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The Application of FEM to the analysis of Loudspeaker Motor Thermal behavior.

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ABSTRACT

A simple 2D Finite element method (FEM) may be applied within the thermal domain to analyse the behaviour of systems where fluid flow is not a significant factor. FEM's ability to analyze problems with arbitrary geometric form makes it a powerful alternative to the lumped element approach for modeling loudspeakers. Applying suitable boundary conditions, and using modified material properties, it is possible to model the thermal behavior of a loudspeaker motor with FEM. This approach minimizes errors due to fluid flow and includes heat radiation to the environment. The FEM technique is applied to a new driver topology with external frame and magnet. Static thermal FEM results are compared to those obtained from the driver by measurement. The material properties for air were derived from experimental results. A separate model, including the full mechanical structure of the coil, is used to derive its bulk thermal properties thus allowing a more efficient solution.

INTRODUCTION

The purpose of this paper is to show how the application of Axisymmetric FEM to the thermal domain can provide a useful and informative alternative to the equivalent circuits used in past work on modeling heat in loudspeakers [1][2]. Before considering the advantages and limitations of FEM applied to the problem of a loudspeaker we should recall that the heat leaves the loudspeaker in four main ways:

Radiation

Heat transfer from a material surface by radiation occurs due to the emission of infrared electromagnetic radiation from the electrons on the surface of the material. Radiation can travel through a vacuum, the amount of radiation depends on the absolute temperature. The heat radiation into free space is given by the Stefan-Boltzmann equation in equation 1.

$$Q = \sigma AT^4 \quad [1]$$

Where Q is the rate of heat transfer in [W], A is the surface area of perfectly radiating body in [m²]. σ is the Stefan-Boltzmann constant, 5.669×10^{-8} [W/m²/K⁴] and T is the absolute temperature of surface [K]. For heatflow between two surfaces we must consider the heat being emitted and absorbed by the surfaces. For two surfaces A_1 and A_2 the radiation exchange Q is given by equation 2.

$$Q = \sigma A_1 \xi_1 F_{1-2} (T_1^4 - T_2^4) \quad [2]$$

Where Q is the rate of heat transfer in [W], A_1 the area of radiating surface 1 [m²], T_1 the absolute temperature of surface one [K], T_2 the absolute temperature of surface two [K]. ξ_1 is the emissivity of surface one ($\xi \leq 1$). F_{1-2} is the view factor which quantifies how well surface two is viewed by surface one ($F_{1-2} \leq 1$).

Conduction

Conduction is the transfer of heat in the form of kinetic motion of molecules within a material. Conduction occurs in solids liquids and gases, however in fluids convection is a much more significant effect. Heat flow due to conduction is described by Fourier's law, equation 3.

$$Q = -kA \frac{dT}{dx} \quad [3]$$

Where Q is the rate of heat transfer in [W], A is the cross-sectional area in [m²], k the thermal conductivity of material in [W/m/C], and dT/dx is the temperature gradient [C/m].

Natural Convection

Convection is the transfer process due to fluid motion. In natural convection, the fluid motion is due entirely to buoyancy forces arising from density variations in the fluid. Traditionally convection heat transfer is estimated using the method of dimensional analysis [3] to develop empirical correlation formulae for various geometric shapes. These formulations relate the heat transfer coefficient, H_{nc} [W/m²/C], to the geometry, temperature and fluid properties. Simplified equations suitable for air cooling are also available – an example of which is given in equation 4 for laminar flow over a vertical flat plate [3]

$$H_{nc} = 1.41 \left(\frac{\Delta T}{L} \right)^{0.25} \quad [4]$$

Where ΔT is the temperature difference between plate surface and ambient fluid temperature and L is the characteristic length of the plate.

It is worth noting that for pure 'Natural Convection' there must be no air movement due to other reasons - in general this is not the case, there at least being a small amount of forced convection as well.

Forced Convection

Forced convection is the transfer of heat by the movement of fluid by an external force. If the fluid velocity is large then turbulence is induced. In such cases the mixing of hot and cold air is more efficient and there will be a significant increase the heat transfer. As with natural convection, there are many empirical correlations for predicting the heat transfer coefficient, H_{fc} , for various geometric shape, fluid properties and velocity. Example of simplified equations suitable for laminar and turbulent air flow with velocity v flowing over a flat plate are given in equations 5 and 6. These equations are derived from those given in [6].

$$H_{fc} = 3.9 \sqrt{\left(\frac{v}{L} \right)} \Delta T \quad [5]$$

$$H_{fc} = \frac{210}{L} (2.71(vL)^{0.8} - 1) \Delta T \quad [6]$$

Turbulence is beneficial in respect to heat transfer however it is likely to cause distortion in a loudspeaker.

THERMAL FINITE ELEMENT METHOD

The FEM software used in this paper was developed for applications such as electric motors and induction heating, although it does not model fluid flow it does offer the capability of coupling thermal magnetic and electrical problems [4]. For the sake of simplicity this paper has not made use of these advanced facilities although they are expected to be useful for future work.

Static Thermal FEM

The static FEM used in this paper is based on equation 7 where Q_H is the thermal source density or dissipated power [4] and T is the temperature. Static FEM solves for the settled temperature when the system has reached thermal equilibria and the only material property required is thermal conductivity.

$$\nabla \cdot (-k \nabla(T)) = Q_H \quad [7]$$

Heat loss from the model is achieved by means of the non-homogenous Neuman boundary condition:

$$k \frac{dT}{dn} = -F_H - h(T - T_a) - \xi \sigma (T^4 - T_a^4) \quad [8]$$

Where T_a is the ambient temperature, dn is an incremental distance along the boundary, F_H is thermal flux from outside, and h is the heat transfer coefficient for convection.

Transient Thermal FEM

The transient analysis is based on the equations 9 and 10 which both have a factor which is the product of the specific heat capacity and dT/dt the rate of change of temperature with time. For this analysis both the specific heat capacity and thermal conductivity are required. The solution is solved in timesteps small enough to allow only a small change of temperatures between steps.

$$\rho_c \frac{dT}{dt} + \nabla \cdot (-k\nabla(T)) = Q_H \quad [9]$$

The non homogenous Neuman boundary condition for the transient case is given below:

$$\rho_c \frac{dT}{dt} + k \frac{dT}{dn} = -F_H - h(T - T_a) - \xi\sigma(T^4 - T_a^4) \quad [10]$$

Application of the thermal finite element method.

A significant limitation of both FEM and lumped element analyses is that fluid flow is not predicted and empirical data is required to determine the magnitude forced and natural convection [1][2]. In the case of FEM, the effects of thermal conductivity and capacity may be evaluated for the desired axisymmetric or three dimensional geometry without the assumptions required to ‘lump’ regions into a few single dimensional elements. For this paper it was decided to use an axisymmetric geometry allowing a rapid model set-up and solution time. Due to the small size of the voice coil structure a separate model is used to develop a composite material representing both conductor and insulator of the voice coil.

Derivation of composite voice coil material properties.

As with most FEM, in Thermal FEM using more elements will give better spatial resolution and convergence towards the correct solution. The mesh should be finest in regions of rapid change of temperature, however a finer mesh will be slower to solve, the solution time being approximately proportional to the square of the number of elements. In Magnetic FEM it is possible to mesh the voice coil region with elements larger than the wire size since the voice coil region is given constant current and has constant magnetic permeability. However, in the case of thermal FEM, the heat flow within the coil depends on the dimensions and material properties of wire insulation and bonding materials. Modeling the full coil geometry would be very time-consuming and would lead to a huge number of elements so we have chosen to derive bulk thermal properties of the voice coil by an additional FEM model of the voice coil.

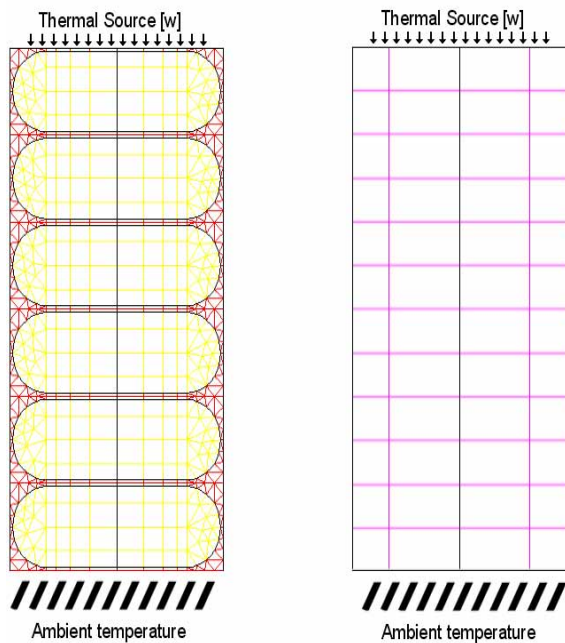


Figure 1. Full voice coil model.

The left hand region in figure 1 shows the full coil modeled with copper and insulating material. The problem is then solved for a range of values of thermal conductivity applied to the right hand mesh. The temperature adjacent to the heat source was plotted as a function of conductivity allowing the correct value of ‘composite’ coil conductivity to be read of the graph. The correct composite material property may be found at the intersection of the temperature curves as shown in Figure 2. Please note that the units used for temperature in the graphs throughout the paper are degrees Celsius.

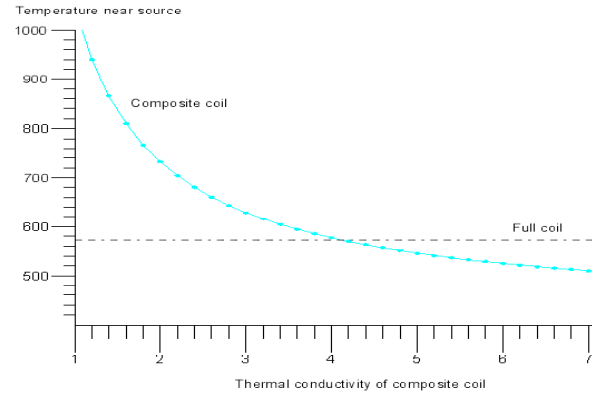


Figure 2. Temperature of composite versus full coil FEM models.

Axisymmetric Geometry development.

While the magnet and coil are axisymmetric the frame is axi-periodic having a similar geometry to a wagon wheel. We need to create an axisymmetric geometry thermally equivalent to the 3D geometry. A number of factors must be considered; firstly the length and cross-sectional area of heat paths must be maintained to give the correct thermal resistance. Secondly the volume must be correct to give the correct thermal capacity. Thirdly the emissivity and convection exchange coefficients must be adjusted to give the correct heat transfer. The axisymmetric version of the frame has too much surface area and the convection and emissivity coefficients must be reduced by the ratio of actual area to modeled area.

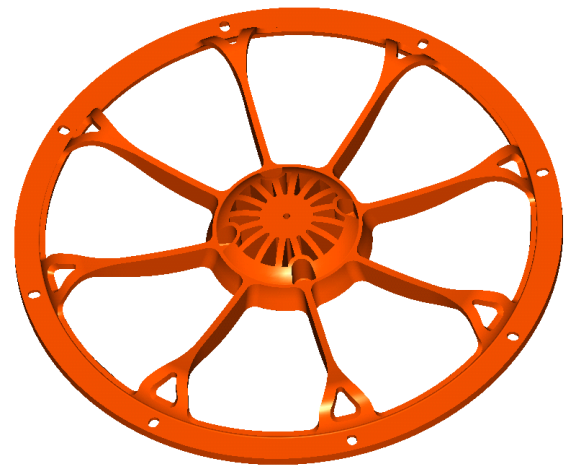


Figure 3. Nti 15 Frame

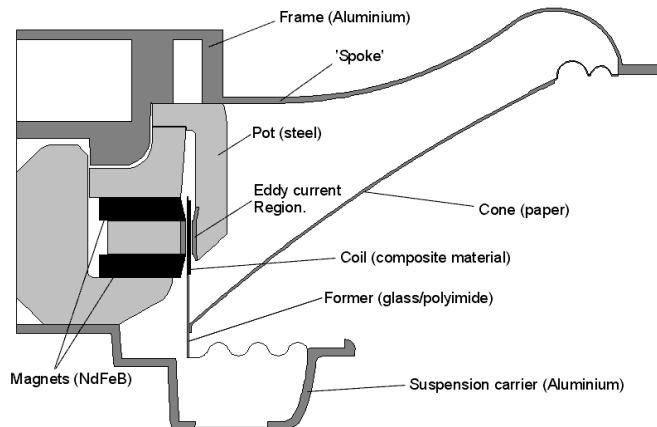


Figure 4. Axisymmetric geometry of Nti15.

Having created the axisymmetric geometry another important consideration is the glue joints; The scale of these is small and requires the use of quadrilateral elements to produce the most efficient mesh. Since the heat flow along the glue joint is very small these quadrilateral elements may have large aspect ratio's avoiding the need for a very fine mesh. The complete mesh with an inset showing the mesh at a glue joint is shown in figure 5.

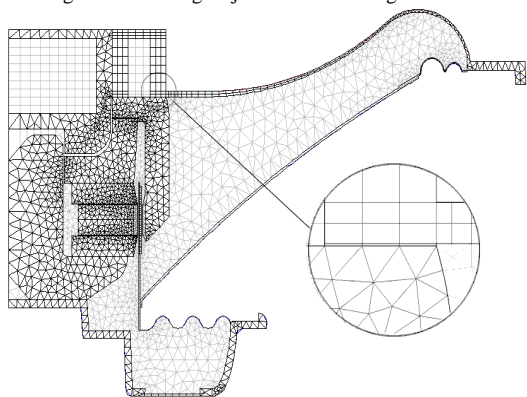


Figure 5. Picture of general mesh with glue joint detail.

Boundary Conditions

For the sake of simplicity the decrease in power due to the rise in coil electrical resistance with temperature was not included in this model. The actual power dissipated in the coil during our experiments calculated from impedance curves measured at the appropriate power and temperature. The voice coil was modeled by a heat source with constant power density and the composite properties derived from the full voice coil model. The power input to the driver was divided into eddy currents that directly heat the poles adjacent to the coil and direct coil heating. The percentage of power found as eddy currents was derived from Transient Magnetic FEM models [5] and was about 6% for the drivers without rings and 14% for the driver with rings

Since the thermal properties of the air in the gap are modified by the voice coil motion, the thermal conductivity of air and its emissivity were derived, by inspection, from experimental results and a series of FEM solutions over ranges of values of the unknown variables.

To find the correct value of emissivity the driver was suspended on thermally insulating mounts in a vacuum chamber the interior that

was treated with highly absorptive coating. The vacuum completely eliminates any heat loss due to convection, leaving the loss due to radiation. A known power was applied to the loudspeaker, which was allowed to reach thermal equilibrium. The coil and magnet temperature were measured and noted. A FEM model of the driver in these conditions was then created using a Non Homogenous Neuman boundary in equation 8 which allows Radiated and Convected heat loss to be modeled. These boundaries were applied around the outside of the mesh with the area correction factor for the spokes. Since the software used cannot calculate thermal absorption, the coil region was modeled with a fluid to allow heat transmission from the coil to the magnet. The input power for all models was calculated for the 'hot' coil impedance using AkaBak. This FEM model was solved without convected losses for a range of emissivity and fluid thermal conductivity. After some iteration a value for emissivity was found.

To find the convective coefficients it was necessary to measure the coil magnet and frame temperatures for the driver in free air. This allowed the convection coefficients for the air in the gap to be determined via comparison with FEM results using the radiation coefficient from the vacuum chamber experiment.

The first driver we will consider is the PA15. It is a fifteen inch driver with a high efficiency NdFeB magnet and a four inch nominal diameter voice coil. The PA15 has a cast aluminium four spoke frame with a rear suspension carrier of thin pressed steel. To keep convection to a minimum it was decided to use a pink noise test signal bandpass filtered at 108Hz and 410Hz using 24dB/octave Butterworth filters at a nominal power level of 100w. The mean voice coil temperature measurement was achieved by measuring coil resistance as described in [1]. Temperature distributions are shown in table 1 for both FEM results and experimental results. The FEM temperature distributions are in good agreement with the experimental results. However, the voice coil and magnet temperatures are unacceptably high considering the desired power rating of the driver. The magnet temperature for a NdFeB structure must be kept low due to the low curie temperature as well as keeping the peak coil temperature below the insulation failure temperature.

Aluminium heat rings were introduced to the ID of the coil creating the driver PA15 Alring. This reduced the thermal resistance between voice coil and magnet assembly, thereby cooling the voice coil, but also resulted in an increased magnet temperature, which is not acceptable given the thermal limitations of NdFeB. The initial results from this driver were somewhat anomalous due to the change in airflow in the gap and also a poor glue joint between frame and magnet. The results for the adjusted model are shown in table 1. Reducing the glue gap to the usual thickness gave a lower, but still unacceptable magnet temperature, leading to the abandonment of the AL rings for this driver.

What transpires from the PA15 result is that the thermal resistance from the magnet assembly to ambient was too high. In order to reduce this thermal resistance, the design of the frame was re-evaluated and the number of spokes, which allow heat transfer out of the central magnet assembly region was increased. Also the suspension carrier at the rear of the driver was redesigned using aluminium, creating the Nti-15 discussed earlier in the axisymmetric geometry section. The experiment described above was repeated on the Nti-15 driver. The results due to this change in geometry are shown in table 1. A significant drop in magnet assembly temperature was achieved over the PA15 Alring and large drop in both voice coil and magnet assembly temperature was achieved over the PA15 driver. Here the agreement between FEM and experimental results is very good. The isotherms and gray shading may be seen in figure 6

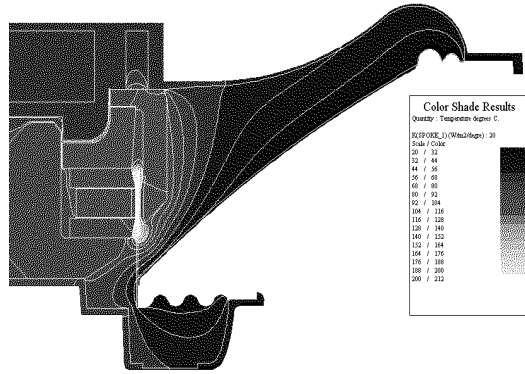


Figure 6. Thermal contour plot of Nti 15.

In order to evaluate the validity of this model at higher powers the experiments were repeated at 200W nominal power input. The results in table 1 again show good agreement between the FEM and measured data.

To investigate the effect of the forced convection produced by movement of the cone and coil the, NTL-15 was re-measured with a 50Hz rather than 108Hz cut off. This produced an 8% temperature decrease in the coil temperature. To model this thermal short circuit, a reduction in input power was applied such that the magnitude of the temperature distributions agreed with experiment. Once again the problem was solved over a range of parameter values to find the correct input power so the temperature distribution could be checked. It was somewhat gratifying to find that this approach gave good agreement with the measured results.

TRANSIENT SOLUTION

Having set up the model it may then be solved as transient problem, in our case for a constant heat input. The results of a transient solution For the Nti 15 driver are illustrated in Figure 7 and figure 8. These show how the temperature distribution along the coil develops with time and how the temperature rises at various points in the driver respectively. It can be seen that the time-constant gets larger for points further from the voice coil.

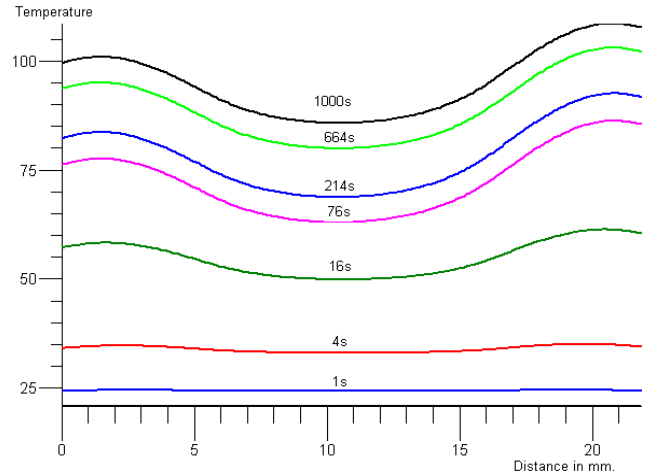


Figure 7. Temperature through the coil plotted at various times.

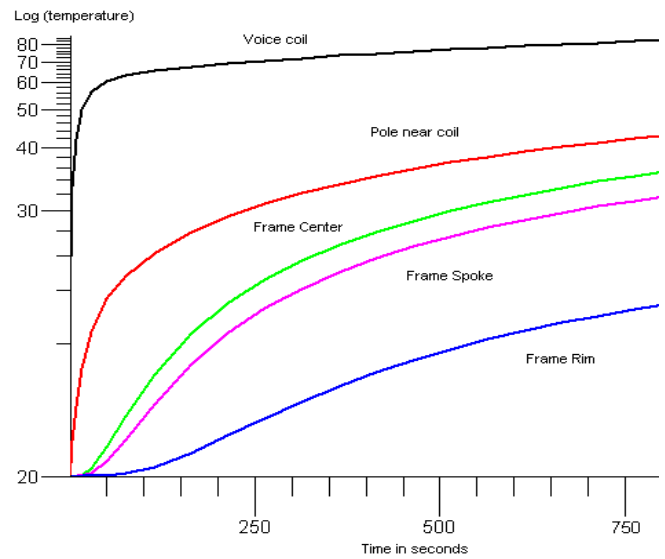


Figure 8. Temperature of various points throughout the driver plotted as a function of time.

Driver	PA15		PA15 Alring		Nti 15		Nti 15		Nti 15	
Nominal Power	100W		100W		100W		200W		100w	
Frequency range of noise input.	110Hz-400H		110Hz-400H		110Hz-400Hz		110Hz-400Hz		50Hz-400Hz	
Data origin	Meas'/ FEM		Meas'/ FEM*		Meas'/ FEM		Meas'/ FEM**		Meas'/ FEM***	
T(v/c)	123	124.7	113	119	115	113	206	201	98	97
T(mag)	73.4	72.5	97	95.3	62.0	60.4	98.1	96	51.5	53.7
T(frame)	63.2	60.2	68.7	68	50.5	48.8	73.1	73.9	41.4	43.9
T(front)	60.4	60.8	67.2	69.1	50.8	49.9	71.5	75.6	42.0	4.4
T(arm)	47.4	46.4	55	52	40.4	40.3	56.9	58.2	32.4	36.7
T(rim @ arm)	31.4	30.8	31.5	33.3	26.4	30	33.0	38.1	21.7	28
T(back mid)	63.4	58.8	95	93.6	47.2	46.8			40.8	42
T(back edge)	29.8	34.5	31	27	35.7	34.3			33.2	31.9

* Frame glue joint thickness increased, air conductivity increased.
 ** Power input reduced by 3.6% to allow for convection form coil to ambient.
 *** Power input reduced by 8% to allow for convection form coil to ambient.

Table 1

CONCLUSION

Thermal FEM can provide a method of quickly and inexpensively experimenting with design parameters. Experimental work is required in order to validate the results obtained from the model.

A solver which allows direct access to geometric and material properties allows this technique to be applied in an efficient manner by solving parametrically over ranges of different variables.

Effects due to complex conditions, such as fluid flow in the gap, can be modeled as modified material properties of the region we are concerned with.

Glue joints have proved to be a significant part of the thermal circuit and should not be neglected, although their small scale does require rectangular elements to produce an efficient mesh.

Complex regions of the driver, such as the voice coil, where the scale requires that a very large number of elements are required may be modeled as a composite material having the same bulk thermal properties.

These techniques will prove vital for the production of 3D FEM models where the solution time will be much longer. While producing the 3D mesh is likely to prove somewhat onerous it will eliminate any need to spend time simplifying the geometry. The question of 2D versus 3D modeling is an important issue and will be the subject of future work by the author.

Further refinements that could be made would be to couple the thermal FEM model to an electric FEM model with coupled circuit & drive from voltage source. This will not only give the resistance rise as a result but will also correctly model the power dissipation in the voice coil.

Acknowledgments.

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