DESIGN ASPECTS OF A RE-ENTRANT HORN SYSTEM SUITABLE FOR HAZARDOUS AREA USE

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1 INTRODUCTION

The loudspeaker commonly employed for the purpose of signalling in hazardous areas such as those with an explosive atmosphere is a re-entrant horn sounder, and these contain design features that pose challenges to acoustic design.

Historically, the types of signals used to alert a fire were tonal and the sounders were only optimised to be efficient at narrow frequency bands. Modern standards such as EN 54-24 [3] are increasingly requesting that sounders be capable of reproducing speech for voice announcements. To satisfy this requirement, the frequency response of the horn sounder must be extended to encompass the upper speech frequency range which increases intelligibility. Making the transition from a horn sounder that is efficient over a narrow frequency range to a loudspeaker that covers a wide band of frequencies is not trivial and additional design features are needed.

Loudspeakers and sounders could be applied to a number of different environments which will affect the final performance of the system and therefore the focus of this paper is on the engineered aspects of the device rather than the performance in the final installation. The paper will describe at component level the main features of an emergency horn loudspeaker system and the development processes concerned with improving its acoustic performance to meet modern fire voice alarm standards.

2 EMERGENCY USE RE-ENTRANT HORNS

The re-entrant horn design offers the principal advantage of being efficient over an extended frequency range but in a small package size. The initial section of the horn from the throat is enclosed by a folded back intermediate expansion, and finally folded forwards to produce the final flare terminating at the mouth. The physical length of the entire flare section is therefore roughly reduced to a third of its effective length. As with most horns, the purpose is to create an acoustic transformer to increase the radiation efficiency of the driver at lower frequencies and to control directivity.
The efficiency of the horn is further increased due to the diaphragm area being greater than the throat area, producing a compression driver. A typical horn loudspeaker is illustrated below.

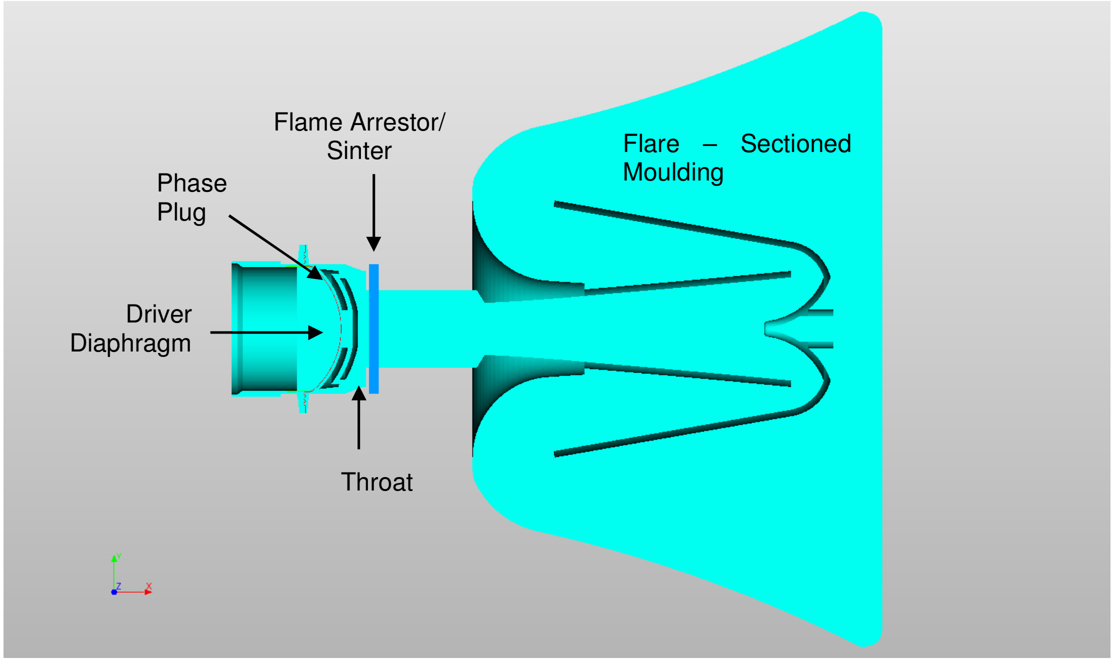


Figure 2-1. Sectioned view of re-entrant horn system

3 DESIGN BRIEF AND CONSTRAINTS:

3.1 The Changing Needs in Emergency Signalling

The signals that are reproduced by sounders are mainly tonal and horns have been designed for maximum efficiency over a narrow bandwidth typically extending to around 4kHz. A "tones table" is usually defined as part of the product specification and the sounder is developed to meet these signalling needs.
The developments of standards such as in [3] have resulted in heightened requirements for sounders. The requirement for frequency response has been stipulated to allow reproduction of voice announcements with a good level of intelligibility. The "tramlines" defining the frequency response limits are shown in **A-1**.

3.2 The Design Challenge

The initial product upon which further developments were based was a sounder that had performance traits meaning that it did not comply with the frequency response requirements outlined in [3]. A severe dip in the frequency response above 4kHz encompassing a large bandwidth meant that this was effectively the upper cut-off frequency for all practical purposes. The tasks were summarised below:

- Meet EN 54-24 frequency response requirement
- Maintain general output level (avoid significant performance loss)
- Avoid changes to outer flare, cannot change the overall package size (length or mouth diameter)

Modifying an existing device as opposed to a ground up design project meant that some design options were unavailable. This situation in loudspeaker development is common because not only can component designs be re-used, returning cost savings on tooling etc., but design risks are reduced.

4 DEVELOPMENT APPROACH

As part of an engineering project, effectively in validation phase, different aspects of the product required attention at different times and it was convenient to divide the loudspeaker into distinct subsystems. This both simplifies and complicates matters because although each component is treated separately, they ultimately need to operate as a coupled system and therefore the overall performance needs to be periodically checked during the development process.
References [1] and [2] provide comprehensive insight relating to the physics of horn loudspeakers and the requirements for drive units and provide some practical direction on non-ideal geometries.

Some basic information regarding the horn could be derived analytically for common horn profiles such as conical and exponential types, but owing to the specific shapes of the components, the approach for developing the loudspeaker was to use the finite and boundary element methods (or FEM/ BEM). These would allow more general shape permutations to be analysed, targeting best performance as agreed with the customer, and then relate these to the real scenario through physical prototypes. The value of detailed physical models is that if done carefully, they offer the benefits of speed, cost and better insight into the operation of the loudspeaker.
All finite element models and measurements were performed with the horn mouth presented flush with an infinite baffle (or practical alternative) and a single axial microphone.

5 SIMULATING THE LOUDSPEAKER SUBSYSTEMS

5.1 Flare Model

An earlier horn version was subject to a number of design changes unrelated to acoustics. The original material was heavy and the fabrication process expensive and this was addressed by making the flare an injection moulded plastic part of constant thickness. A half-section comparison of the two systems is pictured in **A-2**.

To develop an understanding of the performance difference between the two horns, a finite element simulation model was created based initially upon a rigid dome piston excited with unit force.
The resulting frequency response comparison is in **A-3**. The response is far from flat, as is expected because of standing waves in the horn due to the impedance mismatch seen at the horn mouth resulting in wiggles which are exacerbated by discontinuities in the flare expansion caused by the “folding” approximations.
In terms of the geometry difference between the variable and constant wall thickness flares, there was little acoustical variation and the effect of the larger area in the middle return flare section which was an artifact of making the walls a constant thickness, was subtle.

The change of area in the horn’s flare was thought to be approximately exponential. Physical measurements on the prototype have been performed to identify the total flare length and cross section area at the throat and mouth of the horn. Based on calculations from analytical formulae presented in [1] it was subsequently possible to calculate the flare rate (m) and cut-off frequency (f_c) of the device

$$\mathrm{m} = \frac{1}{\text{Flare Length (m)}} \ln\left(\frac{\mathrm{S(Mouth)}m^2}{\mathrm{S(Throat)}m^2}\right) = \frac{1}{0.36} \ln\left(\frac{3.02e^{-2}}{2.01e^{-4}}\right) = 14 \qquad \text{eq. 1}$$

which gives

$$f_c = \frac{mc}{4\pi} = \frac{14 \text{ x } 343}{4\pi} = 380\text{Hz} \qquad \text{eq. 2}$$

The lower cut-off frequency in the FEM model was higher than predicted in the analytical exponential horn calculations. Although the throat and mouth dimensions were easily measurable, the length and flare constant were more difficult to establish because of the folding and this was a probable source of error. This confirmed that a modeling scheme accounting more accurately for the physical attributes of the system was needed.

The flares were also made from different materials. Beranek[1] discusses the effect of using different materials for the flare and suggests that in some cases, resonance effects can be produced where the walls vibrate mechanically. In order to test this, the above simulation was driven both with an infinitely rigid horn (uncoupled) and horns with mechanical properties prescribed to the flare walls. The results shown in **A-4** suggest that in the cases presented, the mechanical vibrations caused by soundwaves transferring energy into the flare structure did not significantly affect the frequency response.

The models were subsequently developed to include the driver electro-mechanical parameters and surrounding acoustical regions more completely.

5.2 Driver Model

The driver was comprised of a dome shaped diaphragm of fabric impregnated with phenolic resin which increased its mechanical stiffness providing structural stability at high output levels and elevated temperatures. This is pictured in **A-5**. Dynamic material properties of the diaphragm were estimated by a process of model updating which was possible because the other components in the moving assembly were made from materials whose properties were known and the boundary conditions were relatively simple.

The results in **A-7** indicate reasonable agreement between the model pictured in **A-6** and the measured driver with the distinct major diaphragm resonance clearly visible at 14kHz. Inspection of the structural deformation in the model suggested that the disagreement at 10.5kHz was due to a slight misalignment between the position of the glue joint/ fixing point between model and prototype but was felt to be a detail not worth pursuing further.

5.3 Driver-horn coupled model: improving the Upper Frequency Limit

Measurements on the prototype have shown that a major feature of the response was the large drop in output between 5 and 10kHz which needed to be addressed for the product to meet the frequency response limits contemplated by the standard EN 54-24. This was confirmed by the finite element analysis on the coupled driver-horn system (red curve in Figure **5.1** below).

Inspection of animations returned by the model indicated that there was a “transverse mode” reducing the acoustical output between 5 and 10kHz. These transverse modes, pictured in **A-10** occur when the acoustic wavelength is equal to the diameter of the airspace in front of the driver and the effect is to cancel the pressure waves entering the throat of the horn.
When such acoustic interferences happen a common solution is that of employing a phase plug, that would break the wave front by guiding sound into paths of different lengths and causing it to recombine in phase at the throat of the horn.

An arbitrary phase plug shape was included in the FEM model as a starting point to understand the mechanism more clearly.

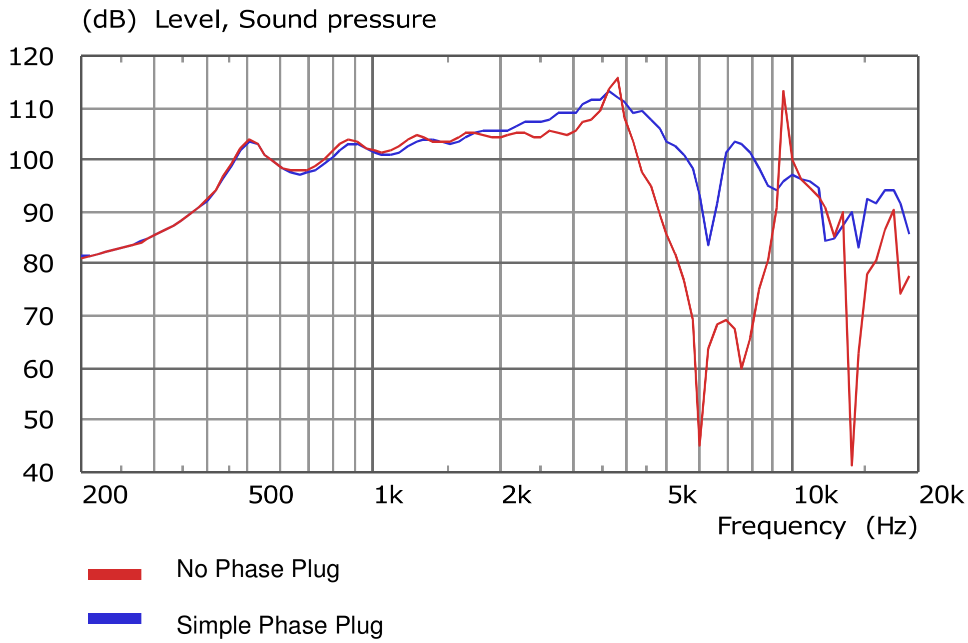


Figure 5-1. Comparison of horn system with and with phase plug in throat cavity

Considerable improvement in the frequency response was evident with the phase plug present and output was generally restored where previously there was little.
The frequency response and animated results revealed how the system was effective in attenuating acoustic interferences, although a distinct improvement was not yet obtained.
On the basis that output was being cancelled within the throat cavity and would therefore not be transmitted into the horn, the focus towards improving the frequency response to meet EN 54-24 shifted to redesigning the geometry at the throat.

6 PARAMETRIC MODEL TO IMPROVE THROAT DESIGN

In compression drivers, a common configuration is to have a reversed diaphragm and converging slits producing a coherent wavefront. This configuration was simulated and is pictured in **A-8** with predicted performance in **A-9**, clearly showing a good frequency response extending well beyond 10kHz and exceeding the EN 54-24 requirements.
In the present case, this is not possible due to restrictions on the gas volume contained within the driver (behind the sinter, discussed later in the paper). A smaller gas volume lowers the temperature rise over the sinter thus minimizing the explosion pressure behind it.
An alternative, but less intuitive method is to configure a forward facing diaphragm with phase plug arrangement aimed at producing similarly convergent wavefronts at the horn throat.

After a number of investigative variations of phase plug design, a 2 part system was conceived. To distinguish the components, the part closest to the diaphragm was called the phase plug and the outer part was referred to as the “dodger”. These parts are illustrated in **A-12**. This system allowed the acoustic wave generated by the motion of the entire diaphragm to propagate along 2 specific routes, avoiding setting up the cross mode and recombining constructively in the throat.

The figures showing the internal wave-field in the throat of the loudspeaker in **A-10** and **A-11** illustrate the 2 part phase plug in contrast to a more conventional design. To complete the work, some optimization was carried out to seek out improvements from small variations on the same 2 part theme.

A parametric model of a number of throat geometric variables was developed based on 2 axisymmetric acoustic boundary elements. Such models have the advantages of flexibility and computational efficiency, which are both very important when geometry optimisation is concerned. A sketch of the parametric model is illustrated in **A-13**.

The first interior boundary element represented the inner throat geometry up to where the transducer interfaces with the horn which is terminated by the second boundary element which modeled the exterior domain as radiation from an infinite baffle. This terminating condition was preferred over a boundary condition mimicking for example, an infinite pipe because there was better scope to reproduce these results experimentally for comparison.

The diagram illustrates the parameters that were varied during the development of the system. The resulting response variations are in **A-14**. A final design was selected and a full loudspeaker system simulation confirmed the benefit, these results are in **A-15**.

This new system proposal was then physically prototyped yielding vast performance improvement shown in the experimental results below.

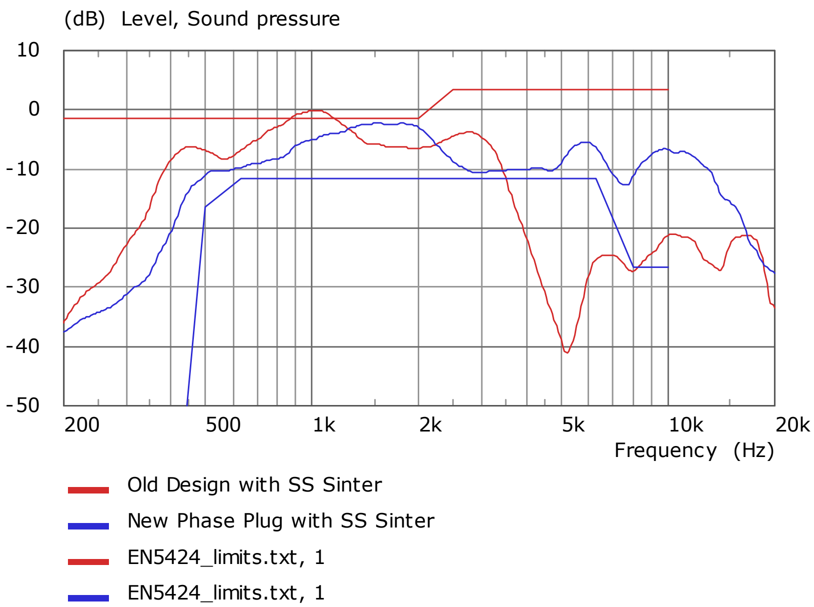


Figure 6-1. Results after implementing phase plug from parametric model

Figure 6-1 shows that the upper frequency range has been extended by the redesign and brings the performance within the tramlines defined by EN 54-24.

7 EXPERIMENTS VARYING SINTER LOCATION

7.1 Sinter and Acoustic Performance

The flame arrestor, or sinter is a device that is positioned in the throat of the horn and is conventionally fabricated by heat fusing small bronze beads or steel flakes to produce a disk shaped component. Its purpose is to contain sources of internal combustion from the transducer by presenting an acoustically permeable barrier between the air surrounding the transducer and that in the exterior atmosphere, which is potentially explosive.

The conflicting requirement for a device to provide an air barrier and yet an acoustical path results in a reduction in the acoustical output by up to 20dB as shown in Figure 7-1.

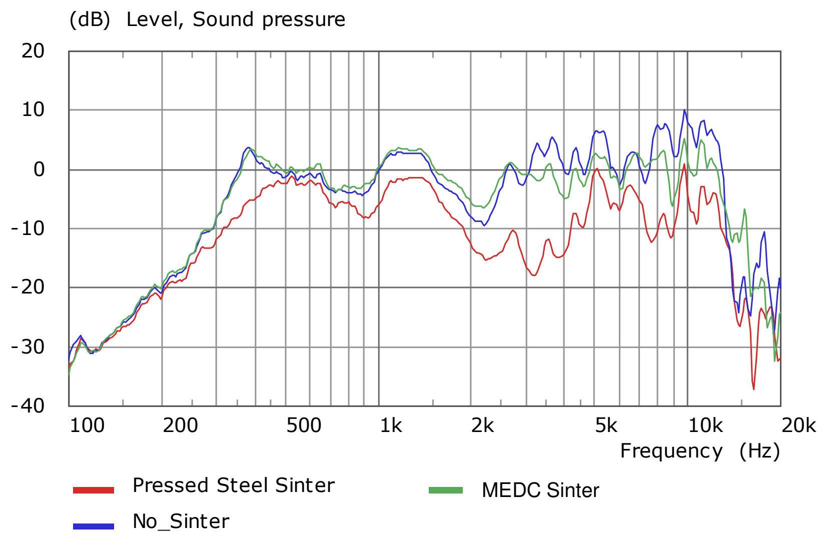


Figure 7-1. Effect of sinters on acoustical response

The sinter composition is a key factor in determining the overall acoustic performance of the loudspeaker and developments in flame arrestor technology, currently covered by NDA by the client have resulted in significant improvements. The frequency response comparison shows that a steel sinter exhibits much deteriorated performance versus the new MEDC sinter which has a much more benign effect. These devices are pictured in below.
A-16 shows electrical impedance measurements of the loudspeaker in 3 states. The bare driver has an impedance response typical of most moving coil transducers with a peak at the fundamental resonance. Adding the horn results in the appearance of additional features due to acoustic loading of the diaphragm. At the cut-off frequency of the horn, the first of the standing waves discussed in **5.1** affects the mobility of the diaphragm resulting in a spike in the impedance curve. The maximum impedance at the coupled resonance of the driver when the horn is added is damped. Given that the acoustic output increased at the frequency of this coupled resonance, around 1.1kHz, this loss was believed to be the result of increased radiation efficiency affecting the mechanical system as added damping.
Adding the sinter further increased the loss as seen by the mechanical system and particularly the peak at the horn cut-off frequency

Although part of a coupled system, the drive unit was found to have a strong influence over the frequency region around its fundamental resonance at 1.1kHz. This was evidenced by additional mechanical damping added to a driver model coupled to the horn system. The damper decreased the magnitude of the region between 600Hz and 2kHz as the results in **A-17** show. The addition of a sinter similarly affected this region and as such, was believed to more generally contribute to the drivers mechanics as a viscous damper.

The sinter also affected regions of the frequency response associated with standing waves in the horn. The magnitude of the peak at where the horn begins to radiate effectively was damped and shifted up in frequency by the presence of the pressed steel flake sinter in particular and the shape of the upper part of the frequency response was also altered.

7.2 Effect of Sinter Location in Throat

Practical experience iteratively prototyping the horn system suggested that the distance of the sinter from the transducer had an effect on the low frequency performance. This was investigated experimentally because it was desirable to minimize the air volume behind the sinter for non-acoustic reasons, and also preserve the low frequency extension.

A straight section of horn was added at the throat to provide a test region for the sinter positions, effectively increasing the horn length by 30mm and reducing the lower cut-off frequency slightly. The 30mm section comprised an assembly of 6 stacked plastic rings, one of which would house the sinter whereas the others were used as spacers – these could be interchanged, effectively moving the sinter further along the throat. This was achieved at fixed intervals (stage 1 is closest to the driver, stage 4 farthest) and the effect on the frequency response was observed. This assembly is illustrated in **A-18**.

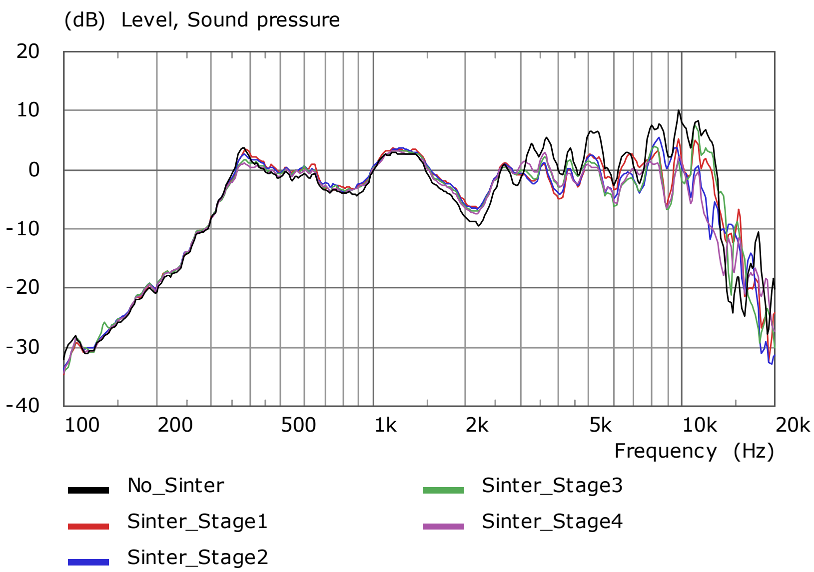


Figure 7-2. Measured MEDC sinter shifted 6mm steps in horn throat

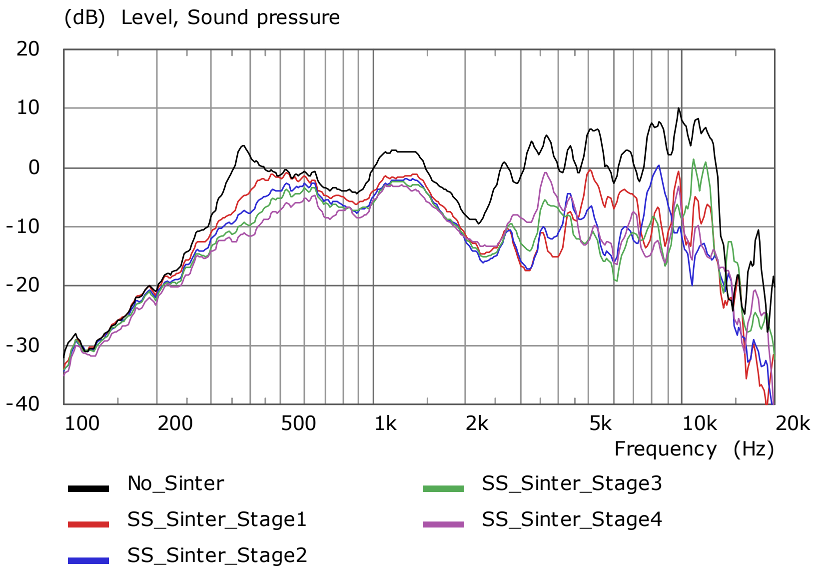


Figure 7-3. Measured pressed steel flake sinter shifted 6mm steps in horn throat

The measurements showed that the MEDC sinter had less impact on the acoustic output than the pressed steel flake type. The pressed steel flake sinter more clearly showed effects resulting from positional variations.

The effect of moving the sinter away from the driver and further along the horn affected the region around the horn cut-off frequency at ~400Hz and the upper horn response shape, but to a lesser degree the region between 800Hz and 2kHz, a region under greater influence of the driver. In the case of the pressed steel flake sinter, the increase in cut off frequency suggested that the sinter was effectively shortening the horn. At higher frequencies, the sinter positioning resulted in dips in the response over a range of frequencies. This is most evident when looking at the extreme cases in Figure **7.3** where the sinter at stage 1 produces higher output between 5 – 6kHz, but at stage 4 results in a null, the opposite being true at other frequencies. Porous absorbers are most effective when positioned in regions of high particle velocity. In the horn, these velocity maxima shifts as frequency is increased, therefore potentially reducing the output when they coincide with the sinter. These effects were generally not as obvious with the MEDC sinter.

7.2.1 Prototyping and Air Leaks

It is important to maintain an airtight seal in the throat section of the horn when prototyping as air leaks alter the performance significantly and make subtle performance changes resulting from optimisation steps difficult to see. In a production unit, the airspaces surrounding the sinter and throat cavity are flooded with resin-like compound, ensuring an excellent seal and consistent performance in the field. In the “virtual” prototyping process using finite element models, a perfect seal is also assumed. The graph below illustrates the effect, where a slight loss of low frequency performance is observed due to a small air leak in the throat cavity.

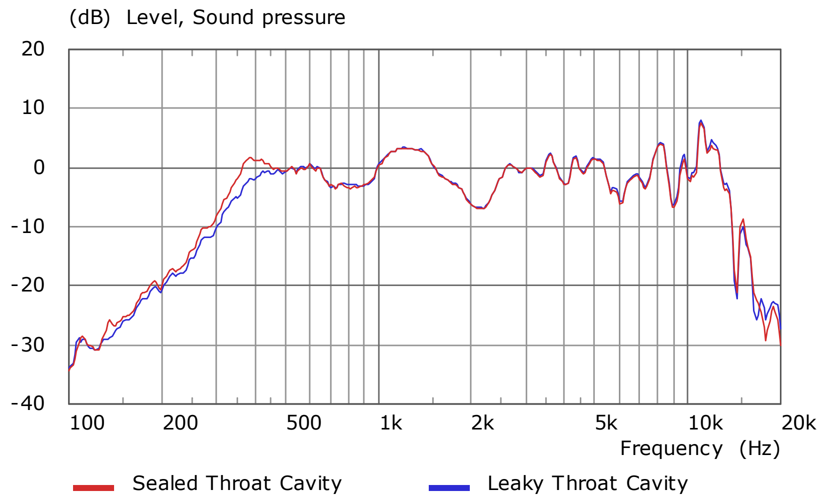


Figure 7-4. Effect of air leak in horn throat

Physical prototyping also presents challenges where many parts are fabricated using non production processes, for example 3D printing where filament deposition techniques create a slightly porous structure as opposed to an injection moulded part. Gaskets and O-rings may also leak if the mating faces are not sealed.

8 CONCLUSIONS

This paper was based on a case study to illustrate some of the design factors involved in emergency horn systems. Using modern simulation methods, a “next generation” horn loudspeaker was proposed as an adaptation of an earlier sounder which comfortably met the EN 54-24 frequency response requirements enabling it to be used in voice alarm applications.

Modifications to the throat detail meant that:

- Acoustic cancellations were prevented resulting in more extended upper frequency response.
- A parametric model was developed allowing a large number of throat geometry variations to be tested. Specifically, a 2 part phase plug was developed to prevent acoustic cancellations in the throat.
- Because the response extension was achieved with changes to the throat only, the product variation from a sounder to a fuller range loudspeaker was confined to a reduced number of components.
- The finite and boundary element methods provide solid guidance in the development of this class of loudspeaker.

The sinter, as a mandatory device in the system was investigated:

- Comparing the horn without a sinter in place and with 2 sinter variants included revealed that the new MEDC sinter had far less negative impact on the acoustic performance than the pressed steel flake sinter, but served equally well as a flame arrestor.
- The MEDC sinter exhibited only small losses even at positions further along the horn throat.
- Positioning the sinter closer to the driver reduced the loss at low frequency. It is possible that placing the sinter in regions of high volume velocity, which is dependent on the acoustic wavelength in the horn, increased the acoustic loss at those frequencies. This

could be examined by extending the finite element model to include a representation of the sinter as a volume absorbency applied to a region of acoustic finite elements.

- Based on the above, the MEDC sinter succeeded the pressed steel flake design.

9 FURTHER WORK

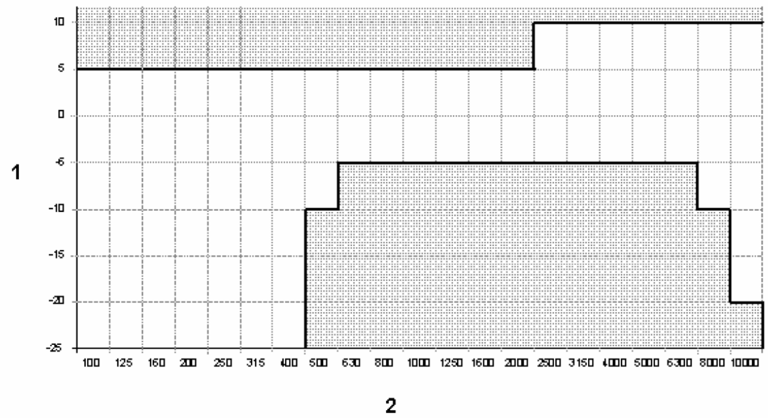
- Investigate sources of error in the horn modelling, particularly the effect of small geometric variations that produce large effects in the acoustic response.
- Transfer of energy from sound travelling through the horn into the structural flare components resulting in mechanical vibrations was not observed in the simulation. It is possible however that vibration could be mechanically transmitted from the driver itself, particularly as the motor is a source. The simulations could not have captured this because the relevant driver details were not included in the models presented.
- Investigate possible sources of viscous loss in the driver, particularly when local air temperature is elevated.
- Review experimental procedures to seek out better ways of sealing the prototypes.

10 REFERENCES

1. L.L.Beranek, Acoustics, Acoustical Society of America, 268-284, 1993.
2. J.Borwick, Loudspeaker and Headphone Handbook, Third Edition, Chapter 1, 2001.
3. European Standard EN 54-24. ‘Fire Detection and Alarm Systems – Part 24: Components of voice alarm systems - Loudspeakers, 2007.
4. M.E. Delany, E.N. Bazley, ‘Acoustical properties of fibrous absorbent materials’, Applied Acoustics. 1970.
5. J.Dinsdale, ‘Horn Loudspeaker Design-2’, Wireless World, May 1974
6. Colloms, M., Darlington, P. (2005) *High Performance Loudspeakers* (6^{th} ed.) John Wiley & Sons.

1. APPENDIX

1-1 EN 54-24 Frequency Response Limits

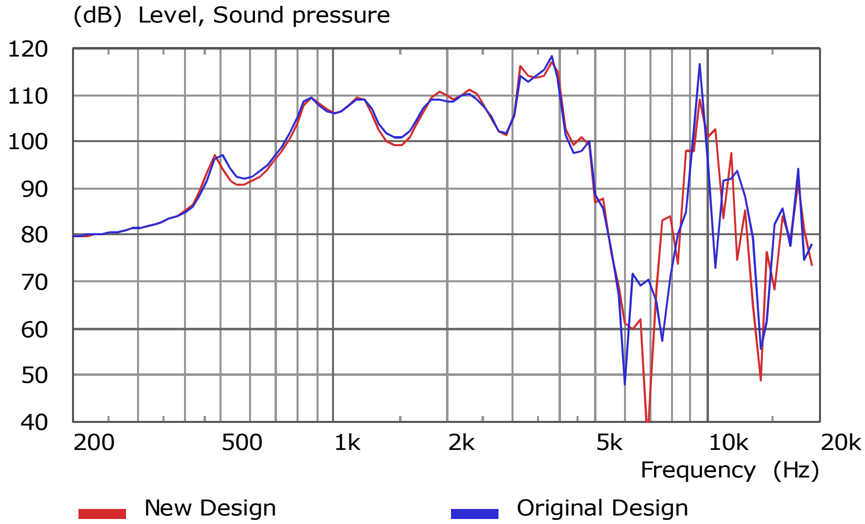


Key

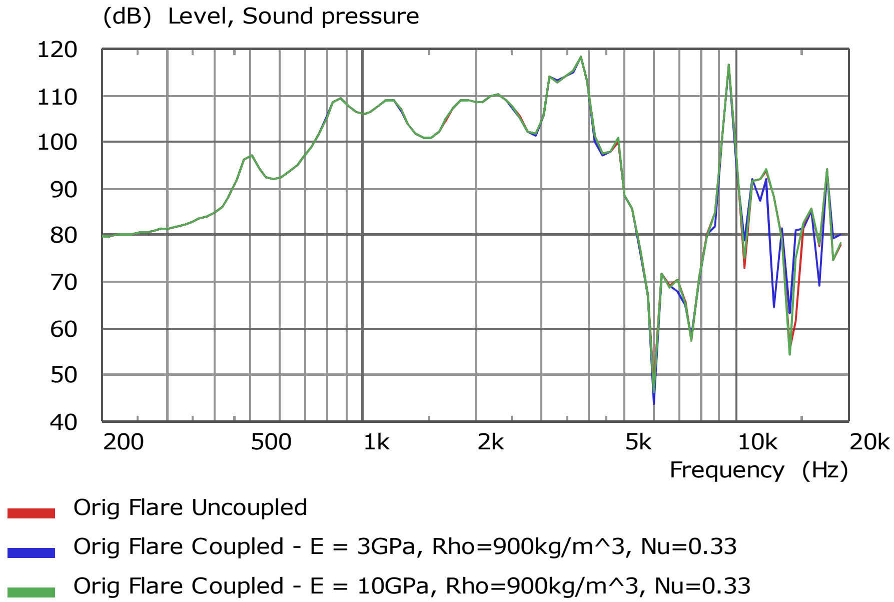
1 relative level [dB]
2 1/3 octave band centre frequency [Hz]

A-2 Flare Re-Design

A1-3 Old Horn versus New Horn Design

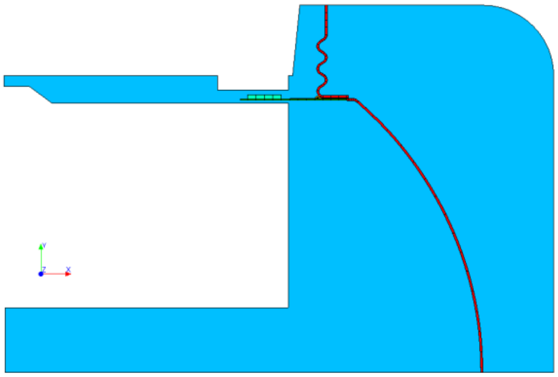


1-4 Comparing Uncoupled Horns with Coupled (with material prop's)

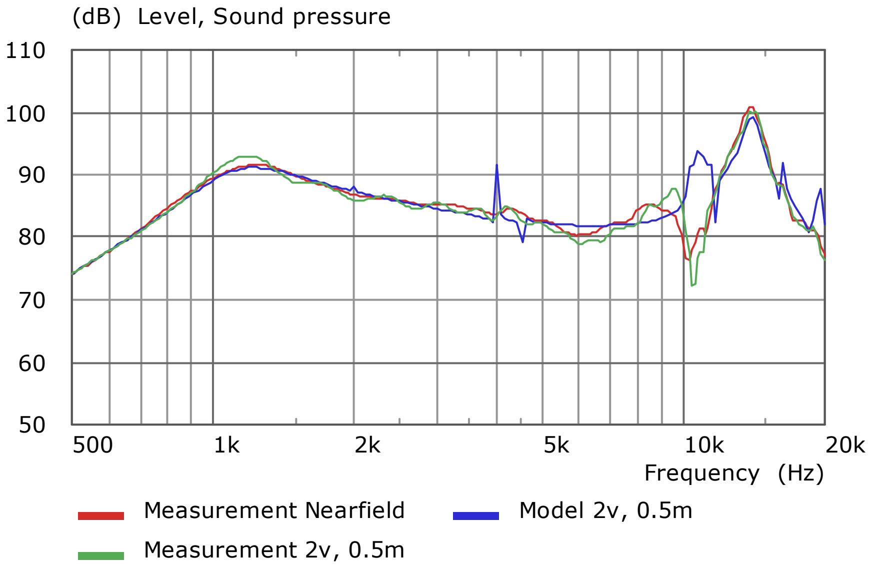


1-5 Photo of drive unit

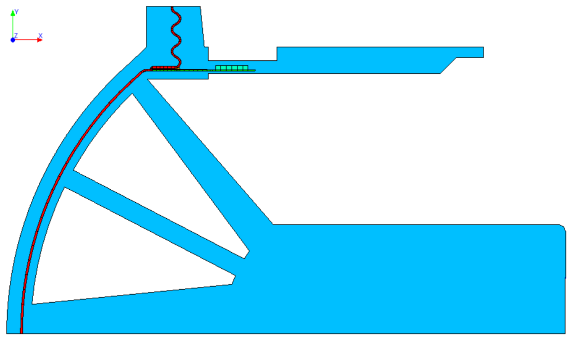
1-6. Simulation model of drive unit including air regions



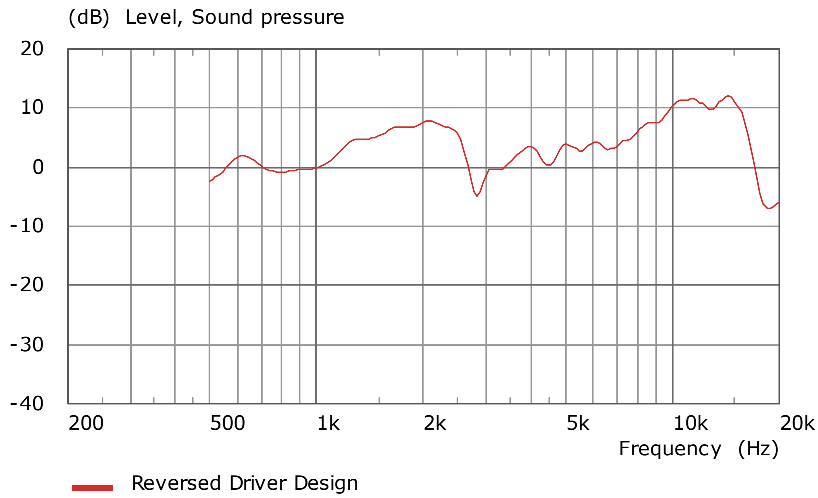
A-7. Measured drive units versus modeled drive unit



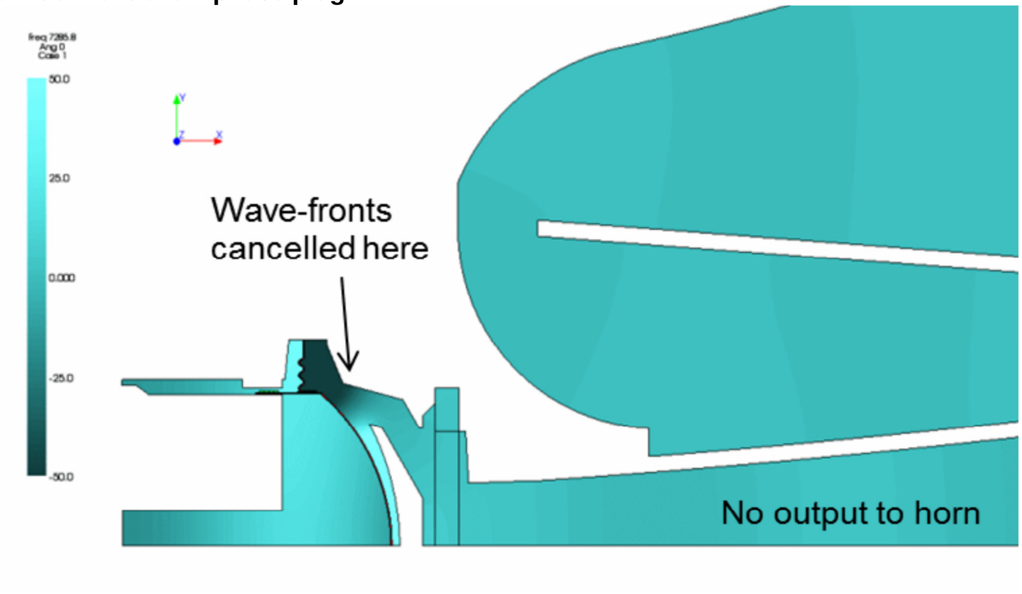
A-8. Reversed driver design showing phase plug detail



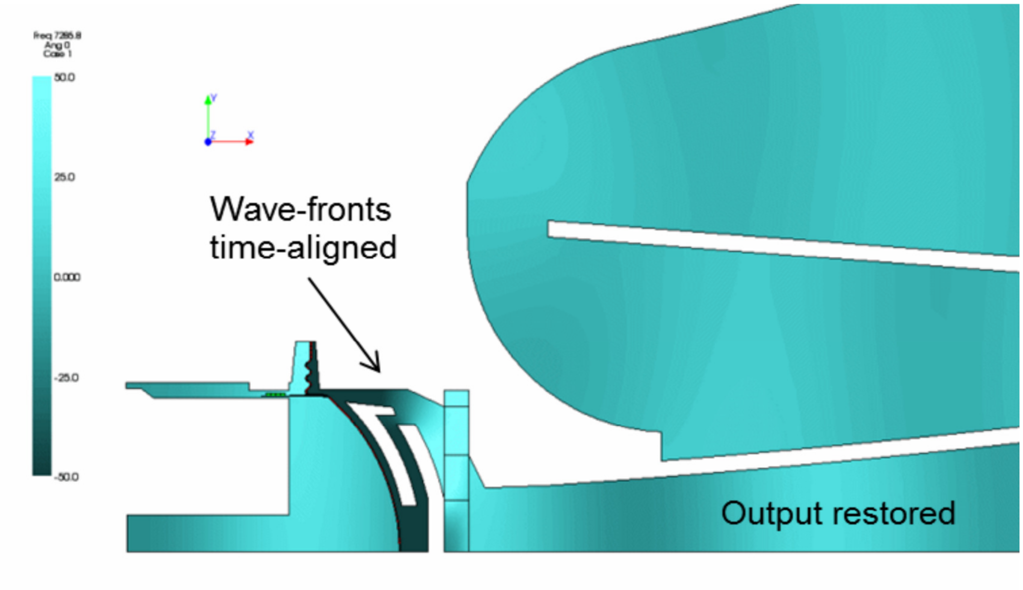
A-9 Reversed driver design when attached to horn



A-10 "Conventional" phase plug

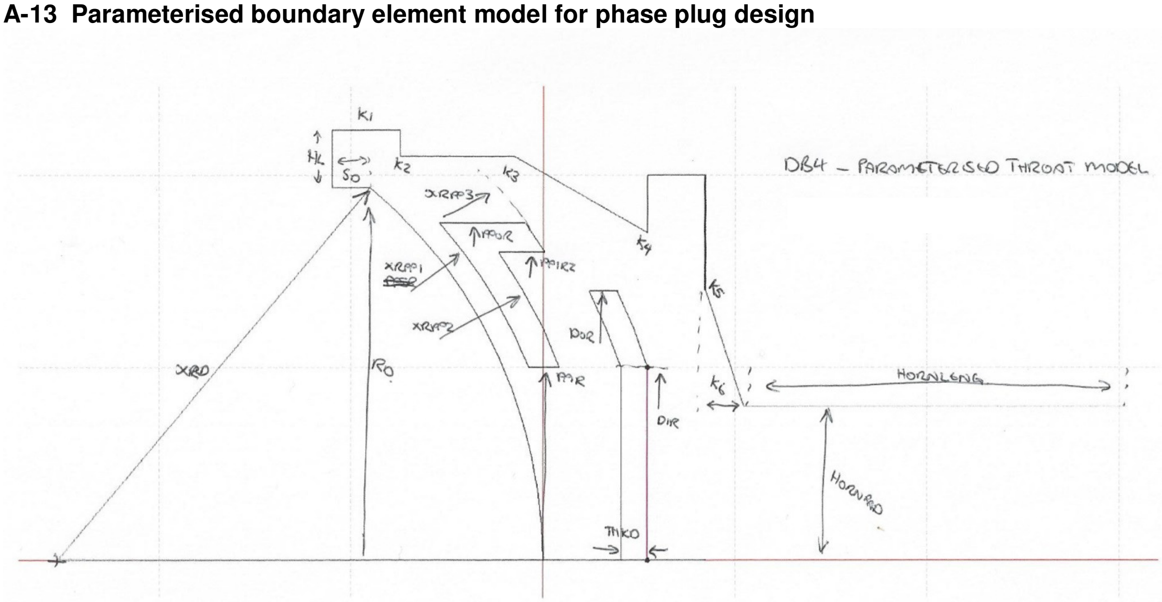


A-11 2 Part phase plug design

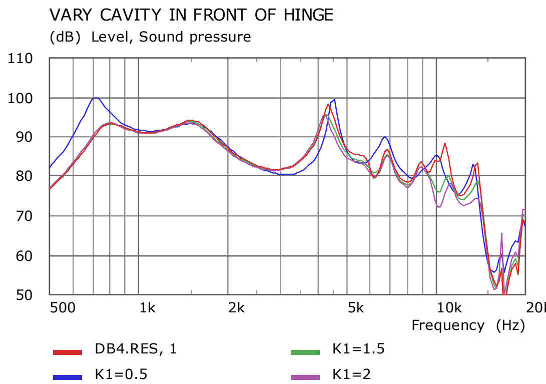


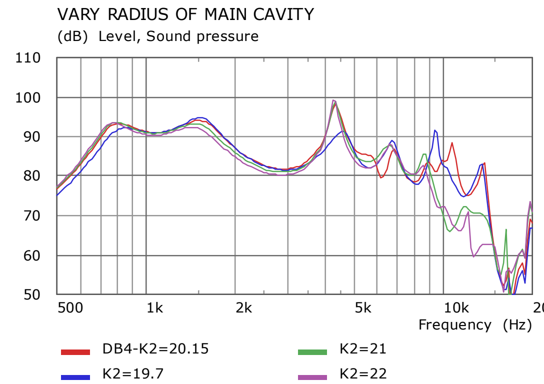
A-12 Photo of phase plug and dodger arrangement

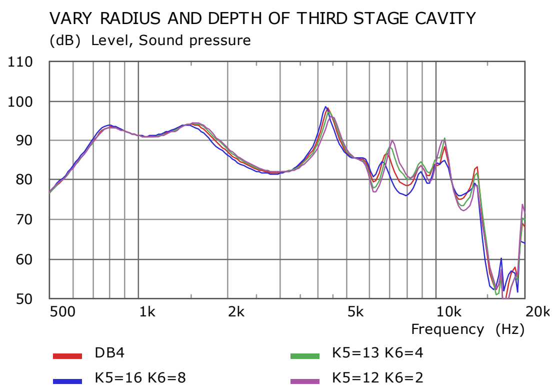
A-13 Parameterised boundary element model for phase plug design

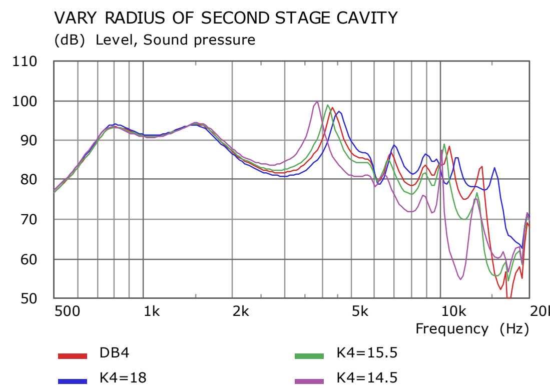


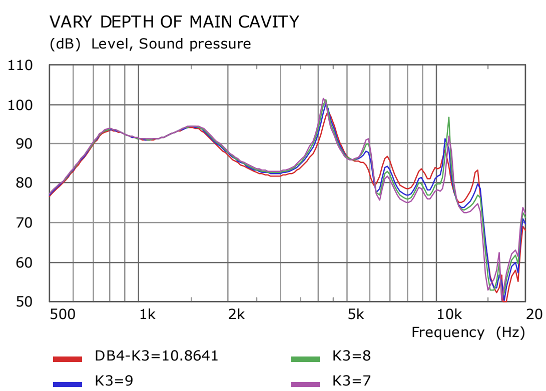
A-14 Results of parameterised model runs

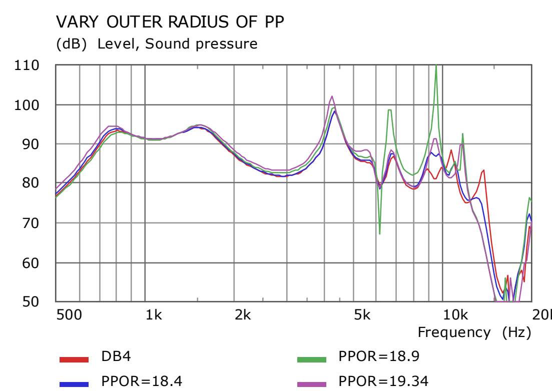


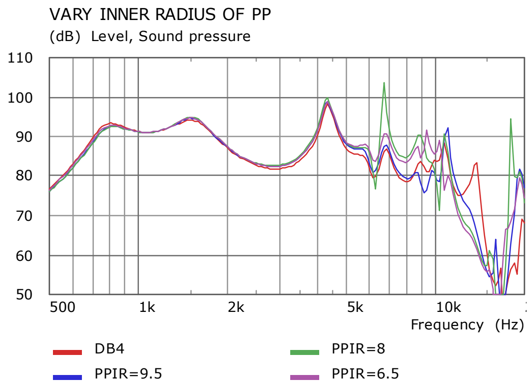











VARY INNER RADIUS OF PP
(dB) Level, Sound pressure
110
100
90
80
70
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1k
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Frequency (Hz)
DB4
PPIR=9.5
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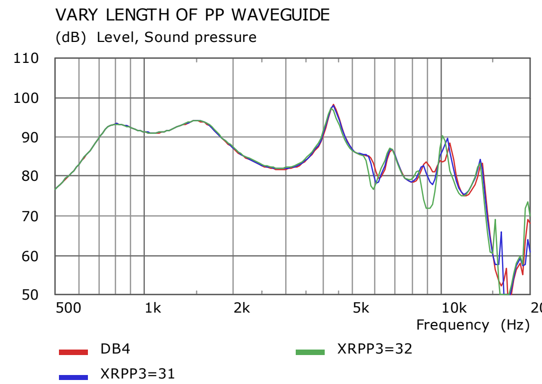
PPIR=6.5


VARY LENGTH OF PP WAVEGUIDE
(dB) Level, Sound pressure
110
100
90
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500
1k
2k
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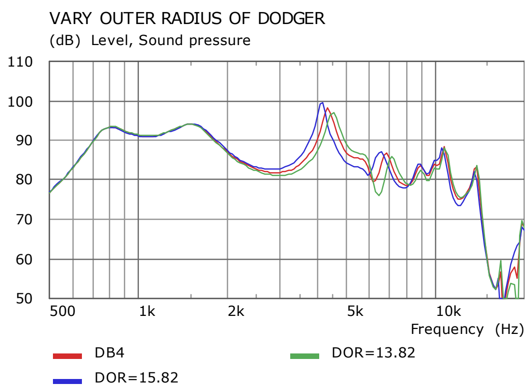
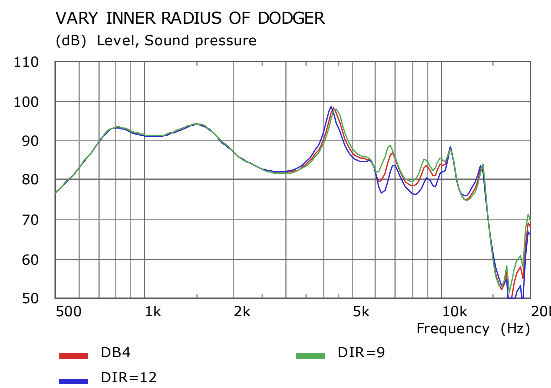
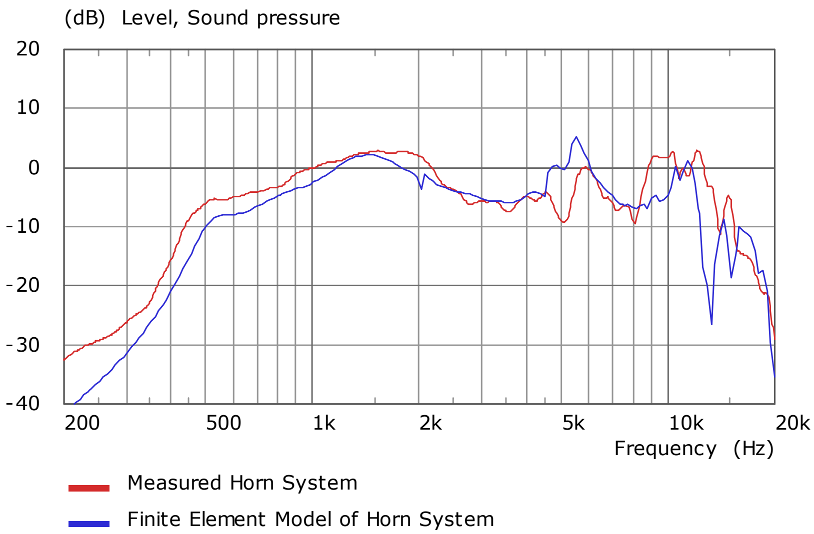
10k
20k
Frequency (Hz)
DB4
XRPP3=31
XRPP3=32

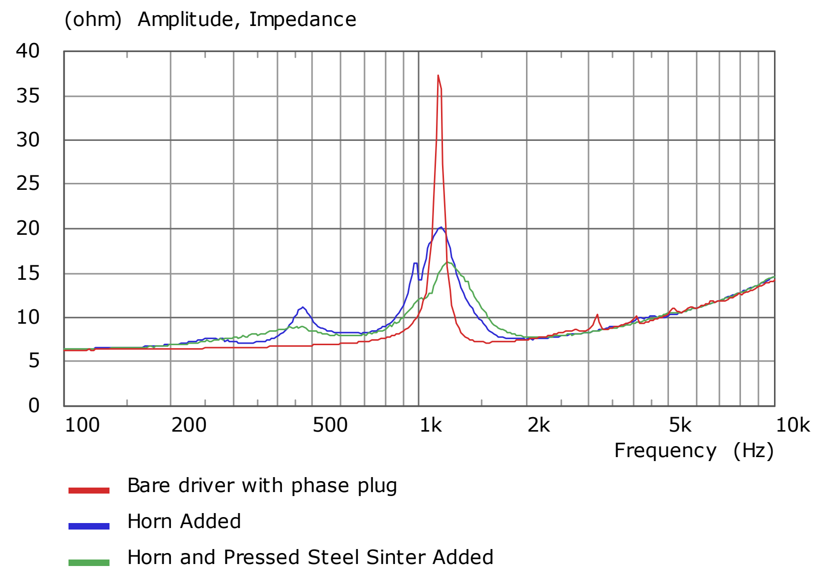

VARY OUTER RADIUS OF DODGER
(dB) Level, Sound pressure
110
100
90
80
70
60
50
500
1k
2k
5k
10k
Frequency (Hz)
DB4
DOR=15.82
DOR=13.82


VARY INNER RADIUS OF DODGER
(dB) Level, Sound pressure
110
100
90
80
70
60
50
500
1k
2k
5k
10k
20k
Frequency (Hz)
DB4
DIR=12
DIR=9

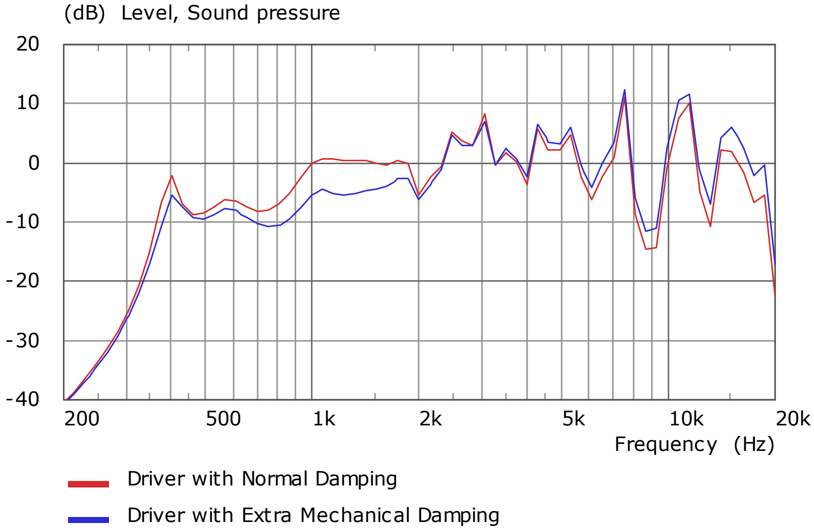
A-15 Full Loudspeaker Model vs Measured



A-16 Electrical Impedance Measurement



A-17 Loudspeaker model - driver with and without added mechanical damping



A-18 6mm Horn Extension Spacers for Sinter Relocating