

An Ironless Large Displacement Flat Piston Loudspeaker

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This paper presents a small wide-band loudspeaker. Particular efforts have been made to reduce the nonlinearities of the loudspeaker as much as possible. The motor structure is completely ironless, the elastomer suspensions are replaced by ferrofluid seals and a monobloc carbon foam piston substitutes the traditional conic membrane. The circular radiating surface, which is flat, has a diameter equal to only 2 cm. Therefore, in order to obtain a sufficient sound pressure level at low frequencies, large displacements of the piston are necessary. After a detailed description of each part of the loudspeaker, theoretical results of the expected performances of this transducer are given.

Keywords: loudspeaker, ironless, flat piston, large displacement.

1. Introduction

The electroacoustic transducer that is presented in this paper has been designed to cover most of the audio bandwidth as well as being as small and light as possible. Sound reproduction accuracy was also an important criterion. Therefore, sources of nonlinearities have been eliminated as much as possible. That is why there is no iron in the motor, elastomer suspensions are replaced by ferrofluid seals and a monobloc cylinder-shaped piston is used instead of a conic membrane. Moreover, the radiating surface of this piston is flat. In order to obtain a reasonable sound pressure level at low frequencies with a small radiating surface, the latter must accept large displacements.

2. The loudspeaker structure

The structure of the studied loudspeaker is presented in Fig. 1. It is composed of an annular ironless motor and a monobloc piston. The voice coil is directly wound around the semi-height of the piston. The guidance is achieved using two ferrofluid seals.

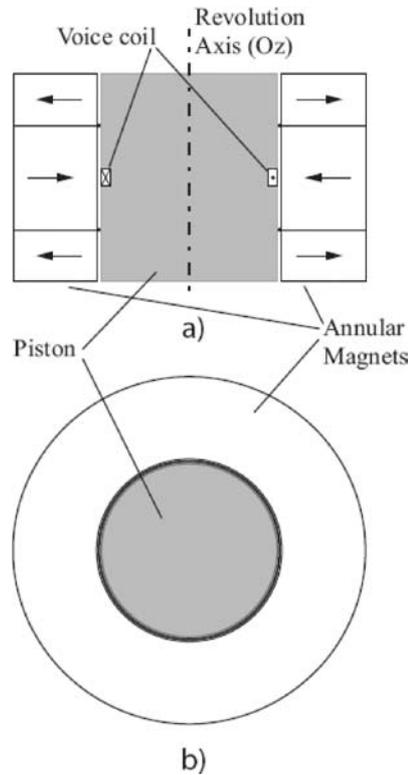


Fig. 1. (a) Cross-section and (b) Top view of the loudspeaker.

2.1. Motor

Traditional electrodynamic loudspeaker motors present a number of well-known drawbacks [1–4]. Mainly, the presence of iron in such motors leads to several kinds of nonlinearities. These include eddy currents, the magnetic saturation of the iron and the variation of the coil inductance with its position causing a reluctant effect. However, it is desirable for the force applied to the moving part to be an image of the driving current. The driving forces applied to the moving part of the loudspeaker can be written as

$$F_{\text{driv}} = F_L + F_r = Bli + \frac{1}{2} \frac{dL}{dx} i^2, \quad (1)$$

where F_L is the Laplace force, F_r the reluctant force, B the induction seen by the voice-coil, i the driving current flowing through the coil, l the length, L the inductance and x the displacement of the coil. Thus, Eq. (1) shows that if the inductance of the coil varies, a reluctant force, proportional to i^2 , occurs and interferes with the Laplace force.

This reluctant force creates a force distortion resulting directly in an audible acoustical distortion. In order to solve these problems, several structures of ironless voice-coil motors have already been proposed [5–10]. That is why the structure of this loudspeaker is made totally of sintered permanent neodymium magnet rings. With such structures, the inductance of the coil no longer depends on its position. This results in the disappearance of the reluctant force and the other nonlinearities due to iron previously listed. In addition, the inductance is diminished and consequently, so is the electrical impedance, especially at high frequencies.

The structure has been designed to obtain a radial magnetic induction that is as linear as possible on the whole voice coil path. The motor is composed of three radially magnetized, permanent magnet rings. The middle ring and the other two, top and bottom rings, are magnetized in opposite directions. The middle ring creates the useful magnetic field that is seen by the voice coil on its path whereas the top and bottom magnets help to guide the magnetic flux and thus, reduce the leakage in the air.

2.2. Moving part

The moving part is composed of a monobloc piston on which the voice coil is directly wound.

2.2.1. Piston

Traditional loudspeakers use a conic membrane typically made of paper. The inconvenience of such membranes is the occurrence of mechanical modes in the audio bandwidth, resulting in a loss of energy and a perturbation of the sound reproduction at some frequencies corresponding to the radiating mechanical modes. The loudspeaker presented here is novel in its use of a piston that is a monobloc full cylinder made of carbon foam. This material has the advantage of being very stiff and light. The shape and the material used enable the frequency of the first mechanical mode to be moved upwards, hopefully out of the audio bandwidth, depending on the size of the piston.

Another characteristic of the piston is that it has a flat circular radiating surface. This enables to fit as well as possible the approximation of circular piston used in the model. This choice is reinforced by the conclusions of QUAEGEBEUR'S work [11].

2.2.2. Suspension

Suspensions are usually a source of nonlinearity, especially for loudspeakers requiring large displacements of the membrane. The magnetic field created by this

motor structure presents a high gradient at the junction between the magnet rings. This high magnetic field gradient permits the use of ferrofluid seals to guide the moving part [12–17]. Ferrofluid seals allow the piston to be guided with a negligible stiffness in the axial direction. Thus, the suspension stiffness is given by the cabinet placed behind the speaker. Furthermore, ferrofluid seals also have a role of thermal bridge, allowing the heat created by the voice-coil to flow through and be dissipated in the motor.

3. Theoretical results

The whole design of the loudspeaker has been done with a diameter of the piston equal to 2 cm and a axial displacement of ± 4 mm.

3.1. Motor specifications

The motor is designed to create a radial magnetic induction as linear as possible over the whole path of the voice coil, i.e. 8 mm. The radial component of the magnetic field is calculated analytically in 3D, taking into account the magnetic pole volume density [18, 19]. The calculations results are presented in Fig. 2. In order to obtain the intended magnetic field, the total motor dimensions are:

- inner diameter: 21 mm,
- outer diameter: 40 mm,
- height: 24 mm.

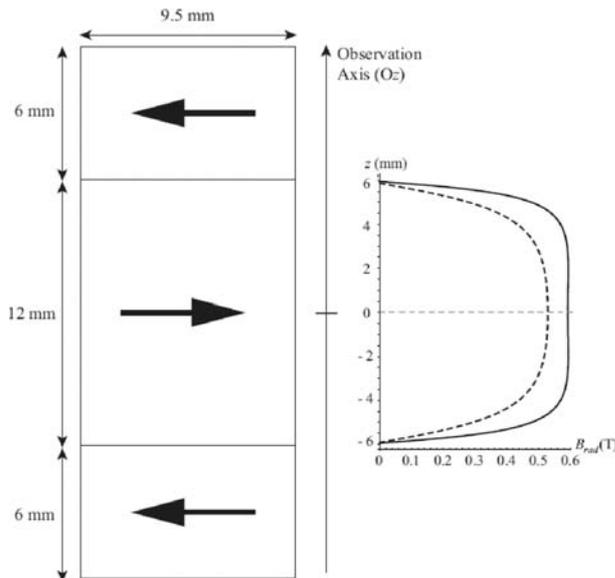


Fig. 2. Motor assembly and radial component of the magnetic field, B_{rad} (T), seen by the coil on its path, 0.5 mm (solid line) and 1.2 mm (dashed line) away from the motor.

The calculations are performed considering the use of Neodymium Iron Boron (NdFeB) magnets with a remanence, B_r , equal to 1.47 T. The density of such material being $7.4 \text{ kg}\cdot\text{m}^{-3}$, the motor weighs 160 g.

3.2. Moving part specifications

3.2.1. Piston specifications

The piston is designed to be as tall as the motor, i.e. 2.4 cm, and is made of a carbon foam from GrafTech International. All the calculations are done with the lightest model available, having a density of $30 \text{ kg}\cdot\text{m}^{-3}$. Thanks to this very light material, the piston weighs only 0.2 g.

Another interesting characteristic of this structure is that the mechanical modes of the moving part appears at frequencies much higher than for a traditional loudspeaker. The Young's modulus of the chosen carbon foam is $4\cdot 10^7 \text{ Pa}$. A finite element calculation has been run to determine the first mechanical modes of the piston; the first one appears at 14.7 kHz and the first radiating mechanical mode at 22.6 kHz, as shown in Fig. 3. The first three modes correspond to torsion modes and the fourth one to a breathing mode. As a consequence, the acoustic radiation should not be disturbed within the audible frequency range.

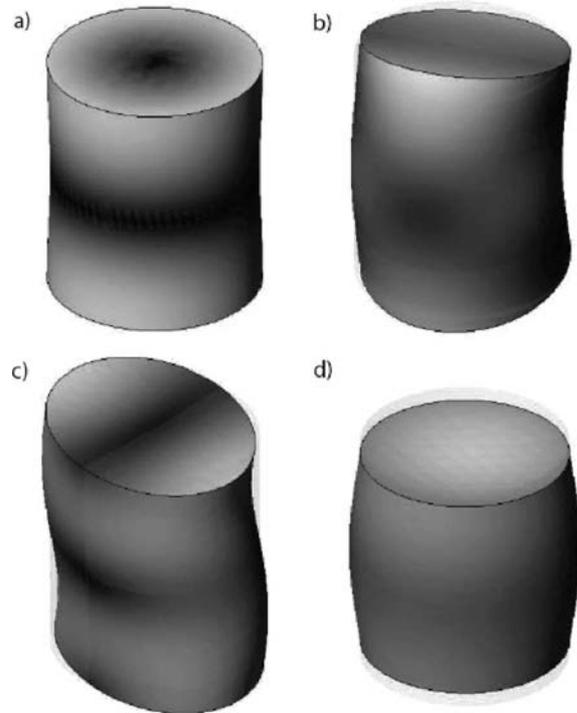


Fig. 3. First four mechanical modes of the piston: a) 14.7 kHz, b) 18 kHz, c) 22.3 kHz, d) 22.6 kHz.

4. Voice coil specifications

A small furrow is cut off all around the piston in order to wind the coil. As a result, the outer radius of the coil is equal to the radius of the piston, as shown in Fig. 4. The voice coil is designed to be as heavy as the piston, i.e. 0.2 g. The chosen voice coil is an aluminium wire whose characteristics are:

- diameter: 0.1601 mm,
- linear resistance: $1.3847 \Omega \cdot \text{m}^{-1}$,
- linear mass: $0.0547 \text{ g} \cdot \text{m}^{-1}$.

Thus, 4.1 m of this wire are necessary, which corresponds to about 66 turns, a resistance of 5.7Ω and a force factor, Bl , of $2.4 \text{ T} \cdot \text{m}$. Its inductance is calculated using the M. BROOKS and H.M. TURNER formula expressed as [20]:

$$L = 4\pi^2 \frac{a^2 n^2}{b + e + R} F' F'' \quad (2)$$

with

$$F' = \frac{10b + 12e + 2R}{10b + 10e + 1.4R} \quad (3)$$

and

$$F'' = 0.5 \log \left(100 + \frac{14R}{2b + 3e} \right), \quad (4)$$

where a is the mean radius of the coil, R its outer radius, b the width of the coil winding, e its thickness, d the diameter of the coil wire and n the number of turns, as shown in Fig. 4. In this case, the theoretical inductance of the voice coil is equal to 0.13 mH.

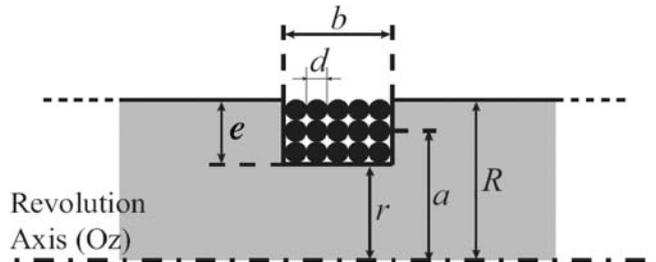


Fig. 4. Voice-coil calculation model parameters.

4.1. Cabinet specifications

Since the axial stiffness of the ferrofluid seals is almost null, the suspension stiffness is given by the air volume in the cabinet at the back of the piston. The size of the cabinet is determined in order to obtain a resonance frequency, f_r , of the moving part equal to 60 Hz. This volume, V_c , is given by:

$$V_c = \frac{\rho_a c^2 S_p^2}{4\pi^2 M_{\text{piston}} f_r^2}, \quad (5)$$

where ρ_a is the air density, c the sound velocity, S_p the surface of the piston and M_{piston} the mass of the moving part. The numerical application gives:

$$V_c = 0.25 \times 10^{-3} \text{ m}^3. \quad (6)$$

The equivalent mechanical compliance, C_{ms} , is then expressed as

$$C_{ms} = \frac{V_c}{\rho c^2 S_p^2} \quad (7)$$

and equal to

$$C_{ms} = 1.5 \times 10^{-2} \text{ s}^2 \cdot \text{kg}^{-1}. \quad (8)$$

4.2. Acoustic radiation

Using all the parameters described previously, a flat piston model is used to evaluate the expected acoustic pressure radiated by the loudspeaker at 1 m for 1 W. The only unknown is the exact value of the mechanical resistance, R_{ms} . This parameter will depend on the quantity and the viscosity of the ferrofluid seals. The electrical impedance and the acoustic pressure are calculated for four different values of R_{ms} : 0.02, 0.05, 0.1 and 0.5 $\text{kg} \cdot \text{s}^{-1}$. The results are presented in Fig. 5.

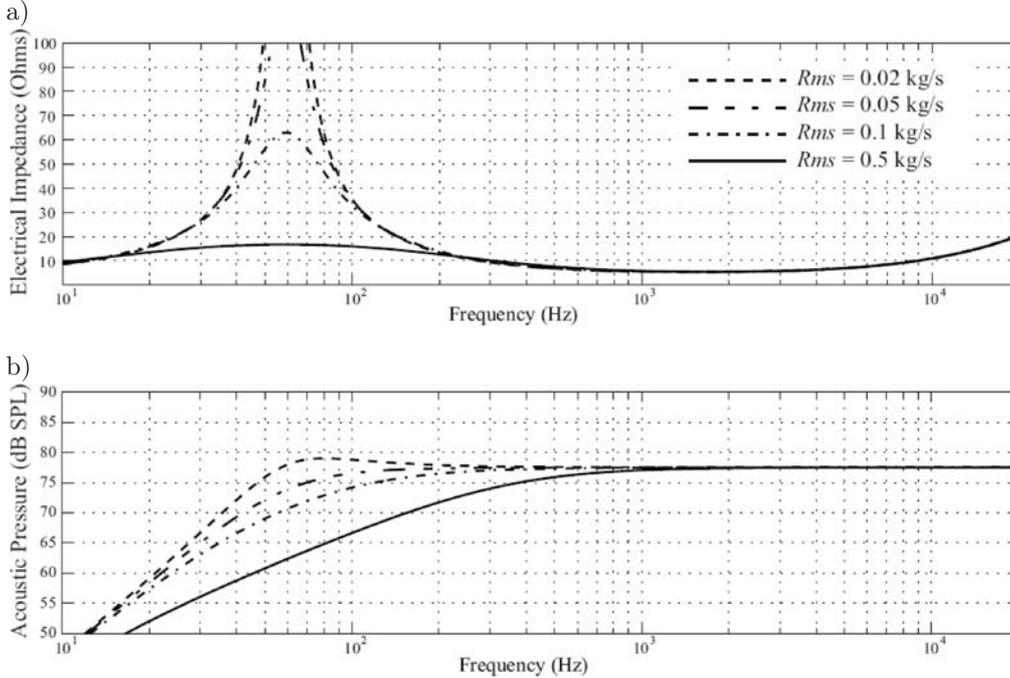


Fig. 5. (a) Electrical impedance and (b) Acoustic pressure radiated by the loudspeaker for 1 W at 1 m as a function of frequency.

These results show that the value of the mechanical resistance has a direct impact on the low frequency response of the loudspeaker. Therefore, particular care must be taken as for the quantity and the type of ferrofluid that will be chosen.

5. Conclusions

A small wide-band loudspeaker devoid of most sources of nonlinearities has been described. First calculations are quite promising for the outcome of the actual loudspeaker. The constant force factor, the absence of iron in the motor, the flat radiating surface and the lack of radiating mechanical modes within the audible bandwidth are the starting points of an accurate loudspeaker. Some prototypes should be realized in the coming year in order to validate these theoretical results.

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