# Accuracy of Fully Coupled Loudspeaker Simulation Using COMSOL

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**Abstract:** Loudspeaker simulation is used to inform the designer as to the performance of a design. In recent years the Finite Element Method (FEM) has been used to model the mechanical and acoustical attributes of a loudspeaker with varying success. This paper shows how a model that incorporates the magnetic, electromagnetic, mechanical and acoustical domains performs. These domains will be coupled where necessary and the model will focus on accuracy while trying to remain simple. The results will be directly compared with measured data showing the performance of the model. The results of the simulation show good correlation with the measurements both in the frequency domain and for the electrical impedance.

**Keywords:** Acoustics, Loudspeaker, Impedance, Sound Pressure Level.

## 1. Introduction

Simulation is a critical part in the development of a loudspeaker. It enables the designer to estimate the performance of the loudspeaker before the first prototype is assembled. Having a simulation method that returns reliable results will enable the designer to speed up the development and reduce the number of prototypes required. The final aim of simulation is to have a model which replicates reality to such an extent that the prototyping stage of development is effectively streamlined.

Advanced simulation methods, like the Finite Element Method (FEM) have up until recently been exclusively used to show theories for research purposes. This is due to the large amount of computing power that methods like FEM require. The evolution of the desktop computer is now at a point where multiple CPU's and large amounts of memory are available. Companies are now able to invest in simulation technologies to innovate their products. Before large amounts of computing power became available today's designers' preferred simple analytical models due to their speed. Designers understand the limits of these methods and how to use them effectively. With the modern designers taking up methods like FEM, research must be carried out into the limitations of the models.

Currently the best simulation methods use the FEM to model the mechanical and acoustical parts of the loudspeaker. The implementation of this often involves minimal coupling between domains and does not includes electromagnetic interactions. These techniques are often calculated using 'in house' software making the development time for a technique very long.

The method used in this paper is based on the Comsol tutorial model 'loudspeaker driver' [1]. This outlines a basic modeling technique which can be used to build a model of a drive unit. The model however is based on a theoretical drive unit and gives no evidence of its accuracy.

This paper will show the accuracy of a model based on this technique and will show how it can be improved so that the models give results that correlate with reality.

Where the modeling technique differs from that used in the tutorial or where important limitations are found the paper will give details.

## 2. Model Design

This section will outline differences between the tutorial modeling technique and the modified technique used in this paper.

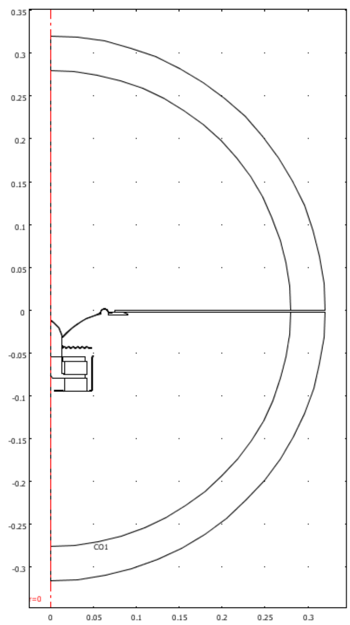


**Figure 1.** Geometry of BRS28-6P

As with the tutorial model given by Comsol the results will be valid for small signals where non-linear integrations are neglected. The modeling is applied to a BRS28-6P loudspeaker unit shown in figure 1. It can be seen here that a PML is employed on the outer boundary to simulate an infinite baffle.

### 2.1. Acoustic Losses

The pressure acoustics application mode according to the COMSOL reference guide models a lossless medium unless a damping term is added. In small gaps laminar flow resistance becomes significant, the pressure acoustics application mode does not model this and therefore this damping must be added. The problem can be seen in figure 2.

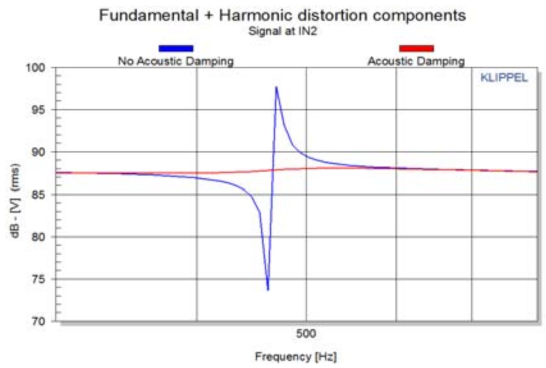


**Figure 2**: Frequency Response of Problem area with and without acoustic damping

Figure 2 also includes the result that is gained after the following method has been used to resolve the problem.

In the model of the BRS28-6P significant damping from the laminar flow resistance occurs along the sides of the voice coil in the magnetic gap. The solution to this problem is to create new subdomains of air either side of the voice coil and apply a damping factor to them in terms of the bulk viscosity.

As there's now damping on either side of the voice coil this will affect the damping of the mass spring system. The graph below shows the difference in the damping around the resonant frequency of the drive unit with and without the method.

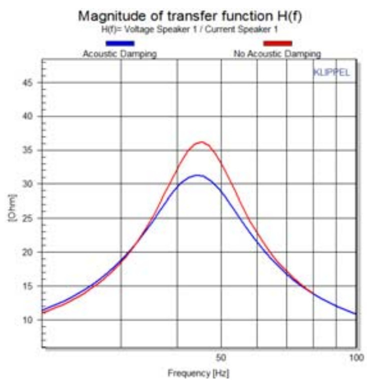


**Figure 3**: Impedance around resonance with and without acoustic damping

This increase in damping should be considered depending on the type of model being described. In models where the suspension is modeled numerically as a resistive force on the voice coil, this damping would have been included in the measurement of $R_{ms.}$ Therefore a modification should be made to the boundaries of the voice coil to ignore the fluid load.

In the full model where the suspension is included this extra damping may be an actual physical effect of the laminar flow resistance. If two plates are considered to be separated by a liquid and the top plate is moving at a constant speed a certain amount of energy will be lost into the liquid as friction [2]. The same case may apply for the coil within the magnetic gap, as this is a dynamic system the loss of energy is seen as damping, this will be frequency dependent.

### 2.2. Surround

In most modern designs the surround if made out of rubber. The surround also has a significant contribution to the response of the drive unit as there will often be up to 5 modes present in the pass band. If the rubber is not modeled correctly then these modes can either cause large variations or not enough variations, both will cause issues for a designer.

To model a rubber correctly the material must first be considered, rubber exhibits large variations in loss factor with respect to frequency. Rubber starts to transition from its rubbery region through to its transition region within the excitation frequencies, present in mid bass driver. This transition causes a large increase in loss factor, it also creates a small increase in the Young's modulus however these changes are small enough to be assumed constant at this stage.

Material parameters of the change in damping with frequency within excitation frequencies of 10-20000Hz are not published. It was therefore set about to find an approximation. This was carried out by modeling a drive unit and finding the modes present in the surround and matching the amplitudes of these modes to a model. This created a table of damping factor vs. frequency data.

This data can be implemented within Comsol by creating a new material in the material library and defining a function of loss factor vs. frequency.

### 2.3. Meshing

There are three stages at which the model is meshed. The first is during the static magnetic mode. This mode is very simple with a simple geometry. The magnets, pole and top plate can easily be meshed by manually defining a fine mesh throughout the magnetic circuit.

The second is the electromagnetic domain. This is a much more complicated mesh and must be considered carefully. The third mesh is for the mechanical and acoustical domains, as these two domains are fully coupled they can share a mesh, although parts of the mesh have to be tailored to the model.

### 2.3.1. Electromagnetic Domain

The primary concern within the electromagnetic domain is that the surface currents on the pole and top plate are modeled correctly. The depth of the surface currents decreases as the frequency increases. This results in an increase in resistance with frequency.

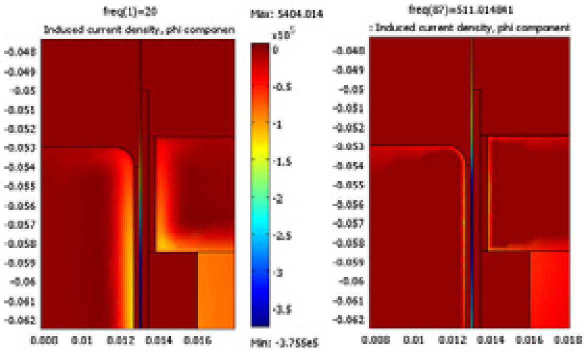

**Figure 4**: Induced Current Density, Left: 20Hz Right: 511Hz

This can be estimated by considering the skin effect [3].

$$\delta = \frac{1}{\sqrt{\pi\mu_0}}\sqrt{\frac{\rho}{\mu_r f}} \qquad (1)$$

where; $\mu_0$ is the magnetic permeability of free space, $\mu_r$ is the relative permeability of the material, $\rho$ is the resistivity of the material ($1/\sigma$) and $\sigma$ is the electrical conductivity.

The skin depth $\delta$ is the decline in current density vs. depth, where the skin depth is the distance it takes for the current to drop to 1/e of its original level. By calculating this at a number of frequencies the skin depth can be compared with the model and an estimation of the mesh size can be made.

| Frequency (Hz) | Calculated Skin Depth (m) | Modeled Skin Depth (m) |
|---|---|---|
| 20 | 5.32E-04 | 1.00E-03 |
| 30 | 4.34E-04 | 7.70E-04 |
| 65 | 2.95E-04 | 5.40E-04 |
| 125 | 2.12E-04 | 3.70E-04 |
| 250 | 1.50E-04 | 2.40E-04 |
| 500 | 1.06E-04 | 1.50E-04 |
| 1000 | 7.52E-05 | 9.00E-05 |
| 2000 | 5.31E-05 | 5.00E-05 |
| 4000 | 3.76E-05 | 3.50E-05 |
| 8000 | 2.66E-05 | 2.50E-05 |

**Table 1**: Calculated and Modelled Skin Depth

The table shows that the skin depth can only be predicted for high frequencies, as at low frequencies the model behaves slightly differently. However for the mesh this is fine as it is the highest frequency of the model that the mesh has to be fitted to. In this case the skin depth at 8000Hz is around $2.5*10^{-5}$, if it then takes one element (2 nodes) to model the change in current, the maximum mesh size will need to be $2.5*10^{-5}$. It is unlikely however that this will provide an accurate result; the maximum element size will be changed around the voice coil, pole and top plate for the final model. The major result from this domain is the block coil impedance which will be directly affected by the surface current density. This result will therefore be compared against the maximum element size.

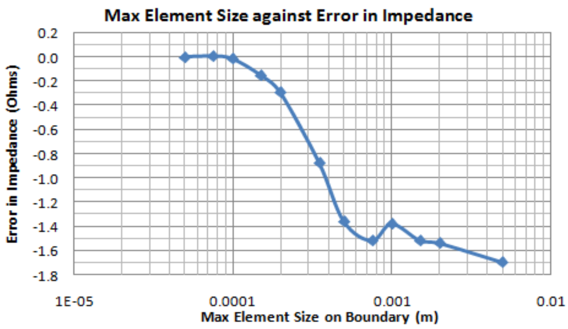


**Figure 5**: Max Element Size against Error in Impedance

Due to limitations in the number of elements that can be solved (60000 elements) the lowest element size that could be reached was 5*10-5. This unfortunately is twice the size of the skin depth, however the results show that the Impedance has converged. In the models a value of 0.00007m will be used as it shows little error and saves on a large number of elements. To keep a good resolution for lower frequencies an element growth rate of 1.2 can be specified to the top plate and pole.

### 2.3.2. Mechanical Acoustical Domain

In the mechanical and acoustical mesh there are two parts that need to be considered, the moving parts that are modeled in the mechanical domain and the acoustic parts. Both of these need different sized meshes to provide a high accuracy model.

The size of the mesh in the air is defined by the maximum frequency that accurate results are required for, in the case of the BRS28-6P this is 10000Hz. Above this frequency the error will increase significantly and it is necessary that the designer understands this limitation. In FEM the points can be determined as nodes, however it is easier to talk in terms of elements as this is how the geometry is meshed. COMSOL has the ability to define the maximum element size in a sub domain or on a boundary; it can also be defined as a global setting. Therefore the mesh needs to be defined in terms of maximum element size.

If two points per wavelength at an arbitrary frequency of 8000Hz is going to be preserved, the maximum size of the mesh has to be 0.02125m.

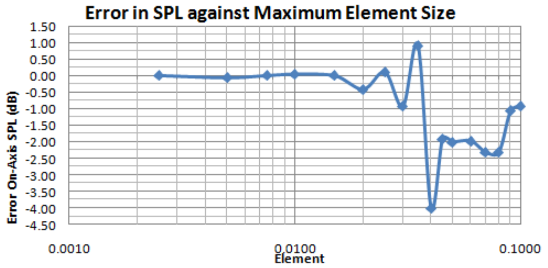


**Figure 6**: Error in SPL against Maximum Element Size

Figure 6 shows the error in SPL against the maximum element size, it shows that when the element size drops below 0.015m the result for the SPL stabilizes. This point is slightly smaller than the predicted point of 0.02125m, however for the simulation the maximum element size will be set at 0.0085m as this is well into the stable region.

The mechanical part of the model concerns mainly the narrow regions. This means that the resolution of the mechanical regions can be set by the global free mesh parameter 'resolution of narrow regions'. To check how the mesh density of these parts affects the results three resolutions were used.

| Resolution of Narrow Regions (Elements) | Number of Elements | SPL On-Axis (dB) |
|---|---|---|
| 1 | 27606 | 87.807 |
| 2 | 52269 | 87.799 |
| 3 | 78491 | 87.799 |

**Table 2**: Mesh Resolution of Mechanical Parts

The results show that this parameter has little effect on the final result of the model. Two frequencies were used, one within the pistonic region and the other within the cones breakup. Both values showed little change, therefore the final model will use a value of 1 for the resolution of narrow regions. This will help to keep the number of elements within the model to a minimum, while preserving accuracy.

## 2.4. Air Load

A drive unit consisting of a piston radiating into an infinite volume of air will have a mass specified by its moving parts and the air load in front and behind the cone. At the typical resonant frequencies of a drive unit, the air motion near the diaphragm acts essentially as a reactive mass, since the real part of the acoustic impedance is much smaller [4].

The mass of the air acting on the cone for a drive unit in an infinite baffle is;

$$M_{air} = 2 * 0.85\rho\pi a^3$$

The mass of the air load is therefore essentially the mass of a sphere of air 1.28 times the radius of the cone. This calculates the mass of air around a piston of radius 61.84mm in an infinite baffle at 1.52 grams. The piston width is the same as the BRS28-6P drive unit.

Using a simplified model that couples the mechanical and acoustical domains, the effects of the air load can be shown on the mechanical resonance of the system. The model will be defined by the drive unit parameters essentially making it a very simplified drive unit and modeled in axis symmetry. The model follows the piston assumption and therefore will be named accordingly.

To correctly model the boundary between the air and the piston the complex pressure must be used. This is critical as the air load is reactive and therefore providing a mass.

Using the model the impedance will be calculated, allowing the resonant frequency of the system to be calculated. The resonant frequency and the compliance can be used to calculate the total moving mass. By calculating this in both a vacuum and normal air density the difference in mass can be found. This difference will be the amount of added mass from the air load.

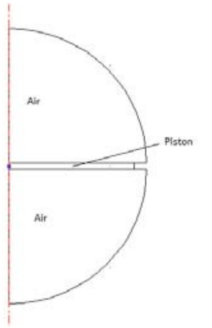


**Figure 7:** Geometry of Piston Model

As it is important to model the air load correctly, a number of models will be run with increasing volumes of air in front of the cone and behind the cone. This will give an idea of how much air must be modeled in order to give a reliable result for the air load on the cone.

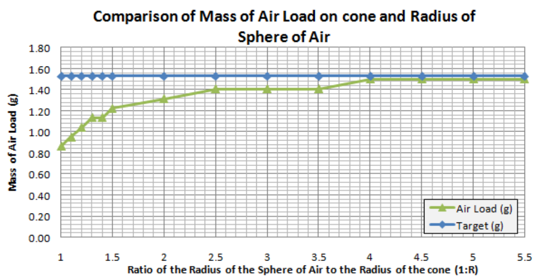


**Figure 8:** Comparison of Mass of Air Load on Cone and Radius of Sphere or Air

The graph above shows the results of 14 models that were run. The x axis shows the size of the radius of the sphere of air in front and behind the cone compared with the radius of the cone. Therefore a value of 1 indicates that the radius of the sphere equals the radius of the cone. As the radius increases it is shown that the mass of the air load increases until it gets to a value that is slightly less than the theoretical one.

The results show that to properly model the mass of the air load on the cone the radius of the sphere containing air in front and behind of the cone must be greater than four times the radius of the cone.

As the model uses the finite element method the mesh size must be investigated as well. The radius of air used in front of the cone is four as this is the smallest size that permits good accuracy.

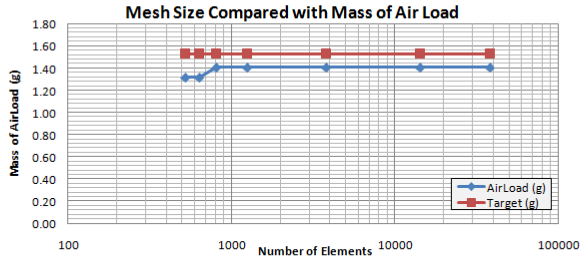


**Figure 9:** Mesh Size compared with Mass of Air Load

This shows that the mesh size has very little effect on the accuracy and therefore need not be considered as the mesh size is limited a lot higher than this to model the high frequencies.

To make sure that the air load is correctly modeled a radius of 4.5 times the piston should be used. This will give results that are very close to the theoretical model given.

## 3. Results

This section will contain both results and some discussion for individual parts. The simulation results will be given for 20Hz to 20kHz. The upper frequency cut off for accurate results is set at 10kHz however minimal error is shown between this and 20kHz.

The measured results were taken from a baffle mounted into an anechoic chamber which approximates an infinite baffle. This enables the measurements to be directly compared to the results without the inclusion of other artifacts.

For the final models all of the material parameters were measured and not matched to the final result. This means that the final results are comparable to what would be given during prototyping before a first off sample was received.

### 3.1. Magnetic Analysis

As large signal nonlinear effects are not being included in the modeling technique, the magnetic analysis can be separated from the rest of the model. Due to the neglect of the large signal nonlinear behavior this value can be considered stable and frequency independent. This means that the value of Bl can be calculated in a static model reducing computational time and computer resources required. The Bl value can then be recorded and included in the final coupled model. The magnets were specified with a remnant flux density from 0.39T to 0.41T. For the first part of the analysis the remnant flux density will be considered to be 0.40T.

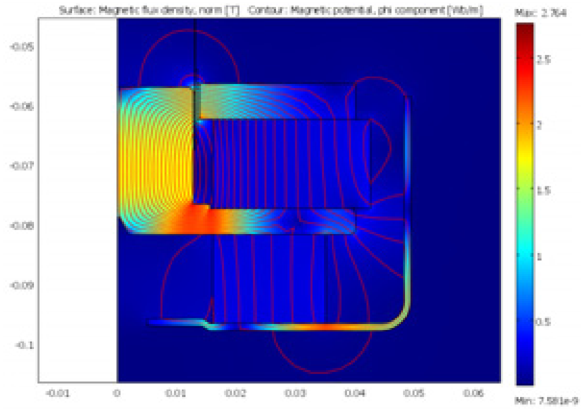


**Figure 10:** Magnetic Flux

Figure 10 shows the magnetic flux density, the red shows places that have a high flux density and the blue show places that have a low flux density. The background is a dark shade of blue corresponding to zero flux density; however the magnets are a lighter shade which corresponds to 0.40 Tesla. The figure shows a green colour throughout the magnetic gap, this corresponds to a level of 1.2 Tesla.

To show the level in the gap more clearly the magnetic flux density has been measured along the length of the voice coil and output in figure 11. The rise in level occurs at the point of the voice coil which is within the magnetic gap.

The highest point will be the section of voice coil that is within the magnetic gap; at this point the model is showing a level of 1.209 Tesla. Outside of the gap the level is a lot lower. This type of voice coil is referred to as long coil. A short coil voice coil would have the whole coil within the magnetic gap.

The peak flux density was measured in the magnetic gap at 1.19T this shows that the model is predicting the magnetic field with a good level of accuracy.

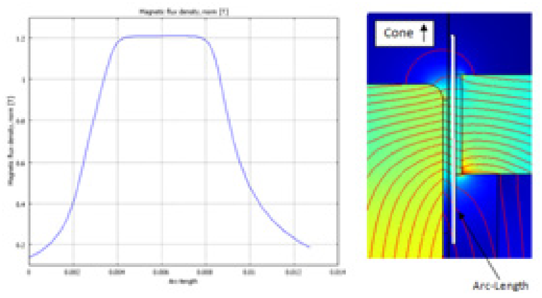


**Figure 11:** Magnetic flux density along the length of the voice coil. (0 relates to the top edge, closest to the cone of the voice coil)

The most important value for the model to get correct is Bl, this is the magnetic force factor and is a product of the average magnetic flux density and the length of the wire on the voice coil. For the drive unit under test the voice coil wire length was 9.992m and the average magnetic flux density was 0.676 Tesla. Giving a total Bl product of 6.75 Tm.

The average Bl product from a batch of 16 drive units measured by Klippel was 6.852 Tm with a range of 0.465Tm. This gives a difference between the measured average and the modeled value of 0.102Tm.

Due to the way drive units are magnetized after they are constructed the remnant flux density cannot be measured directly and so some of the error may be due to the ideal remnant flux of 0.40T not being achieved.

### 3.2. Blocked Coil Impedance

The results for the Blocked coil impedance do not correlate with the measurements at very high frequencies this is due to there being a slit in the aluminum voice coil former. This slit stops currents circulating perfectly around the voice coil former but still allows some current interactions with the voice coil. Due to the model being axis symmetric this cannot be modeled. It is however possible to run two models one with the voice coil former set to aluminum with perfect circulating currents and the other set with a non-conducting voice coil former. These two results straddle the measurement showing that a full 3D model would be needed to gain correct results.

The result shows good correlation through to 3kHz at which point the results and measurements diverge. In models with non-conductive voice coil formers, the correlation would be much closer due to the lack of error in the modeling of the currents in the former.

If there is another material placed in the magnetic gap that has a high conductivity the effect of the aluminum voice coil is less and the results match much closer.

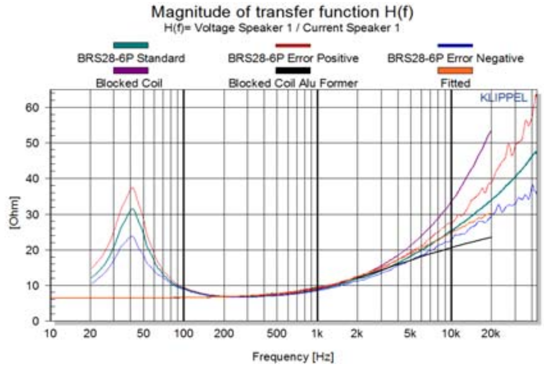


**Figure 12:** Blocked Coil Impedance Response

So that further analysis can take place without the compounding errors of this result, which in many cases will not be present as non-conductive voice coil formers will be used, the high frequency impedance will be matched to that of the measurements. This can be carried out by modifying the voice coil formers conductivity.

### 3.3. Impedance

The magnetic analysis and the blocked coil impedance both lay the ground for the structural-acoustical modeling, it is at this stage which the impedance and sound pressure level for a complete drive unit are calculated.

At every stage the material parameters have been very important to the accuracy of the model, the impedance is no different. The resonance in the impedance curve informs the designer of the main mechanical parameters of the drive unit. It is generally affected by the compliance and mass which is derived from the surround and spider and the mass of the moving parts plus the air load. This means that to get an accurate result all of these have to be measured and implemented correctly

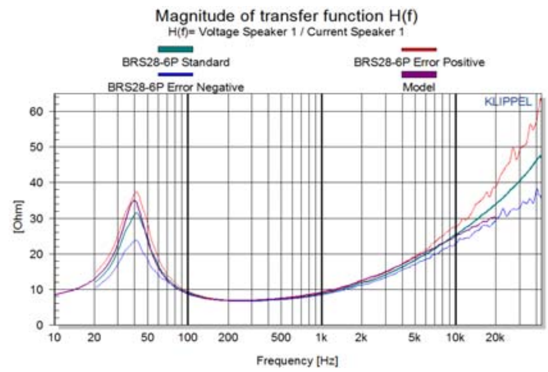


**Figure 13:** Impedance Response

The impedance result shows very good correlation with the measured results. The peak is placed at the correct frequency and the level is very similar. It is well within the range set down by the batch of measurements showing that this result is representative of the drive unit.

As the height of the resonant peak determines the mass spring damping of the system any deviation from the average measurement will indicate differences in the levels of damping. In the result there is a difference of 3.6 Ohms.

From the impedance curve a number of Thiele Small parameters can be calculated [5]. These inform the designer about the electrical and mechanical parameters of the drive unit.

| Parameter | Measured | Model |
|---|---|---|
| Qms | 2.327 | 2.287 |
| Qes | 0.500 | 0.502 |

**Table 3:** Q Factors

The quality factors show good correlation with the average of 16 measurements. This further shows the level of accuracy that this model has attained.

### 3.4. Sound Pressure Level

The sound pressure level is the main reference measurement that designers use to help predict the sound of a speaker. It is important that if a model is used for prototyping that this result is very accurate.

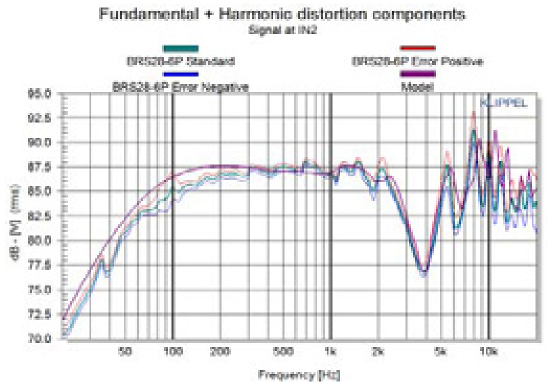


**Figure 14:** SPL Response

The low frequencies show error in the bass alignment; this indicates that there is too much energy in this region. Unfortunately without damping the resonance of the drive unit further this cannot be solved and this would also effect the alignment of the impedance response which currently is very good. This may indicate that there are further refinements to make with the modeling technique. This does not cause too much of an issue as classic techniques can predict the low frequency response of drive units very accurately. It must also be noted that when the surround and spider are replaced with a numerical solution the results are very good.

This difference indicates that there is error in the low frequency response that comes from the modeling of the spider and surround. Using a composite model which employs a numerical solution for the suspension at low frequencies can improve the results. Care must be taken with the surface area of the cone as a proportion of the surround is active at low frequencies.

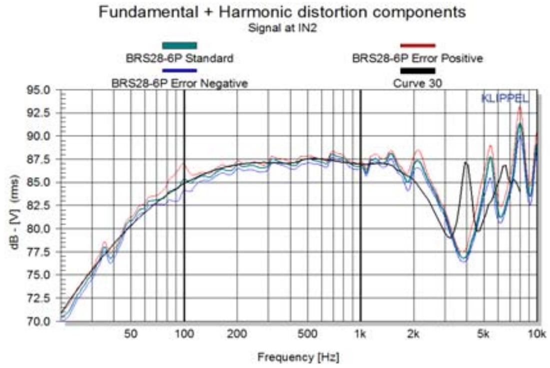


**Figure 15:** Frequency Response using numerical solution for suspension.

Through the mid band and up to 2400Hz the response is very representative of the measurement (figure 14). There are two bumps in the response one large one at 1400Hz and another at 2300 Hz. These represent two modes of the surround.

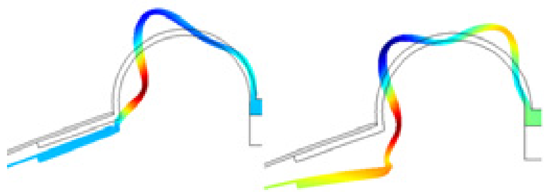

**Figure 16:** Left: 1200Hz, Right: 2300Hz

Looking at these modes in more detail reveals that they are mode 4 and 5. Controlling surround issues like these can improve the performance of a drive unit and having a method of predicting them is extremely useful.

At higher frequencies above the pass band of the drive unit there are a number of sharp spikes these represent the cone breakup. The results from the model show a good correlation with them, this indicates that the cone and its material parameters are very good. Making sure that the break up region is modeled well is critical for a designer as this can affect the crossover and compromise a design.

## 4. Conclusion

It has been shown through results that it is possible to use Comsol Multiphysics to create a model of a drive unit with a good level of accuracy.

The fundamental problem over all of the modeling domains comes from knowing the parameters of the materials being modeled. These parameters could do with further research and in some cases the application of more accurate approaches to their measurement. This will prove to be a large area of FEA development for loudspeakers in the future.

The largest error in the model was from the Electromagnetics application mode which modeled the blocked coil impedance, this was found to be due to the use of an axial symmetric model and would not cause issues in design which have non conductive voice coil formers like kapton or a 3D model.

The mechanical and acoustical domains have been shown to give good results both modeling the domains very well. The final results show good correlation with the measurements and show that with further improvements to the material parameters the model could improve in accuracy.

The methods developed in this report provide a solid base for modeling loudspeakers. Problems with the models have been highlighted and areas of note or inaccuracy have been found all of which aid in the development of models.

With the increase in competitiveness in the loudspeaker industry it is imperative that companies keep competitive advantage. In terms of research, the processes shown in this report would allow a designer the freedom to design new innovative loudspeakers. This will provide companies with new ways to test ideas leading to a series of breakthroughs in loudspeaker technology.

During the development of the model many obstacles were encountered, these were mainly in the form of geometry inaccuracies and material parameters it is therefore very important that no modeling occurs until these aspects have been thoroughly researched.

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