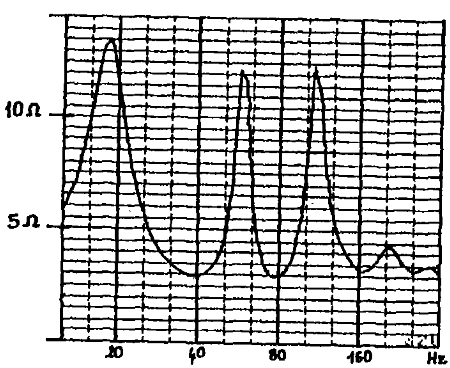


Fig 10 : Computed impedance curve, single bass element, full space.

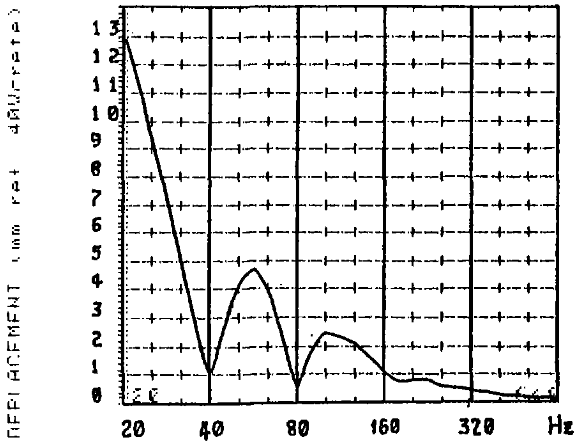


Fig 11 : Peak diaphragm displacement for bass element with 40 Volts peak on input terminals. Single cabinet, half space. Note well controlled behaviour over 30 Hz, usefull notches at 40 Hz and 80 Hz are respectively due to port and horn.

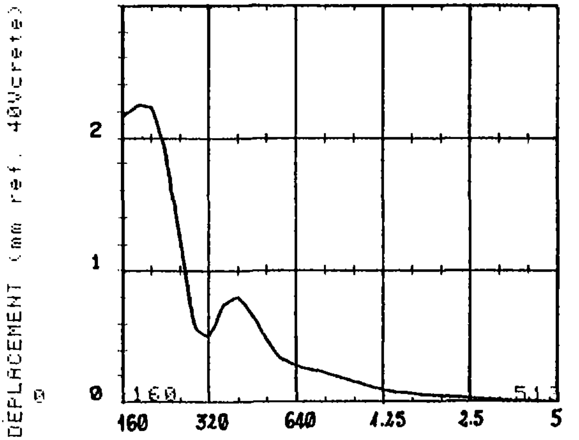


Fig 12 : Peak diaphragm displacement on midrange element with 40 Volts at input terminals. Notch at 320 Hz due to horn allowing -6 dB crossover points from 250 Hz up .

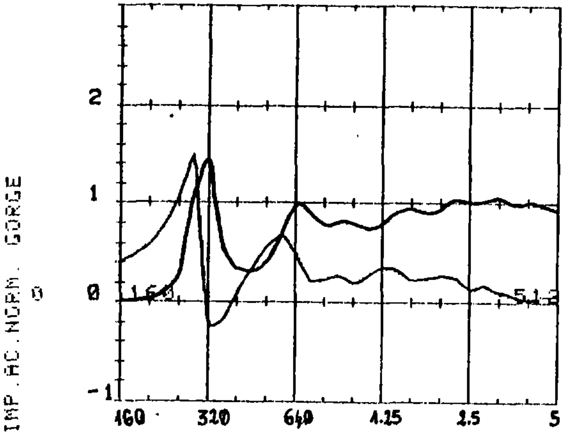


Fig 13 : normalised throat acoustic impedance (resistance and reactance) for the unbaffled midrange horn. Single unit.

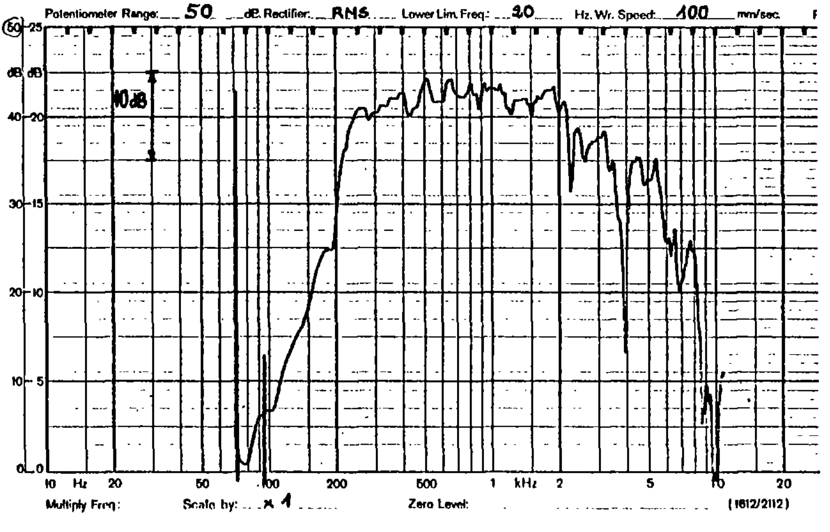


Fig 14 : detailed farfield on axis response,single midrange element,(unwarbled sine wave). Zero level is 55 dB SPL at 4 meters with 4 Volts RMS on 8 Ohm nominal impedance (minimum impedance 9 Ohms) yielding a reference sensitivity of 107 dB/1 W/1 m.

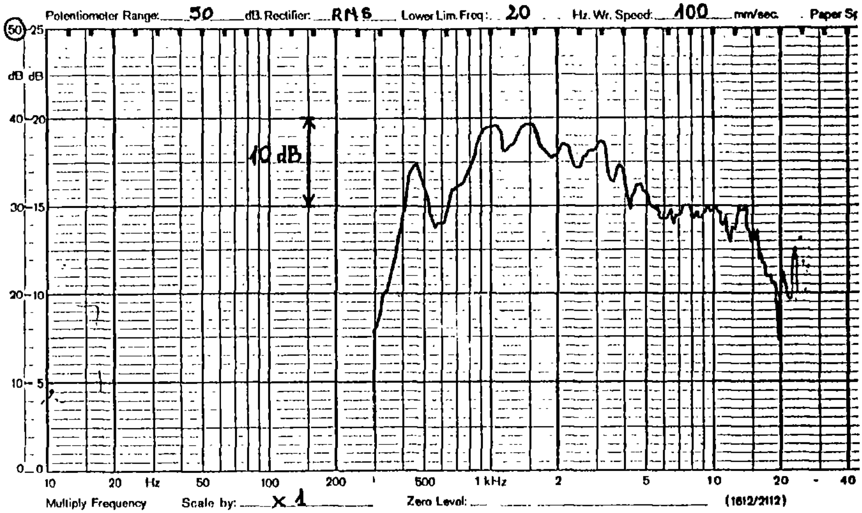


Fig 15 : Unequalised on axis response of unbaffled treble element (unwarbled sine wave)

Fig 16 : a) same as Fig 15, different measurement conditions.
b) same as a) with power sensitivity compensation (constant directivity effect)

Fig 16 a)

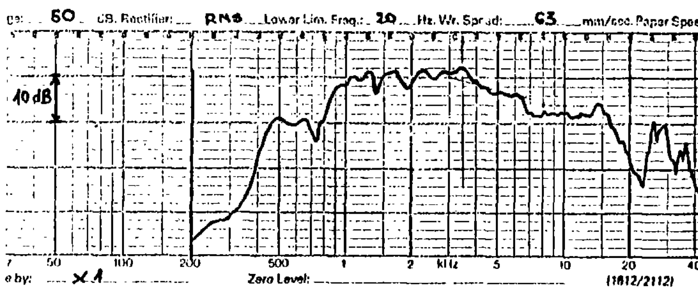
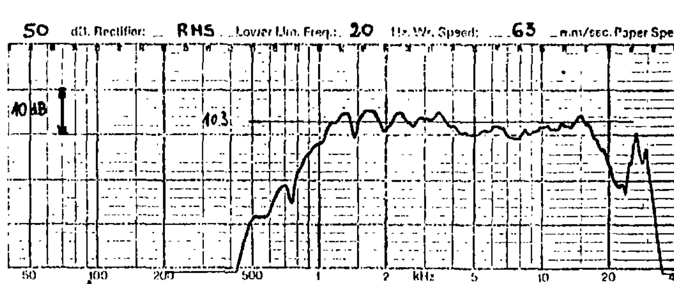
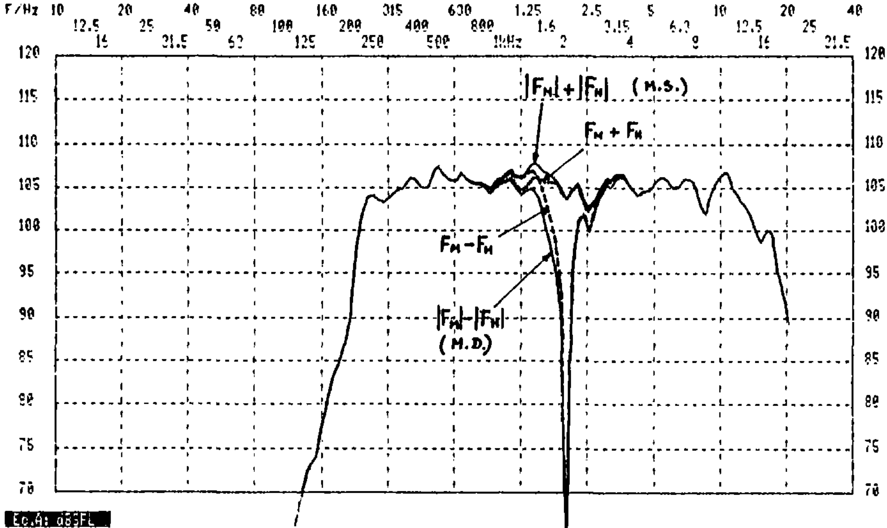


Fig 16 b)

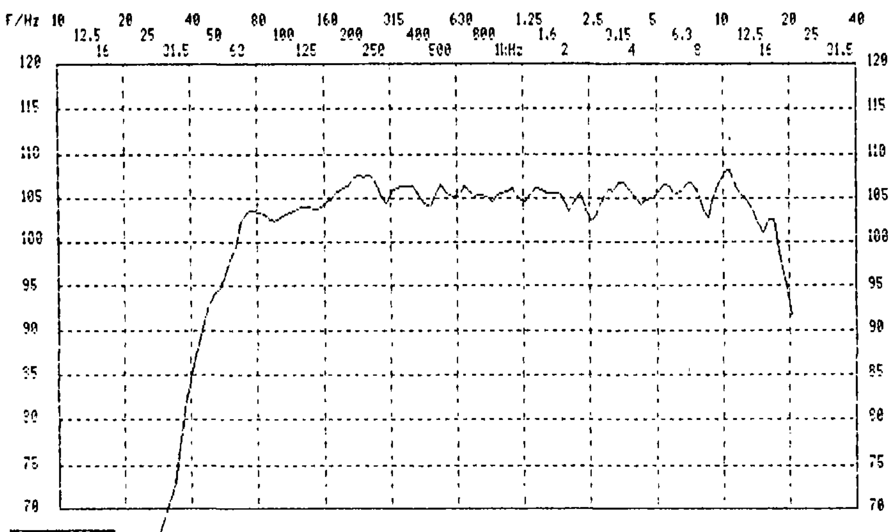




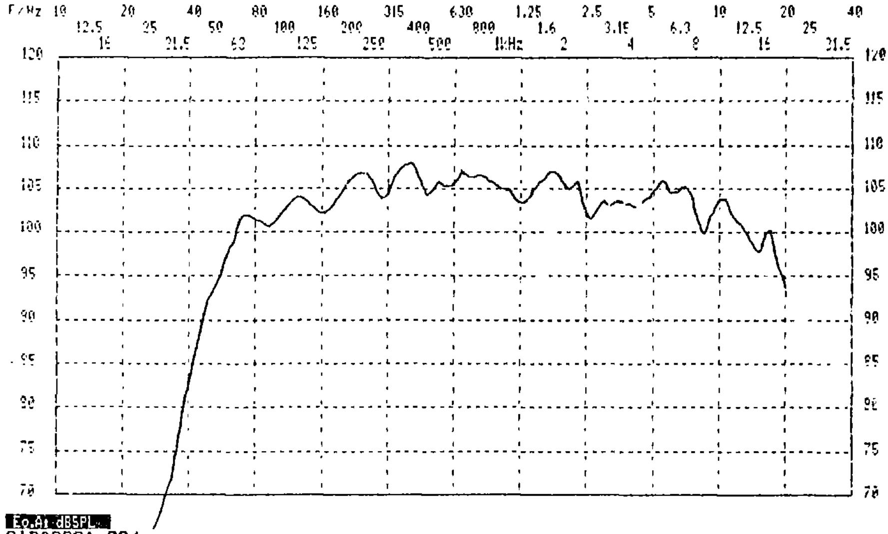
Filtre passif S.I. - SIBMFC02 & SIHMFC12

Fig 17 : MS, MD, addition and subtraction on midrange and treble elements (Note the narrow MIB).

Fig 18 : Simulated anechoic farfield on axis response (full space), single cabinet in two way active configuration. Refer to Fig 6 through 8 for LF performance in multiples and half space.



SIAC2V03



SIPASS01.304

Fig 19 : Simulated anechoic farfield (full space) on axis response for a single cabinet in passive configuration (Note the differences with Fig 18 due to adverse effects of the low pass filter on all the LF spectrum)

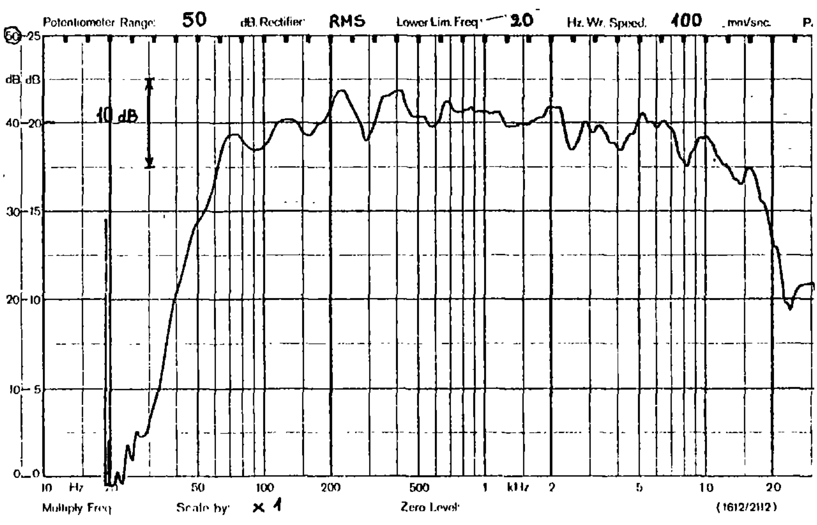


Fig 20 : Single cabinet measured at 5 meters in a large anechoic room (unwarbled sine wave), passive configuration. Note agreement with Fig 19. Refer to Fig 18 for active configuration.

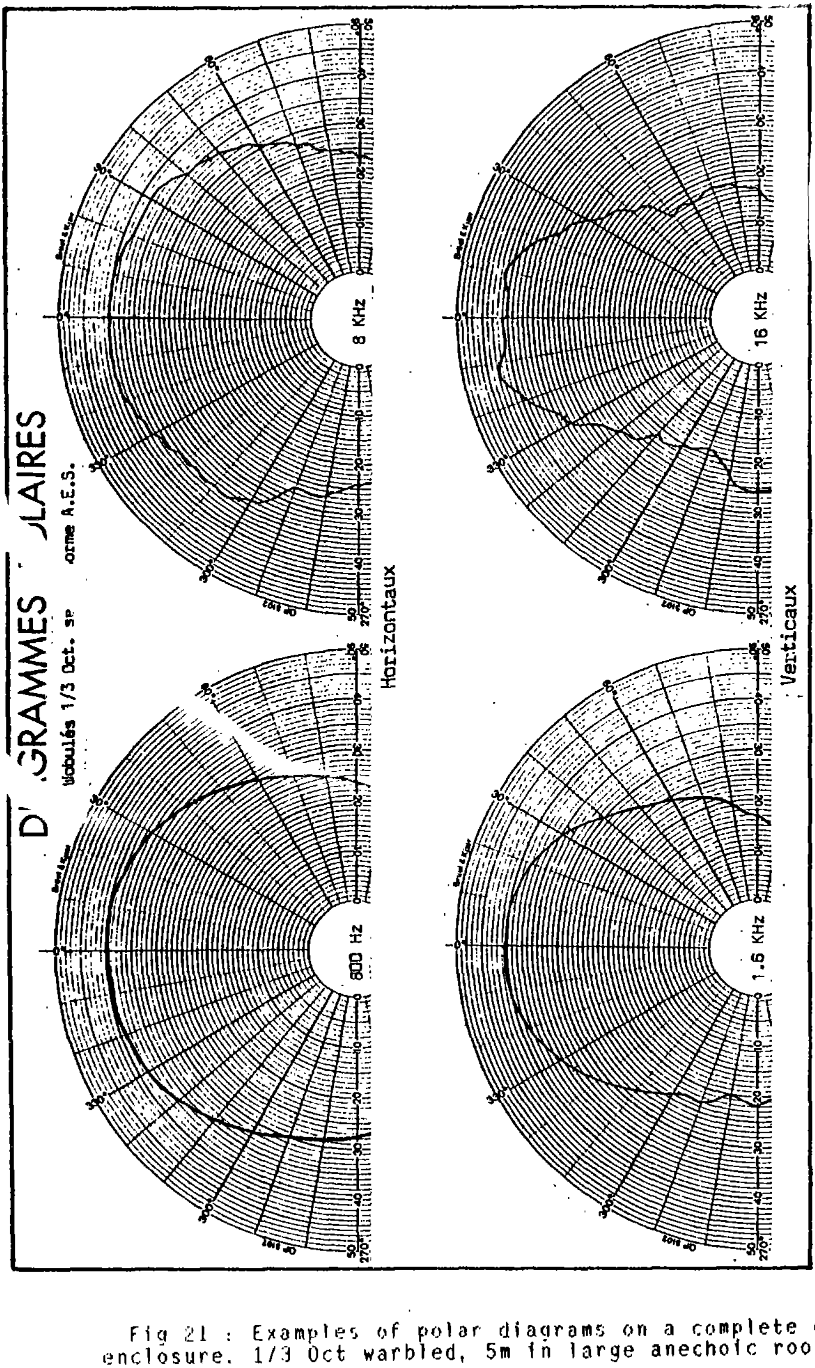
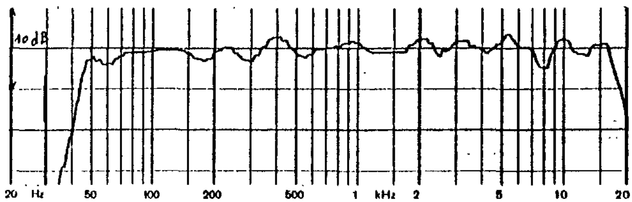
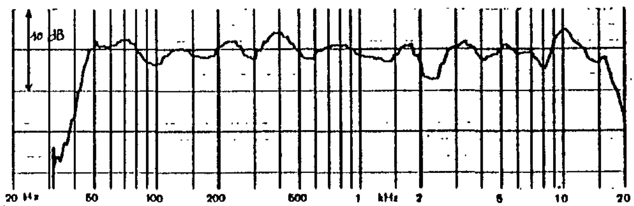


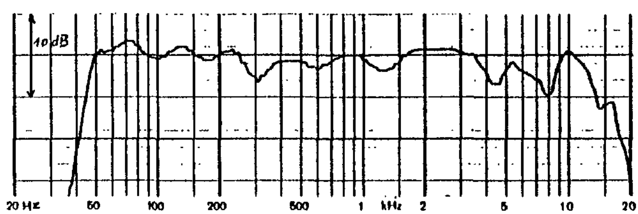
Fig 21 : Examples of polar diagrams on a complete operational enclosure. 1/3 Oct warbled, 5m in large anechoic room.

REFERENCE DANS L'AXE

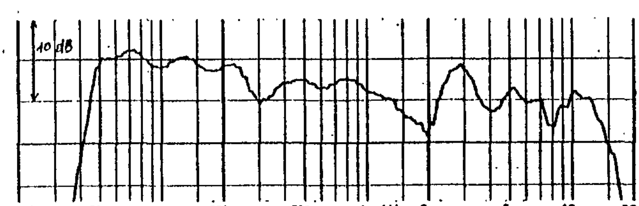




15 Degrés Horizontal



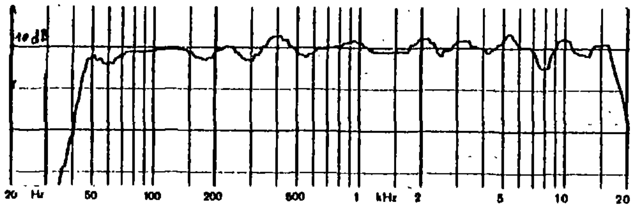
30 Degrés Horizontal

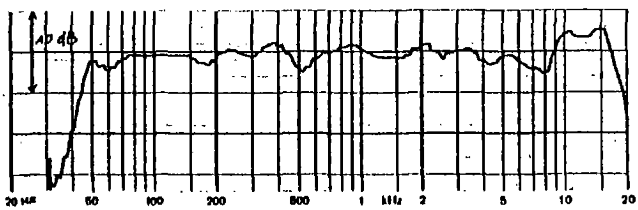


45 Degrés Horizontal

Fig 22 : Off axis response , refer to notes on Fig 23.

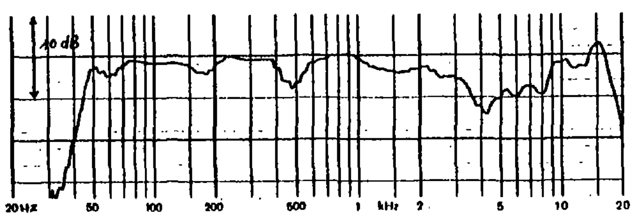
REFERENCE DANS L'AXE





10 Degrés

Vertical



20 Degrés

Vertical

Fig 23 : Off axis response. As in Fig 22 these are measured in a large anechoic room at 5 meters with a 1/3 octave warbled tone. The system is equalised with a standard 1/3 octave unit to provide an on axis reference (AES recommended practice). Passive configuration, active configuration yields better performance in crossover areas.

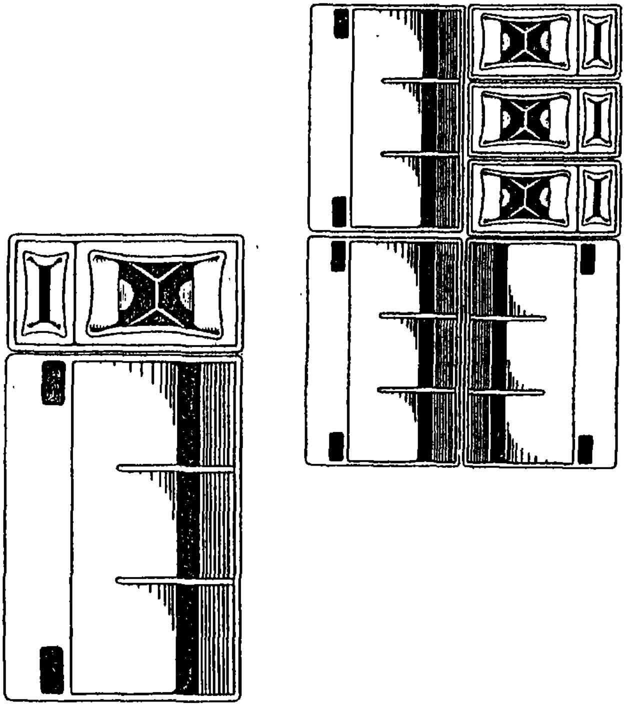


Fig 24 : General aspect for a single system and a stack of three units. Note the reason for the unorthodox relative placement of mid and treble elements.