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Multiphysical Simulation Methods for Loudspeakers - Nonlinear CAE-based Simulations

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ABSTRACT

This is the third in a series of papers on the details of loudspeaker design using multiphysical computer aided engineering simulation methods. In this paper, the simulation methodology for accurately modeling the nonlinear electromagnetics and structural dynamics of a loudspeaker will be presented. Primarily, the calculation of nonlinear force factor $Bl(x)$, nonlinear inductance $L_e(x)$ and stiffness $K_{ms}(x)$ in the virtual world will be demonstrated. Finally, results will be presented, correlating the simulated model results to the measured physical parameters. From that, the important aspects of the modeling which determine its accuracy will be discussed.

1. INTRODUCTION

While most loudspeakers can be treated as axisymmetric systems, and thus simplified 2D models can be applied, its multidimensional vibration pattern can have a significant impact on the acoustical performance. Depending on the final application, e.g. non-axisymmetric enclosures, non-axisymmetrical behavior can be important, and advanced 3D models need to be used. Thus, typically finite element models

for detailed vibration system design and optimization are highly valuable and efficient design tools.

For system or subsystem level simulations (without the goal of designing a vibration system) 1D lumped models are highly efficient. A lumped parameter model can be found in [1]. The basic differences between 1D, 2D and 3D models has been described in that work and current work. [2], [3]

As we have seen in earlier work, even when the loudspeaker is driven in the region of nominal power,

the electromagnetic factor, $Bl(x)$, inductance $L_e(x)$, and stiffness $K_{ms}(x)$ are non-linear in their extremes. A significant portion of the voice coil can move out of the main flux field, and less mechanical force induced. This nonlinear effect is very essential and causes unwanted distortion in the radiated sound. Voice coil inductance is also dependent on voice coil excursion and also on current. Even if we would have a super-linear material for suspensions, the change in geometric stiffness leads to non-linear behavior. Non-linear modeling from linearization for dynamic calculations is needed to accurately simulation that behavior. [4] The challenges in comparison to theory and theory-based measurement systems are in the practical limitations of the theory.

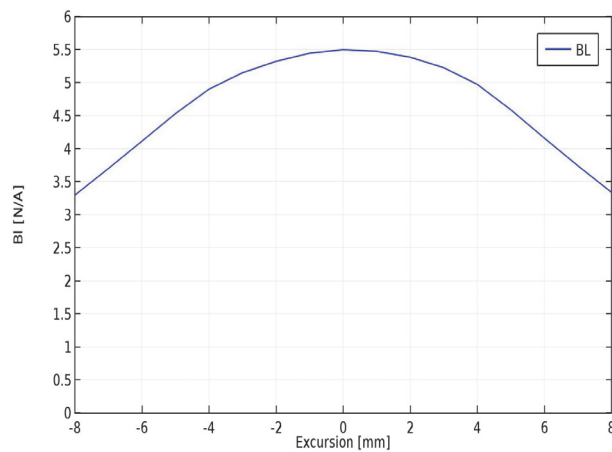


Figure 1. BL vs. x , calculated from moving mesh

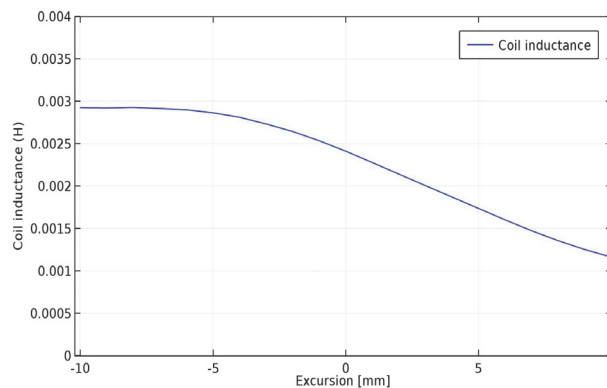


Figure 2. Electrical Inductance vs. Coil Excursion

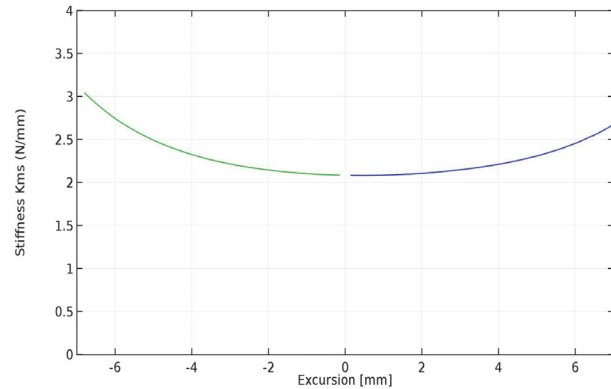


Figure 3. Stiffness vs. Coil Excursion

2. THEORY

2.1. Mathematical and Physical Background

There is a long tradition to treat loudspeakers as a multiphysical device. As early as the 1970's several technical papers were presented to the AES [5] that resulted in a model which is based on the "Thiele-Small Parameters". A.N. Thiele and R.H. Small developed a set of parameters and one-dimensional equations defining the relationship between a loudspeaker, a particular enclosure and the radiation of sound waves. [6] Over time there have been many developments to extend these early lumped parameter models to more sophisticated models with improved accuracy. Especially the recent work of W. Klippel on the large signal behavior (i.e. the nonlinear behavior of the loudspeaker) should be mentioned here [7].

However, all these lumped parameters models have one significant drawback: they are based on one-dimensional, scalar equations to describe a physical domain. Even though the usage of a lumped parameter model for voice coil electromagnetics turned out to be very useful, for the mechanical domain and for the acoustical domain there can be a lot of limitations. E.g. the mechanical system is described in a lumped parameter model by a set of scalar values of stiffness, mass and damping.

This fact led to the development of models based on matrix methods. A fully coupled electrical-mechanical acoustical simulation model for loudspeakers is used, which uses a lumped parameter model for voice coil electromagnetics, a finite element model for the

mechanical domain and a finite or boundary element model for the acoustical domain.

The most important challenge in this is the strong coupling of all physical domains involved. Strong coupling within that context means that each physical domain interacts bi-directionally with other domains.

The original theory for the lumped parameter models, the solution of the static solution and expansion into a non-linear result follows.

2.1.1. Electromagnetics

In the basic lumped parameter model for voice coil electromagnetics the electrical force f^e acting on the coil under the assumption of a constant voltage source U is defined as:

$$f^e = U \frac{Bl}{R + i\omega L} - \frac{(Bl)^2}{R + i\omega L} v^e = f^{e,L} + f^{e,EMF} \quad (1)$$

Where B is the flux density, l the length of the wire in B , R the DC resistance of the coil, ω is the angular frequency, L the inductance of the coil and v^e is the velocity of the moving coil. $f^{e,L}$ is the Lorentz force and $f^{e,EMF}$ is the back electromagnetic force.

This is just the basic form of the voice coil model. The current implementation also includes extensions to account for semi-inductive behavior due to eddy currents and ‘skin effect’ in the pole structure as well as transformer coupling between the voice coil and pole piece.

The loudspeaker is driven by a time-harmonic voltage, $V = V_o e^{i\omega t}$, applied to the voice coil. There is an electromagnetic relationship of the current in the voice coil and the driving force that this current generates. [8] Once the relation between the driving voltage and the force is set up, the force is applied in a structure interaction analysis to finally compute the impedance curve.

The current through the voice coil relates to the applied voltage as

$$I = V_o + V_{be} / Z_b \quad (2)$$

where Z_b is the blocked electric impedance and $-V_{be}$ denotes the back EMF.

To evaluate the back EMF, consider the same wire of length L in the magnetic flux density \mathbf{B} , but now traveling at a velocity \mathbf{v} . The wire gets an induced back EMF equal to $L\mathbf{v} \times \mathbf{B}$. The total back EMF in the coil equals to

$$-V_{be} = -v \frac{2\pi N_o}{A} \int r B_r dA \quad (3)$$

with B_r being the r -component of the magnetic flux density, and the integral evaluated over the volume occupied by the coil domain.

The first model step solves for a static solution of the magnetic circuit in the loudspeaker motor structure. The iron in the pole piece and top plate is modeled as a nonlinear magnetic material, with the relation between the B and H fields coming from measured data. The static solution provides the magnetic field density at any point in the model. And, it provides the local effective relative permeability,

$$\mu_r = B / (\mu_o H). \quad (4)$$

As stated before, these static solutions can be expanded into the frequency domain. The linearization discussed here is for the nonlinear material model used for dynamic calculations. These are the calculations used to derive the Impedance curve.

The numerical sub steps in the model have a stationary step followed by a frequency domain step, such that the stationary solution defines the linearization point for the subsequent frequency domain solution. This means that the magnetic field derives and uses a differential permeability inherited from the one computed by the stationary step.

To calculate the force factor (Bl) versus coil excursion (x) and Inductance (L_e) vs. x , there is a simplification performed. A moving mesh for the voice coil is used to obtain a static solution for each voice coil position in excursion. The results are the nonlinear behavior of the coil in a nonlinear field. Thus dynamics are excluded, which is valid as large excursions only happen in the low frequency range. In the case of Inductance (L_e), it is calculated at a given excursion value of x , and from the

imaginary part of the impedance for a given frequency (ω).

2.1.2. Structural Dynamics

From previous work, the governing equation for the mechanical vibrations in the frequency domain, discretized by means of finite elements, can be written as follows:

$$(K^m + i\omega D^m - \omega^2 M^m)u^m = f^m \quad (5)$$

And we remember that even though there seems to be only a little difference in the governing equations by matrix methods and by lumped parameter models. There is a big difference in the dimension of the system. In the finite element governing equation stiffness, mass and damping are being described via matrices. K^m is the stiffness matrix, D^m is the damping matrix and M^m is the mass matrix. Furthermore, u^m is the vector of displacements and f^m is the vector of mechanical forces exciting the system. ω is the angular frequency. Typically the dimension is of several of thousands degrees of freedom. In fact the governing equation is a system of equations describing the mechanical vibrations with respect to a detailed definition of the geometry (CAD model) discretized via finite elements. Thus it is possible to use these models for the whole audible frequency range which is typically from 20 Hz up to 20 kHz where a lot of non-pistonic and non-axisymmetric motion patterns occur.

For the structural solution, for $\omega = 0$ we get a static solution. The lumped stiffness of the system, typically referred to as K_{ms} , can be calculated, or its inverse, the compliance C_{ms} .

Again, as with the electromagnetic expansion into nonlinear estimates, the numerical sub steps in the model have a stationary step followed by a frequency domain step, such that the stationary solution defines the linearization point for the subsequent frequency domain solution

3. PRACTICAL LIMITATIONS

3.1. Modern Measurement & Linearization

In the last decade, the available modern techniques for measuring the small signal and large signal parameters

for loudspeakers have evolved greatly in providing a representation of nonlinear behavior. And, the comparisons made in this paper use measurements from one of the better measurement systems available today. Yet, there are practical limitations in using those measurements when comparing to simulations.

For example, the original definition of K_{ms} [6] was a static value. It was assumed to be unvarying with frequency in the range of use. So re-using it and including the effects of frequency can be dangerous. In measurements that are meant to represent non-linear measurements of Bl , K_{ms} , and L_e , dynamics are included in the derivation of those static parameters. In the case of Bl and K_{ms} , parameter estimations are performed so that the nonlinear curves can be developed through power series expansions. [9], [10], [11] And, in the derivation of L_e , a parametric curve fit is done in a similar way, regressing to an error curve.[12] These are assumptions made in the definitions of the parameters.

In the simulation for the example of K_{ms} , a frequency of zero (0) for $K_{ms}(\omega)$ is used. And a stepwise solution for K_{ms} is performed: a frequency response linearized around the static solution. That is to say: The frequency domain solution is solved in every position of the coil, moving it statically, with no dynamics considered or estimated. It is a direct calculation at every step.

So, by definition, the results of direct calculations in the simulation without dynamics estimated shouldn't match measurements made and estimated with dynamics included in the derivation.

3.2. Loudspeaker Manufacturing Tolerances

There are other limitations when comparing virtual simulation results to measurements of real world loudspeakers. If the manufacturing tolerances of the loudspeaker components and system are not considered, then correlation to loudspeaker measurements can also have areas of disagreement. Manufacturing tolerances can be much higher than the error tolerance in the simulations, which are much more accurate.

Some typical component and loudspeaker system tolerances for branded level design specifications are as such: Cone Mass ($\pm 10\%$), Speaker Resonance ($\pm 20\%$), Spider Displacement ($\pm 15\%$), Voice Coil DCR ($\pm 25\%$),

and allowable loudspeaker frequency response deviations of $\pm 2\text{dB}$ (a 4dB spread) in the piston band, which could be even higher in the frequency bands above that. These are practical tolerance ranges for automated and non-automated manufacturing lines today. This does not consider the details of glue weight and glue placement variations in manufacturing or the variation that exists within assembly process for a given loudspeaker design, all of which can be significant.

It is therefore unwise to simulate a loudspeaker purely on the facts from a specification sheet, unless there is nothing else for the basis of the simulation. If there is nothing else, then an understanding of the real world variation is needed before assessing simulation results.

4. MEAUREMENT & SIMULATION

The following are simulations results for a given woofer and a comparison to real world measurements for the nonlinear force factor $Bl(x)$, nonlinear inductance $L_e(x)$ and stiffness $K_{ms}(x)$. The simulation is shown here in 2D.

4.1. Electromagnetic System Model

Setting up a model is straight forward and relatively simple (at least in 2D). Starting point is a 2D (cleaned-up) cross section as given in the following figure:

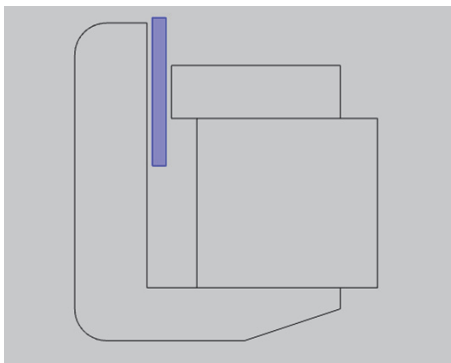


Figure 4. 2D cross-section of a typical coil and motor system.

For details on general CAE modeling, such as geometry and material definition, input parameters, applying physical laws, and parametric studies, we refer to [12]

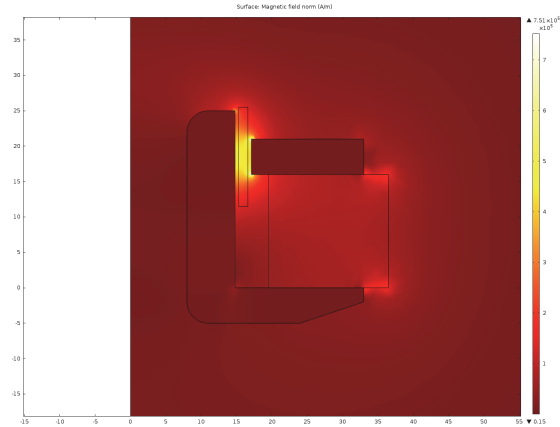


Figure 5. Magnetic Field Norm

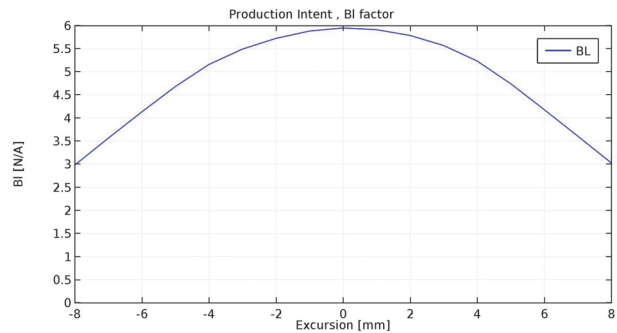


Figure 6. Bl Factor based on Design Intent

The Bl factor is calculated as explained before, step the voice coil through the radial B_r field. Shown in Figure 6 is what the design intent of the Bl factor.

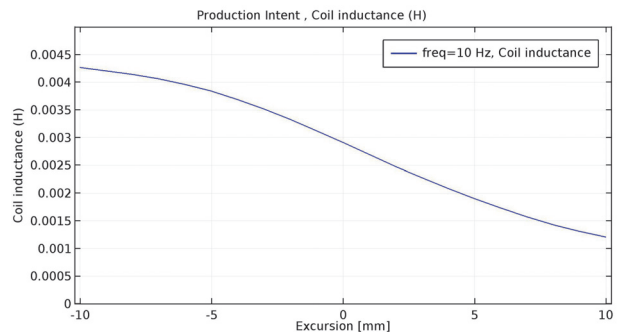


Figure 7. L_e based on Design Intent (@10Hz)

4.2. Structural Vibration System Model

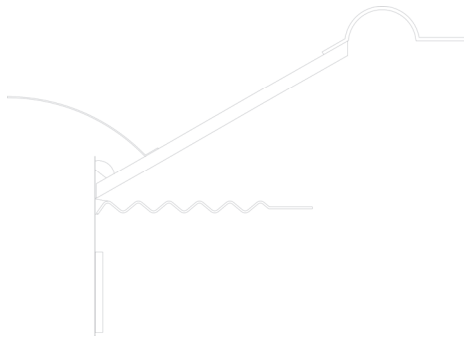


Figure 8.2D cross-section of a typical vibration system

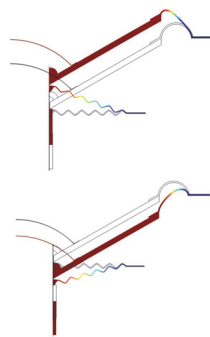


Figure 9. Positive & Negative Displacements

To simulate the Stiffness vs excursion of the spider and surround we run two different sweeps that start from zero force. The first sweep is in the positive direction (z direction) ranging from 0 to a reasonable extreme force value with small force steps. The second sweep is in the negative direction. The geometric non linearity is included in the stationary solver.

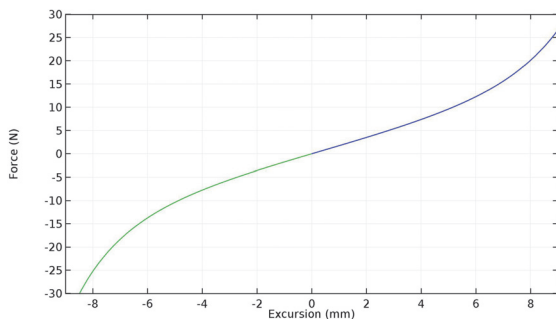


Figure 10. Force vs. Displacement Design Intent

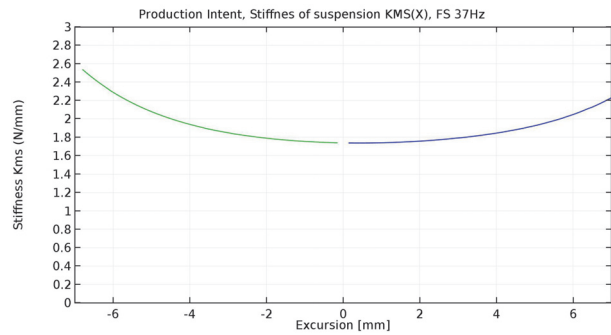


Figure 11. Stiffness (K_{ms}) Suspension Design Intent

4.3. Measurements vs. Simulation Results

The following are some comparison of simulation and measurements using Klippel measurements of a woofer that was built in a sample lot to the specification of the design intent.

The simulations, the results of which are shown in Figures 6, 7, 10, and 11, are made using the parameters from the specification for the design intent of the woofer.

From the measurements, we first find the resonance of the loudspeaker to be $F_s = 28$ Hz. The simulation calculated $F_s = 37$ Hz. During the linear parameter measurement (LPM) and large signal measurement (LSI) the suspensions are “broken in” and the system resonance drops from design intent by 25%. This is more rightly a combination of variation of cone parameters and measurement process.

Second, we find that the voice coil is not centered at the zero excursion point. The measurements show an off - set of approximately -1.5mm. (Figure 12) This is attributable to variation in the suspensions, coil placement during the build (spider offset in the gluing area -- tilting, no glue outside, only inside -- and cone shifting during assembly), and any softening of the suspension during test.

We also see an off-set in the maximum Bl value for the measurement when compared to simulation. The measured $Bl(0) = 5.66$ T·m, and the simulated $Bl(0) = 5.99$ T·m. This magnitude off-set of 5.5% is attributable to the approximation algorithm in the measurement for deriving Bl magnitudes. There are typically no significant magnitude differences in motor structures

unless there is a fault in the magnetization during the build.

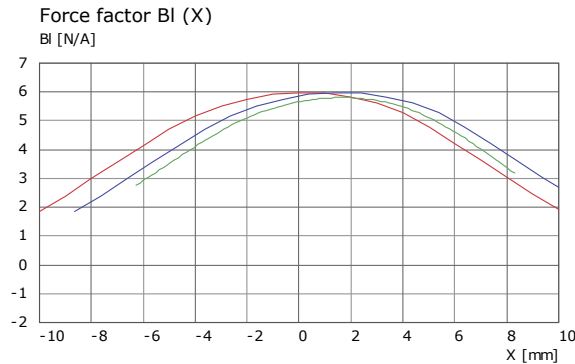


Figure 12. Force factor (BI) Comparison

- Simulation Design Intent
- Sim: Suspension CAD mod to match BI offset (-1.37 mm)
- Klippel BI offset of -1.5 mm

We also see from the stiffness measurements (Figure 13) that there is a difference between the measurement and simulation values. This is also attributable to the conditions just mentioned and the approximation algorithm for deriving K_{ms} nonlinear curves.

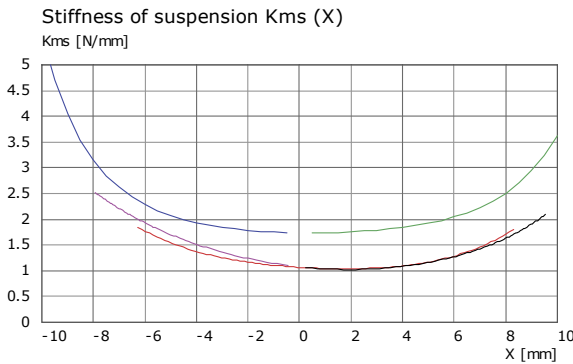


Figure 13. Stiffness (K_{ms}) Comparison

- iO: Simulation Negative X, Design Intent F_s 37Hz
- iO: Simulation Positive X, Design Intent F_s 37Hz
- iO: Klippel F_s =28 Hz during measurement
- iO: Sim Neg X, Suspension CAD to match F_s 28Hz
- iO: Sim Pos X, Suspension CAD to match F_s 28Hz

If we compensate the CAD geometry (Figure 14) for the differences in coil position from design intent and the sample by -1.5mm, and then compensate suspension components in the CAD and the model for “break-in” and build variation, then we can see closer agreement

between the simulation curves and measurement curves in Figures 12 and 13.

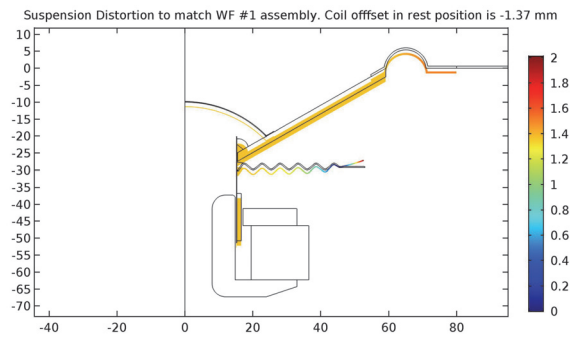


Figure 14. Force factor (BI) Comparison

All the differences we’ve seen were in the variation between the sample build and the design intent and the effect the measurement technique had on the results. They were easy to investigate and understand in a virtual world.

When we look at the Electrical Inductance (L_e) results (Figure 15), we see great disparity in magnitude in the measurements by Klippel methods and the simulation, even with large magnitudes of scaling added to the simulation in order to make the visual comparison. Woofers rarely, if ever, have magnitudes of electrical inductance on the order that was measured here ($> 1mH$). The methodology for approximating the nonlinear curve for electrical inductance [12], [14], [15] is still in its infancy and could mature. And, it is more likely the stepwise modeling based on localized permeability is closer to the real event.

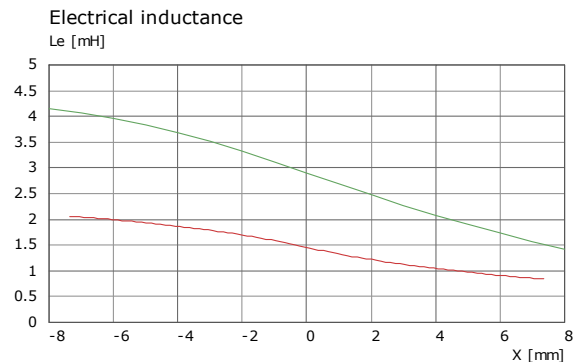


Figure 15. Electrical Inductance (L_e) Comparison

- Klippel
- Simulation @10Hz (scaled)

5. FINAL COMMENTS

While there are some practical limits in the measurement techniques and the use of untested design specifications to correlate virtual simulations to real sample builds, the differences are easily understood, and greatly demonstrated in the simulations. Manufacturing tolerances and the measurement techniques have a larger variability in themselves than the simulation technique, which is at greater levels of accuracy. The area of greatest disparity, points to a greater need for growth in useful inductance measurement techniques. The value of these simulations to aid in making accurate design decisions remains incomparable in the loudspeaker industry.

The audio industry, as well as most industries today, is challenged by the need to constantly increase engineering efficiency. Computer-Aided Engineering (CAE) based on simulation and analysis of the functional performance of products has already played a key role for more than two decades. CAE methodologies are today typically used at every stage of the development cycle, from first concept studies up to detailed engineering for final product development to be released to the market place (including modeling of the manufacturing processes as well). During the last years a strong trend for moving CAE upfront in the design process (to be applied already in the concept phase) can be monitored. Thus the term frontloaded Virtual Product Development (VPD) is often used. The advantages for moving CAE upfront are: [16], [17], [18]

- more freedom in the design decisions
- design changes can be made at lower costs

These advantages additionally fulfill the above mentioned basic requirements to increase engineering efficiency. Similar thoughts have been applied to the automotive industry [19].

This frontloaded approach was first (successfully) introduced for the development of automotive and aerospace key components at the Automotive Original Equipment Manufacturers (OEMs). A good example for a first application of VPD is the development of car body structures with respect to crashworthiness, fatigue or NVH behavior. Today nearly every industry is finding the need to follow the path of a frontloaded VPD cycle as well.

In this paper, we can precisely see that advanced CAE methods can accurately predict the structural dynamics of loudspeaker's vibration systems. Thus we can optimize designs in a very early design stage where no physical prototypes exist. Ultimately these results can be used to see the behavior of the loudspeaker when coupled to an enclosure (to be published in a future paper), whether that enclosure it is simple or complex geometry. Complex geometries are more and more commonplace as the need to innovate new designs and integrate loudspeakers in unique ways grows rapidly. Here again, CAE methods are key technologies to optimize product development in terms of performance and cost, resulting in optimization of engineering efficiency in general.

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