Suspension design, modelling and testing of a Thermo-Acoustic driven Linear Alternator

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***Abstract*— The Score-Stove™ generates electricity from a wood-burning cooking stove using a thermo-acoustic engine that converts heat to sound through a Linear Alternator (LA). This paper introduces a prototype hemi-toroidal suspension that was refined into a segmented trapezoidal shape that gave a higher cyclic life for the linear alternator and includes a critical evaluation that compares a theoretical analysis with experimental results. The results show an improvement from the 40% efficiency of a standard loudspeaker used in reverse as an LA to 70 to 80% efficiency with the new suspension and a double Halbach array magnetic topology.**

***Keywords*—Alternators, Acoustic, Suspension, Thermo-acoustic and Resonance.**

1. Introduction

There has been growing interest in the thermo-acoustic technology over the last few years, however little until very recently has been done on its application. A Thermo-Acoustic Engine (TAE) converts thermal energy into sound energy by transferring heat between a working medium (gas) and a hot heat exchanger and one at ambient temperature. Between the two is a porous solid structure called a stack in the case of standing wave engines, or regenerator for the travelling wave type. This sound energy drives a Linear Alternator (LA) to generate electricity. In a number of applications for example the Score-Stove™ [1], a clean burning stove that generates electricity for poor rural areas, production cost and efficiency are the major challenges for researchers [2], [3]. Commercially available LA's use moving magnet [4], [5] or flux switching elements [6] that are too expensive for the rural applications [7]. A more suitable alternative is moving coil technology found in the construction of commercial loud-speakers that can be used in reverse as a LA [8]. However these devices are less efficient when used as a LA (This is typically 40%, and their suspension possesses limited mechanical stability [9]) due to the higher damping in their suspension; a necessary feature for hi-fidelity audio that is unimportant when used as an electrical generator. Additionally, previous work [10] has shown that increasing the TAE operating frequency reduces the cost of the LA due to less magnetic material, but requires significant improvements in suspension design. For a looped tube TAE, a resonant LA, when at the same frequency as the acoustic loop, presents a real impedance to the acoustic wave increasing overall system efficiency by reducing acoustic losses due to lower peak pressures in the feedback pipe [11]. This paper presents two designs of segmented sealed suspensions not previously defined in the literature; a hemi-toroidal and trapezoidal shaped suspension, along with a theoretical model of the latter validated with experimental results from a prototype.

2. Alternator Suspension material

Moving coil loudspeakers have poor performance as LAs due to stroke limitation, unreliability of their suspension and high pumping loss [8]. The loudspeaker has low stiffness (high compliance) and its free air resonant frequency decreases with aging due to suspension degradation as they are usually made from a paper or woven fabric. The limited stroke problem can be addressed by changing the magnetic topology to a double Halbach array which has the added advantage of increasing magnetic strength in the coil cavity by 2 times at the expense of a non-sinusoidal waveform [10]. The remaining challenge is to design an LA suspension using low damping loss spring material with high cyclic life.

The mass on a spring has a single resonant frequency determined from its spring constant (k) and the mass (m). The resonant frequency of the spring can be expressed as;

$$\omega_n = \sqrt{k/m} \tag{1}$$

A low damping loss suspension can provide high displacements at its resonant frequency, but it needs to withstand the required displacement within the suspension material infinite fatigue limit. Table 1 shows the common spring material damping and elastic properties summarised from references [9], [12], [13], and [14]. It can be seen from the table that although rubber has a high strain capability, it also has the highest damping loss and lowest mechanical strength compared with the other materials. Steel has the highest Ultimate Tensile Strength (UTS) and lowest losses, and so is the material chosen. Further improvements in steel material properties were obtained by using a proprietary heat treated spring steel. The design target displacement of the alternator is 18 mm at 100 Hz frequency to achieve the 150 W output requirement [6]. If we consider a simple cantilever suspension the maximum yield stress can be calculated using the following formula [15];

$$\sigma = \frac{6\delta E t}{4L^2} \tag{2}$$

Where δ is the vertical deflection, E is the young modulus, t is the thickness and L is the length of the material. The resonant frequency of the cantilever is expressed as;

$$f_n = \frac{1}{4}\frac{t}{L^2}\sqrt{\frac{E}{\rho}} \tag{3}$$

Where t is the thickness, L is the length and ρ is the density of the material.

Using equation (3) the stress becomes;

$$\sigma = 6\delta f_n \sqrt{\rho E} \tag{3b}$$

The calculated stress for Al-alloy, Brass and Steel are 157 MPa, 260 MPa and 442 MPa respectively. The next section introduces the hemi-toroidal shape stainless steel suspension alternator and discusses the advantages and disadvantages of this structure.

Table 1: Material mechanical strength and loss factor

Material	Yield Stress (MPa)	Ultimate Stress (MPa)	Loss factor
Al-alloy	35-500	100-550	10^{-4}-10^{-5}
Brass	70-550	200-620	10^{-2}-10^{-3}
Steel	280-1600	340-1900	10^{-4}
Rubber	1-7	7-20	10^{-1}-10^{-2}

4. Hemi-toroidal suspension structure of the linear Alternator

Early experiments used stainless steel, using a continuous toroid fig.1 (a), were found to have very high hoop stresses; these were relieved by introducing a plurality of slots in the hemi-toroid as shown in fig.1 (b). The suspension gap is sealed with latex or polymer in order to reduce the air flow and hence damping loss to achieve a high "Q" (i.e. high resonance) factor. A linear alternator with eight segment stainless steel hemi-toroidal shape suspension as shown on fig. 2 was built and tested in order to evaluate the stiffness and quality factor of the suspension. The coil former is attached to the free end of the suspension in order that the coil and suspension can vibrate up and down linearly when an external force is applied to the generator housing. The prototype was built using a standard Dai-Ichi (a Filipino loudspeaker company) magnet and coil from their 15" woofer range. Table 2 shows the details parameters of the alternator.

Fig.1. Hemi-toroidal shape suspension a) without slot b) with slot

Fig.2: Hemi-toroidal shape alternator.

Table 2: LA parameters of the hemi-toroidal shape

Parameters	Dimension
Coil resistance (Ω)	7
Force factor – Bl (T-m)	22
Suspension outer diameter (mm)	165
Suspension thickness (mm)	0.25
Moving mass (g)	95
Resonant frequency (Hz)	194

The static displacement of the suspension was measured for different mass as shown in fig. 3 using dial gauges in order to allow a calculation of the stiffness of the suspension. It can be seen that the displacement of less than 1 mm for 10 N force gives a calculated spring constant of 120 kN/m.

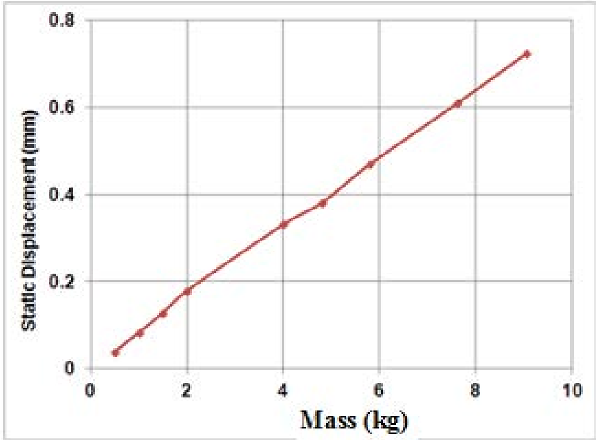

Fig. 3: Displacement Curve.

The quality factor ("Q") was measured initially without sealing the gap between the segments then after sealing the gap using latex material in order to see the variation of two cases. Fig. 4 shows the variation of impedance with changing frequency. The mechanical quality factor has been measured in free air using Thiele parameters [9];

$$Q_m = f_n \frac{\sqrt{\frac{Z_{res}}{R_c}}}{f_2 - f_1} \tag{4}$$

Where, f_n is the free air resonant frequency, Z_{res} is the resonant frequency impedance, R_c is the coil resistance, f_2 and f_1 are the upper and lower -3db cut-off frequency. The measured mechanical quality factors sealed and not sealed is 48 and 24 respectively. From these measurements it can be seen that the mechanical loss $R_m = m\omega_n / Q_m$) is reduced significantly when the gap between segments is sealed. As a result of the high stiffness, it required a high driving force to get sufficient displacement when the driving frequency was away from resonance. Without the slots the suspension is limited to 1 mm peak excursion level due to high hoop stress. The experimental results of the stainless steel design showed high hoop stress, failed after 10 minutes operation and was significantly stiffer than expected so is not suited to a large coil excursions.

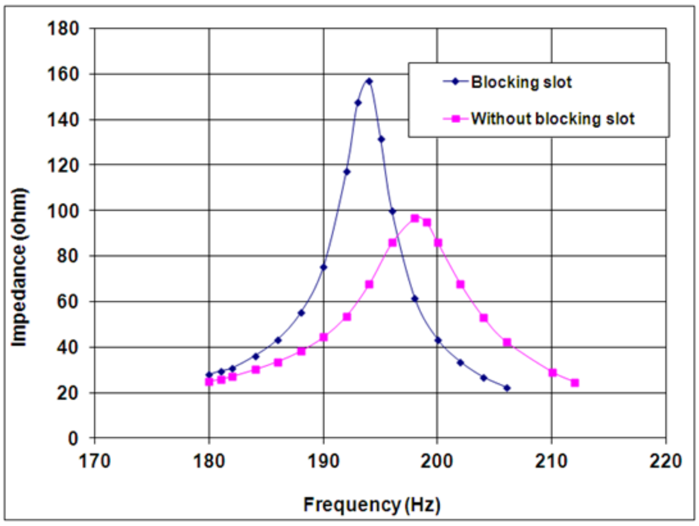

Fig. 4: Free air quality factor of the alternator with and without blocking slot.

In order to increase cyclic life, a segmented trapezoidal semi-circular shaped cantilever as shown on fig.5 is proposed utilising low-loss spring-steel due to its high fatigue life. The coil former is attached to the free end of the cantilever and the coil and suspension and again this suspension can vibrate up and down linearly when external pressure is applied. The proposed structure exhibits a large excursion and provides a high Q factor compared to the hemi-toroidal structure. The suspension was tested over an extensive time period at 5 mm peak excursion, with no failure. Future testing is limited to 5 mm until larger thermo-acoustic pressures are achieved from the target TAE. A double Halbach array LA with a 16 segment trapezoidal shape spring-steel suspension was built and tested to understand the acoustic to electrical efficiency. The measured mechanical quality factor and calculated efficiency are 81 % and 70% both of which are significantly higher compared with commercial loudspeakers.

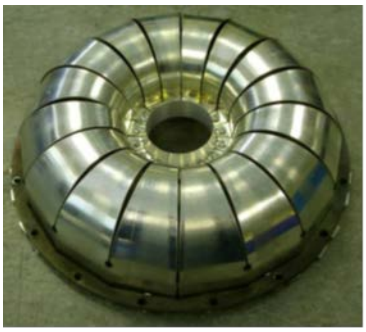

Fig.5. Segmented trapezoidal shape suspension.

5. Theoretical analysis of the semi-circular cantilever theory

Consider a trapezoidal semi-circular cantilever beam fixed at point A with a concentrated vertical load attached to the free end as shown in fig.6. In order to understand the stiffness of the beam it is necessary to calculate the displacement at the free end due to end load. If we let the semi-circular cantilever curve be contained in the *xy* plane, the small incremental area of the beam can be defined by [15];

$$\delta s = r.\delta\theta \tag{5}$$

Where r is the radius of the leaf arc and $\delta\theta$ is the small incremental angular displacement. Incremental angular displacement is defined by the following formula;

$$\delta\theta = \frac{M.\delta s}{EI} \tag{6}$$

Where EI is the flexural rigidity, E is the modulus of elasticity, I is the moment of inertia, M is the bending moment and δs is the small surface area.

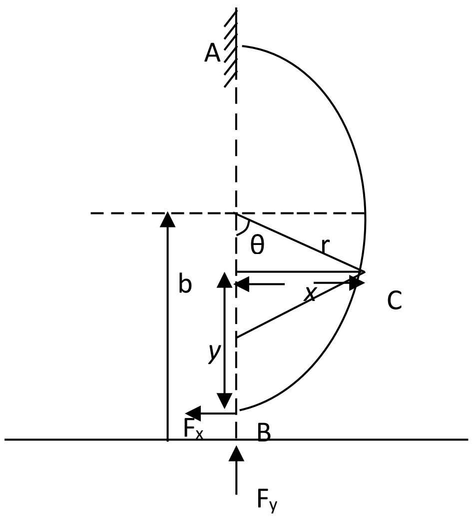

Fig.6: Semi-circular cantilever analysis

The parametric equations chosen to represent the circular arch geometry are defined by [15];

$$x = r\sin\theta \tag{7}$$

$$y = r(1-\cos\theta) \tag{8}$$

$$w = \frac{b\pi}{8}(1-\frac{r}{b}\cos\theta) \tag{9}$$

Where b is the semicircle centre distance from the neutral axis.

The moment of inertia about the neutral axis is given by [15];

$$I = \frac{wt^3}{12} = \frac{\pi.b}{96}(1-\frac{r}{b}\cos\theta)t^3$$

Let F_x and F_y be the the applied load at the free end in *X* and *Y* direction. Therefore the deflection V_x, V_y and V_θ along the *X*, *Y* and angular direction can be expressed by;

$$G\begin{pmatrix} r^2A & r^2B & rC \\ r^2B & r^2D & rE \\ rC & rE & F \end{pmatrix}\begin{pmatrix} F_x \\ F_y \\ M_B \end{pmatrix} = \begin{pmatrix} V_x \\ V_y \\ V_\theta \end{pmatrix} \tag{10}$$

The details calculation for A, B, C, D, E, F and G can be found in appendixes.

6. Implementation of the Trapezoidal semi-circular shape LA suspension

To design the alternator suspension to achieve sufficient excursion from the driving force of the thermos-acoustic engine without hitting an end-stop, so the hemi-toroidal concept was modified to a trapezoidal semi-circular shape using sixteen segments as shown in fig. 7. To evaluate the theory a spring-steel trapezoidal cantilever leaf beam as shown in figure 5 was built and tested. Table III shows the leaf parameters. The static displacement at the free end of spring was measured for differently loaded mass. It can be seen from Figure 8 that 17 mm displacement is achieved from 16.5 N forces and the measured spring constant is 0.96 N/mm in vertical direction. The calculated vertical force for 1 mm displacement using equation (10) is 1.069 N which is equivalent to a spring constant of 1.069 N/mm. Thus the experimental result agrees with the theoretical value. It can be seen from the measured results that the stiffness of the trapezoidal shape is significantly lower than the hemi-toroidal shape.

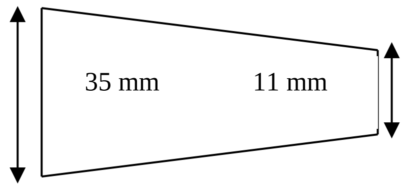

Fig 7: Double leaf trapezoidal shape semi-circular cantilever.

Table 3: Dimension of the leaf parameters for the semi-circular cantilever.

Parameters	Dimension (mm)
Fixed end width	35
Free end width	11
Beam thickness	0.7
Arc diameter	58

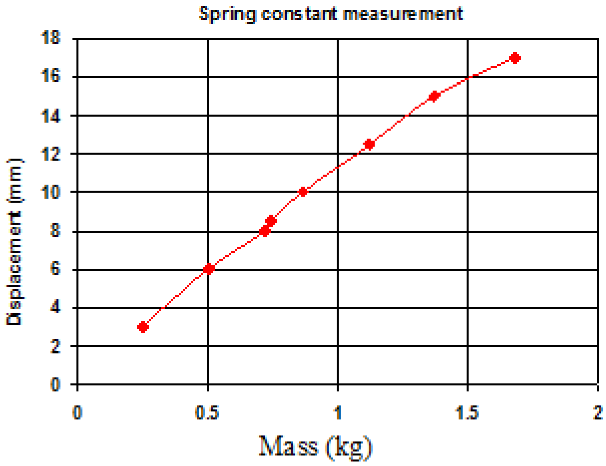

Fig 8: Measured displacement for different mass using dial gauge.

7. Conclusions

The hemi-toroidal and trapezoidal semicircular shape suspension of the linear alternator are presented in this paper. The hemi-toroidal shape linear alternator was been built and tested using a standard loudspeaker magnet and coil. The measured results of the hemi-toroidal shape suspension showed that the high stiffness and high hoop stress of this structure would not be reliable for a high excursion moving coil linear alternator. The theoretical analysis of the trapezoidal semicircular cantilever suspension was described and the theory was verified via experimental results. The trapezoidal suspension produced better outcomes, with predicted stress levels below the fatigue limit at +/- 18mm excursion. Testing at +/-5mm was successful in terms of zero failure rate. Therefore measured results indicated that the trapezoidal shape is a suitable option as a replacement for the hemi-toroidal shape in the high excursion linear alternator due to lower stiffness and improved/lower hoop stress factor.

Acknowledgment

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References

[1] www.score.uk.com

[2] Liu J, S Garrett. Characterization of a small moving magnet electrodynamic linear motor. Journal of Acoustical Society of America October 2005; 118(4).

[3] Riley P. Designing a low-cost electricity-generating cooking stove for high-volume implementation. PhD thesis, 2014, University of Nottingham.

[4] Boldea I, Nasar S A, “Permanent-magnet linear alternators-part II: design guidelines, IEEE Transactions on Aerospace and electronic Systems 1987; 23(1): 79-82.

[5] Wang J, West M, Howe D, La Parra H.Z-D, Arshad W M. Design and experimental verification of a linear permanent magnet generator for a free-piston energy converter. IEEE transactions on Energy Conversion 2007; 22(2): 299-305.

[6] Faiz J, Rezaeealam B, Yamada S. Reciprocating Flux-Concentrated Induction Generator for Free-Piston Generator. IEEE Transaction on Magnetics 2006; 42 (9).

[7] Riley P, Johnson M. Generating electricity in developing countries using thermo-acoustics powered by burning wood. Acoustics 08; Paris June 2008.

[8] Yu Z, Saechan P, Jaworski A. A method of characterising performance of audio loudspeakers for linear alternator applications in low-cost thermo-acoustic electricity generators. Applied Acoustics 2011; 72: 260-267.

[9] John Borwick, Loudspeaker and Headphone Handbook, Focal press, Second edition.

[10] Saha C, Riley P, Yu Z, Jaworski A, Johnson C M. Halbach array linear alternator for thermo-acoustic engine. Sensor and Actuators A Physicals: 2012; Volume 178, 179-187.

[11] Yu Z, Backhaus S, Jaworski A. Design and testing of a travelling-wave looped-tube engine for low cost electricity generators in remote and rural areas. American Institute of Aeronautics and Astronautics 2009.

[12] William D. Callister Jr. Fundamentals of Materials Science and Engineering, Fifth Edition, the University of Utah.

[13] Lee T, Lakes S R. Damping Properties of Lead Metaniobate. IEEE Transaction on Ultrasonics. Ferroelectrics and Frequency Control 2001; 48.

[14] Andrew, D., & Dimarogonas, S. H. Vibration for engineering. Prentice-Hall International editions, chapter 12, page 605.

[15] Warren C Y. Roark’s Formulas for Stress and Strain. 6th Edition, McGraw-Hill International.