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Stiffness and damping characterization of ferrofluid seals on the new loudspeaker design

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The aim of this paper is to determine experimentally the behavior of the ferrofluid seals used in loudspeaker having no viscoelastic suspensions [1]. In this case, the ferrofluid is saturated by a strong magnetic field (300-500 kA/m) relatively to its magnetization (9-32 kA/m). Two parameters are investigated: the radial stiffness that centers the moving part and the axial damping influencing the dynamic response of the loudspeaker. These parameters are measured as function of ferrofluid volume, magnetization, viscosity, shear rate and frequency of oscillation. The behavior is found coherent with the results of literature for relatively weak magnetic fields.

1 Introduction

One major drawback of classical loudspeakers comes from the non linearity of their viscoelastic suspensions. In order to overcome these limitations, a new loudspeaker design was proposed [2]. Its structure is composed by a set of permanent magnets and the moving part is an amagnetic cylindrical piston (Fig. 1.a). In this case, viscoelastic suspensions are replaced by ferrofluid seals which center the moving part of the loudspeaker. For relatively weak magnetic field, the ferrofluid is not saturated: its properties are function of the magnetic field amplitude [3]. For example, the viscosity increases with the magnetic field [4]. In the case of loudspeaker application [5] the magnetic field is strong enough (300-500 kA/m) to saturate the ferrofluid.

The aim of this paper is to investigate experimentally the equivalent macroscopic behavior of ferrofluid seals in the case they are saturated by a high magnetic field. Their behavior is modelised by a static stiffness k in the radial direction $\mathbf{r}$ and a viscous coefficient c in the axial direction $\mathbf{z}$ (Fig.1.b). In a first part, the evolution of the static radial stiffness with the seal volume and the magnetization of ferrofluid is studied. The second part focuses on the dynamic behavior of the seals by measuring the axial viscous coefficient as function of the ferrofluid characteristics, the frequency of oscillation and the ferrofluid volume.

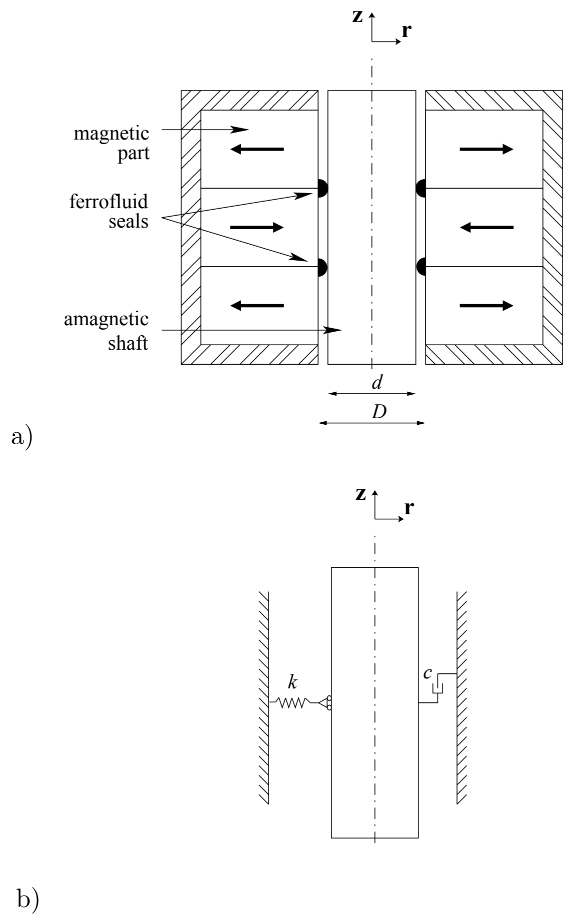


Figure 1: a) Schematic diagram of the dynamic measurements; b) Mechanical representation.

2 Static radial stiffness

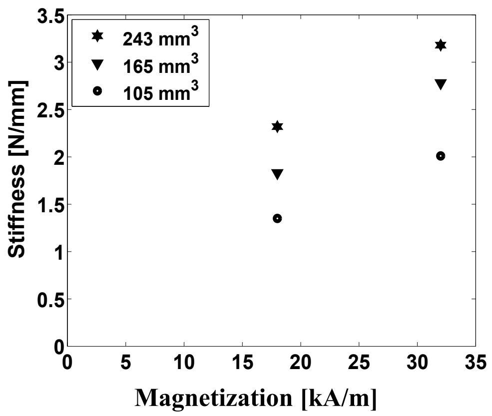
The radial stiffness of the ferrofluid seals is evaluated in the following configuration. The diameter d of the shaft is 49.0 mm and the inner diameter D of the magnetic part is 49.7 mm, so the thickness of the ferrofluid seal is 0.35 mm. Only one seal is used to simplify the measurement procedure.

To determine the radial stiffness, the seal is crushed by the shaft in the radial direction: displacement δ_r of the shaft and the associated force F_r are measured. Some more details on the experimental procedure can be found in reference [6]. The static stiffness is then given by

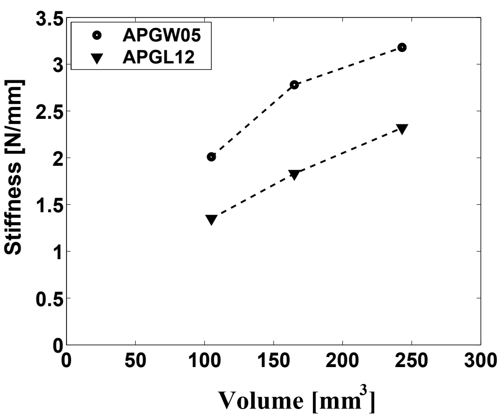
$$k = F_r/\delta_r. \quad (1)$$

The influence of the seal volume and the magnetization are investigated (Fig.2). Measurements are performed with three volumes of ferrofluid: V_1 =105 mm^3, V_2 =165 mm^3 and V_3 =243 mm^3. Two types of ferrofluid of different magnetization and same viscosity are used: APGW05 and APGL12 (Table 1).

Figure 2.a shows that the static radial stiffness is proportional to the ferrofluid magnetization. Figure 2.b shows that the static radial stiffness increases with the seal volume but is not proportional. Indeed, the stiffness depends on the magnetic field amplitude where the free surface of the seal is located. In the present configuration, the magnetic field is greater in the vicinity of magnet junctions. Adding ferrofluid moves the free surface in a weaker magnetic field area, so the increase of the stiffness slows down with the volume of ferrofluid. Magnetic iso-pressure curves in the area of the seal are presented in Fig.1 of reference [6].



a)



b)

Figure 2: Experimental static radial stiffness versus a) the ferrofluid magnetization, b) the volume of the ferrofluid seal.

3 Dynamic axial damping

3.1 Loudspeaker configuration

The axial viscous damping of the ferrofluid seals is evaluated in a different configuration (Fig. 3). The inner diameter of the magnetic part is $D = 21.0$ mm. The amagnetic shaft is a plexiglas cylinder of diameter $d = 20.4$ mm. The thickness of the seals is $e = 0.3$ mm.

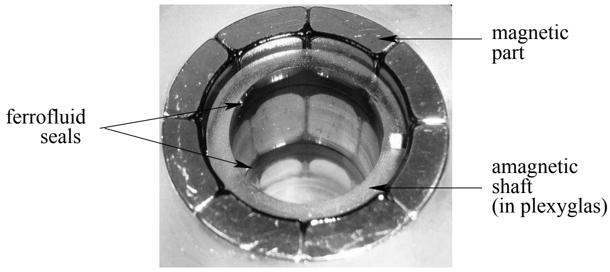


Figure 3: Experimental setup of loudspeaker configuration.

Tests are performed using a visco-analyser (DMA-2980 TA-Instruments) on which the static part is clamped. A sinusoidal displacement of the piston in the axial direction is achieved at a frequency of oscillation ω. The amplitude of the displacement is kept constant at 40 μm for all configurations. The complex frequency response function $H(\omega)$, excitation force over axial displacement, is then estimated. From $H(\omega)$, it is possible to get the viscous damping coefficient c. Indeed the frequency response function writes

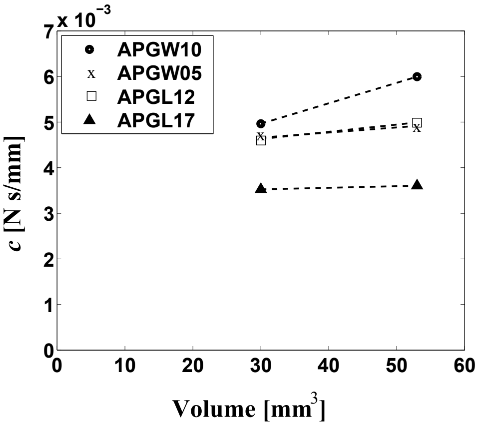
$$H(\omega) = (k - \omega^2 m) + j\omega c, \tag{2}$$

with k the stiffness in the axial direction and m the mass of the moving part (piston and added mass induced by the ferrofluid). The damping coefficient is thus given by,

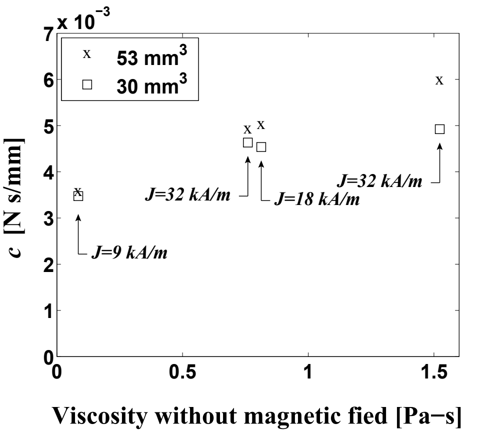
$$c = \Im(H(\omega))/\omega. \tag{3}$$

It is determined for four types of ferrofluid of different magnetizations and viscosities (Table 1), two volumes (V_1=30 mm^3 and V_2=53 mm^3) and three frequencies (40 Hz, 60 Hz and 80 Hz). Figure 4.a shows the results at 80 Hz: the increase of the volume induces only a small increase of the damping coefficient. Figure 4.b shows the influence of viscosity without magnetic field ν_o at the same frequency: the damping coefficient c increases with the viscosity, but without proportionality. This increase behavior is also observed for weak magnetic field level [4, 3]. On the other hand, the increase of ferrofluid volume affects slightly the damping. Assuming that the viscosity increases with the intensity of the magnetic field, it is maximal in the vicinity of the highest magnetic field zone. Adding ferrofluid far from this zone will have only a slight effect. Note that the damping effect is not influenced by the ferrofluid magnetization.

Figure 5 shows the influence of the frequency of oscillation and the shear rate. The seals volume is hold constant at V=30 mm^3. It is shown that the damping decreases with the frequency of oscillation. This is consistent with the magnetoviscous effects in ferrofluid described by Odenbach and Raj [7] for weak magnetic field ($H \leq 30$ kA/m). In presence of magnetic field, chain-like structures are formed in the ferrofluid. These tends to be broken with the increase of shear rate inducing a decrease of viscosity. In our case, once the amplitude of the displacement and the thickness of the ferrofluid seals are kept constant, the shear rate is directly proportional to frequency. Viscosity decreases with increasing



a)



b)

Figure 4: Axial viscous damping (f = 80 Hz) versus a) the volume of the seals, b) the ferrofluid viscosity.

of shear rate and of frequency of oscillation is verified for ferrofluid in high magnetic field.

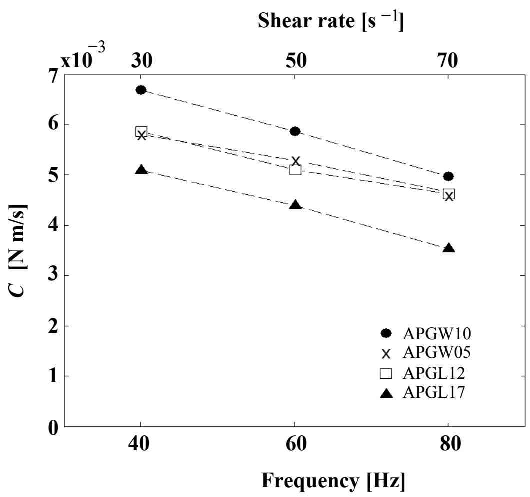


Figure 5: Influence of frequency on the damping effect of ferrofluid seals ($V = 30\text{mm}^3$).

3.2 Parallel plate configuration

In order to separate frequency of oscillation and shear rate effects on the viscosity variation of ferrofluid, the following experimental setup is actually used. Consider for the lower ferrofluid seal of Fig.6 two rigid, horizontal, parallel and uniform plates separated by a ferrofluid film. The lower plate is fixed and the upper plate oscillates under a sinusoidal excitation with constant amplitude. The impedance head measures the acceleration of the moving plate $\ddot{x}$ and the force F to move it. The velocity of the moving plate $\dot{x}$ can be calculated by the integration of $\ddot{x}$. The shear rate maximum can be seted by the ratio of the amplitude of excitation velocity to the gap. An external magnetostatic field can be applied in perpendicular direction by two permanent magnets. Its level is controlled by the distance between the moving plate and the magnets and assumed uniform in the gap. The influence of the shear rate and the frequency of oscillation on the viscosity variation of ferrofluids in the presence of high magnetic field will be presented in the future work.

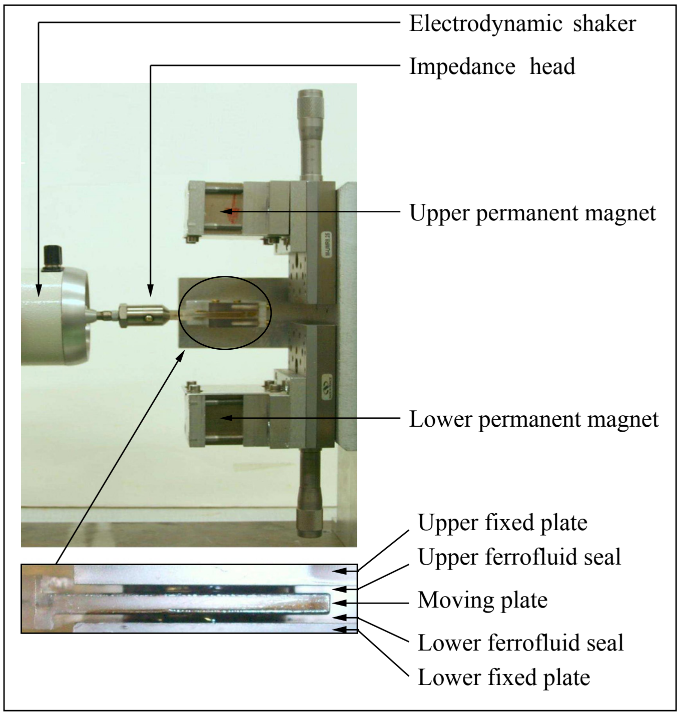


Figure 6: Experimental setup of parallel plate configuration.

Ferrofluid (Ferrotec)	Magnetiz. Saturation J [kA/m]	Nominal viscosity ν_o [mPa-s]	Measured viscosity ν_o [mPa-s]
APGW10	32	1000	1530
APGW05	32	500	720
APGL12	18	500	780
APGL17	9	60	82

Table 1: Characteristics of the used ferrofluids: viscosity values without magnetic field.

4 Conclusion

Radial static stiffness and axial viscous damping of ferrofluid seals in the presence of a high magnetic field have been measured. The results confirm that the stiffness is proportional to the magnetization. The viscous damping increases with the viscosity of ferrofluid (measured without magnetic field) and is not influenced by the ferrofluid magnetization. Both quantities increases with the volume of ferrofluid. However the increase slows down with the volume of ferrofluid. These analyses are coherent with the literature for relatively weak magnetic fields. Experimental analysis will be made to separate the effects of the shear rate and the frequency of oscillation on the viscosity variation of ferrofluids.

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