STABLE MAGNETIC SUSPENSION IN FLUIDS USING TUNED LC-CIRCUITS

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ABSTRACT

The use of tuned LC-circuits in conjunction with active magnetic bearings offers semistable suspension of objects without any sensors. In order to achieve a stable behaviour, however, it is necessary to implement a damping mechanism into the electromechanic suspension system. Since active magnetic bearings are widely used for fluid applications, the inherent squeeze film damping in the gap between stator and rotor can be employed for the dynamic stabilisation of the entire system. In this paper it is shown by experiment that the combination of tuned LC-circuits and squeeze film damping is appropriate for such active magnetic bearing applications and can be realized at reasonable costs.

INTRODUCTION

In most magnetic bearing applications a highly sophisticated technology is used for the magnetic suspension of objects. This might be due to the fact that scientists like to apply the latest technology for special and challenging applications. Nevertheless, for some of these applications it seems to be justified to utilise modern control achievements and technology, at least if this procedure can help to encounter more technical challenges. In many applications, however, it might be an academic overshoot and does not reflect the practical necessities.

Compared to the state of the art, tuned LC-circuits represent a simple approach to the magnetic suspension problem. Unfortunately, they lack of stability, because the unstable magnetic system is only changed into a semistable one. Roughly speaking, for a simple magnetic bearing model the poles on the real axis of the complex pole plane are moved onto the imaginary axis by the proportional gain brought in by the tuned LC-circuit. Therefore, the system needs additional damping. The most simple way to cope with this problem is to involve a complex control concept, e.g. by estimating the rotor position and by modulating the amplitude of the sinusoidal excitation signal.

In this paper a simple but pragmatic solution for the stable operation of tuned LC-circuits in active magnetic bearings is chosen. Since many magnetic bearing applications lie in the field of pumping of sensible fluids, the damping properties of this fluid can be employed to stabilise the entire system, if the design is appropriate. The result is a simple and cheap bearing system for the magnetic suspension of objects surrounded by fluids.

TUNED LC-CIRCUITS

The magnetic suspension of an object at a constant magnetomotive force is unstable by its nature, because the magnetic resistance increases as the object is separated from the magnet which decreases the flux and hence the magnetic force, and vice versa. This is true for a permanent magnet as well as for an electromagnet whose coil current is constant. Alternatively, this behaviour can be inverted, if the flux of a magnet is generated by a coil in conjunction with a capacitor within a LC-circuit driven by a sinusoidal voltage signal. A simple model of such a system can be seen in Fig. 1.

The LC-circuit consists of the inductance $L(l)$ as function of the air gap width l of the electromagnet itself and a capacitor C in series, both determining the resonance frequency. Since this circuit is excited by a sinusoidal voltage signal with the amplitude E and frequency ω_e, the current amplitude and hence the magnetic force depend further only on the air

gap width.
An increasing air gap will decrease the inductance and thus increase both the resonance frequency and the magnetic resistance. The exciting frequency is then chosen such that the circuit approaches resonance as the air gap increases, which leads to increasing current amplitude. Furthermore, the current magnitude depends on the relative damping of the electric circuit determined by its effective resistance R. If then the increasing coil current can compensate the increasing magnetic resistance, the resulting magnetic force will increase, too, and the unstable mechanism of an ordinary electromagnet is inverted. Thus, using a tuned LC-circuit the electromagnet acts like a mechanical spring whose properties depend on electrical and mechanical parameters, i.e. the tuned LC-circuit statically stabilises the system. In order to be entirely stable, the semistable magnetic suspension system needs additional damping.

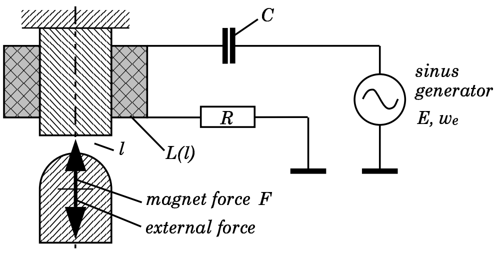

FIGURE 1: *Principle of the magnetic suspension using tuned LC-circuits.*

TUNED LC-CIRCUITS FOR AMB

The tuned LC-circuit principle can be used for all magnetic bearing applications, if the necessary damping is generated by an additional electronic device or controller. However, if the magnetic bearing is used for an application with fluids, e.g. in a fluid pump, the fluid itself provides a damping mechanism. In most of these applications the fluid to be conveyed flows through the bearing gap. If not, this can be achieved by an appropriate mechanical design. Thus, a damping of vibrations perpendicular to the gap surface can be expected at a certain damping ratio depending on the geometry. The question then is, how much damping can be provided by the fluid, and how much damping is necessary to reach a desired system behaviour at a given stiffness.

Stiffness

The stiffness of the magnetic bearing using tuned LC-circuits is determined by its design parameters and the point of operation. In order to derive a stiffness for the tuned LC-circuit system it is necessary to make some simplification.
Since the excitation frequency is far beyond the eigenfrequencies of the mechanical system, it is assumed that the inductance does not depend on time, but only on the air gap width l. Therefore, it is sufficient to regard only the amplitude of the harmonic signal and its effective value. Using this assumption, the amplitude of the magnetic force F acting on a suspended object in Fig. 1 is

$$F = \frac{1}{2\,\mu_0\,A}\,\Phi^2, \tag{1}$$

with the flux amplitude Φ, a nominal cross section area A and the magnetic permeability for vacuum μ_0. The flux depends on both the inductance $L(l)$ which is assumed to be time invariant for slow motion, and the current amplitude I such that

$$\Phi = \frac{1}{N}\,L(l)\,I, \tag{2}$$

with N being the number of windings. The amplitude I of the coil current is determined by the transfer function of the tuned LC-circuit as

$$\begin{aligned} I &= \text{abs}\left(G(s, L(l))\big|_{s=i\,\omega_e}\right)\,E \\ &= A_e(L(l))\,E, \end{aligned} \tag{3}$$

at the chosen frequency ω_e and voltage amplitude E with the transfer function

$$G(s, L(l)) = \frac{C\,s}{L(l)C\,s^2 + RC\,s + 1}. \tag{4}$$

Hence the amplitude of the magnetic force can be put as

$$F = k_F\,L(l)^2\,A_e(L(l))^2\,E^2, \tag{5}$$

with k_F being a magnet constant for the chosen geometry such that

$$k_F = \frac{1}{2\,\mu_0\,A\,N^2}. \tag{6}$$

Note that the magnet force is modulated by the square of sinus, although the exciting signal is simply sinusoidal. This will influence its root mean square by the factor $\sqrt{3}/2$. The stiffness parameter of the magnetic bearing using a tuned LC-circuit can then be easily calculated as partial derivative of the effective magnet force F_{eff} at the point of operation l_0 with

$$\begin{aligned} k_l &= \left.\frac{\partial\,F_{eff}}{\partial\,l}\right|_{l=l_0} = \left.\frac{\sqrt{3}}{2}\frac{\partial\,F}{\partial\,l}\right|_{l=l_0} \\ &= \sqrt{3}\,k_F\,A_e(L(l_0))\,L(l_0) \\ &\quad \times\,[A_e(L(l_0))\,k_L + L(l_0)\,k_A]\,E^2, \end{aligned} \tag{7}$$

with

$$k_L = \left.\frac{\partial L}{\partial l}\right|_{l=l_0} \quad \text{and} \tag{8}$$

$$k_A = \left.\frac{\partial A_e}{\partial l}\right|_{l=l_0} = \left.\frac{\partial A_e}{\partial L}\right|_{L=L_0} \left.\frac{\partial L}{\partial l}\right|_{l=l_0}. \tag{9}$$

If the magnetic bearing should have the behaviour of a spring, then k_l has to be positive. Thus, the term in square brackets of Eqn. (7) has to be positive, too, which yields

$$\left.\frac{\partial A_e}{\partial L}\right|_{L=L_0} = k_{AL} > -\frac{A_e(L(l_0))}{L(l_0)}. \tag{10}$$

This means that the stiffness of the magnetic bearing is determined by the constant k_{AL} which depends on the transfer function $G(s, L(l))$, i.e. on the initial inductance $L(l_0)$, the capacitor C, the resistance R and the exciting frequency ω_e. Although these parameters can only be varied within narrow bounds, there is plenty of space for an appropriate control design.

The stiffness which can be achieved using tuned LC-circuits is not very high and is determined by design parameters. In many bearing applications, however, the achievable stiffness may be sufficient. Therefore, the necessary damping does not need to be high as well.

Damping

To achieve dynamic system stability in inherently unstable magnetic bearing systems, many control strategies have been introduced. Due to the lack of direct position information in tuned LC-circuits, such control strategies are of not only high complexity, but also of insufficient [2] or unsatisfactory efficiency with respect to achievable damping [1]. This is obviously a result of the low signal to noise ratio in position signals derived from current measurement which will consequently impose restrictions on the achievable velocity signal quality. Since this is a systematic problem of having to differentiate a ‘noisy’ signal twice to obtain the required feedback, a totally different approach was chosen in this paper.

During the design of a magnetic bearing system, control layout must take into account system inherent damping which is basically part of the plant, but can be utilised as favorable system property to be seen as variable for a comprehensive control design. By centering design around these inherent damping qualities of many bearing applications, such as pumps or lubricant submerged (machine) actuators, a wide range of applications can benefit from the favorable properties of magnetic suspension, especially lack of ‘roll-contact’, of slip-stick effects and the possibility to chose coating materials independent of mechanical properties (of course with restrictions concerning magnetic and electric properties).

In the presented system damping is only rested upon geometric design criteria which results from the theory of squeeze film dampers [3]. The damping parameter depending on the gap width l can be calculated from

$$c(l) = H\,\eta\,\pi^2\,\frac{d^4}{l^3} \tag{11}$$

with η being the dynamic viscosity, d the diameter of the damper, and H a scaling factor, respectively. This scaling factor describes the squeeze film geometry for a circular gap as

$$H = \frac{3\,\pi}{32}. \tag{12}$$

Limitations

The system poles are determined by the mass m of the system to be suspended, the stiffness k_l and the damping factor c. In order to achieve the relative damping ζ, the damping constant $c(l)$ must fulfill for every gap width l

$$c(l) > 2\,\zeta\,\sqrt{k_l\,m}. \tag{13}$$

The higher the stiffness of a magnetic bearing is, the higher the damping has to be in order to stabilise the entire system at a given relative damping. The primary limiting criteria of fluid-film dampers is the reachable relative damping with respect to the damper-area involved. This is of course not too drastic with magnetic bearings, because the achievable relative loads are also low compared to classic bearings and therefore, in general, it is not difficult to provide sufficient damper-area.

A further design limit for the damper is the gap width. Nevertheless, this is not a limit in actual applications, because large gaps are rarely a requirement. Limitations due to media-viscosity, however, might be critical in applications involving low-viscous fluids. On the whole, the squeeze film damping mechanism is sufficient for many applications in fluids at a reasonable stiffness.

EXPERIMENTAL SETUP

By experiment it should be shown that the combination of tuned LC-circuits and squeeze film damping is appropriate for many active magnetic bearing applications. Additionally, only little electronic effort is necessary to solve this bearing problem. The setup is depicted in Fig. 2.

Magnetic suspension system

The presented experimental setup was mainly built in order to demonstrate the suspension of an object in magnetic bearings in a fluid environment using tuned LC-circuits with no respect to any application. Therefore, neither attention has been paid to the quantities of the setup, nor to the mechanical and electrical parameters, because the setup can be scaled up and down for the appropriate size of an application.

In addition it should be shown that five degrees of freedom of the object can be controlled with the proposed mechanical design of the pole faces. Replacing the gravity force by an opposite magnet which could either be active or passive, the experimental setup results in a simple full suspension system.

TABLE 1: Parameters of electromechanic system

damper diameter	d	$57 \cdot 10^{-3}$	m
fluid viscosity	η	10^{-3}	$\mathrm{Ns/m^2}$
fluid density	ρ	10^3	$\mathrm{kg/m^3}$
object mass	m	0.114	kg
object volume	V	$456 \cdot 10^{-3}$	$\mathrm{m^3}$

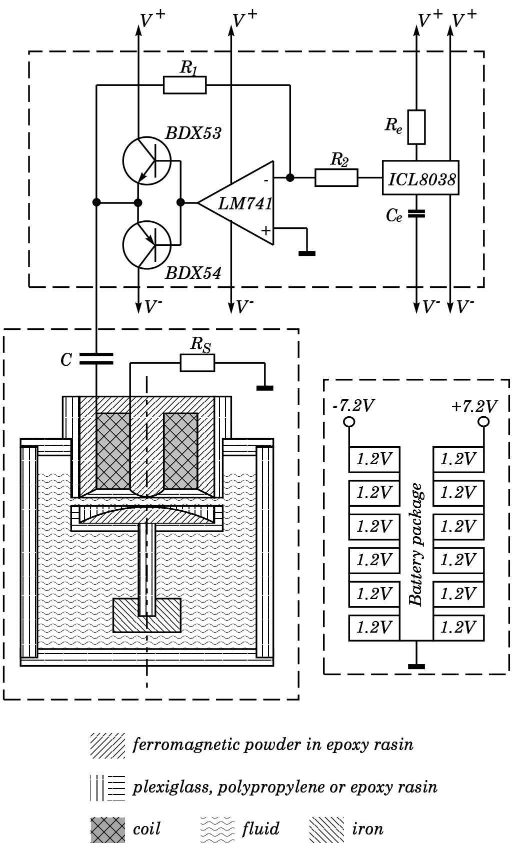

FIGURE 2: *Experimental setup with signal generator, magnetic bearing and power supply.*

Inductance

As mentioned above, the mechanical design of the pole faces have to be designed specifically. Since the special shape with spherical heads of the C-core for the electromagnet was commercially not available, the core material was made from ferromagnetic powder in epoxy rasin at a filling rate of approximately 75%. Depending on the properties of this powder and the mechanical design, the following parameters result.

TABLE 2: Parameters of inductance

resistance at 1kHz	R_L	0.536	Ω
inductance at zero gap	L_0	$3.462 \cdot 10^{-3}$	H

Capacitor

For this application a metal foil capacitor has been used for the sake of a low serial resistance.

TABLE 3: Parameters of capacitor

capacity	C	$31.16 \cdot 10^{-6}$	F
serial resistance	R_C	0.06	Ω

Resistance

An external shunt resistor R_S has been used for the measurement of the coil current. The total resistance of the LC-circuit then yields $R = R_S + R_L + R_C$.

TABLE 4: Parameters of resistors

shunt resistance	R_S	0.100	Ω
total resistance	R	0.695	Ω

Signal generator

A simple signal generator was built for this purpose, consisting of a ICL8038 as sinus generator with the resistor R_e and the capacitor C_e determining the excitation frequency ω_e. To provide the necessary power, the complementary transistors BDX53 and BDX54 have been applied with an operational amplifier LM741 as voltage controller. The resistors R_1 and R_2 determine the amplitude of the sinusoidal voltage signal. In general, this basic electronics should visualise the simplicity of the proposed solution.

TABLE 5: Parameters of signal generator

excitation frequency	ω_e	$3.17 \cdot 10^3$	$\mathrm{s^{-1}}$
excitation amplitude	E	3.0	V

Power supply

In order to demonstrate the low power consumption of the suspension of the object, a battery package was used.

TABLE 6: Parameters of power supply

positive voltage	$V+$	+7.2	V
negative voltage	$V-$	-7.2	V

EXPERIMENTAL RESULTS

Using the system with its parameters as prescribed both static and dynamic investigations of the magnetic bearing system with a tuned LC-circuit have been carried out.

Since the means for this experiments were rather simple, certain tricks have been applied to achieve measurements. Concerning the static investigations, thin paper wafers of appropriate thickness have been utilised for the measurement of the gap width, because no position sensor was available. Furthermore, it was not possible to directly measure a transient gap width with the test setup as described above. Therefore, the gap width was computed from measured current signals with the static relation between the gap width and the current amplitude. The latter was retrieved from the rectified and low passed current signal.

For all measurements the same is true as for the entire setup: not the numbers count, but the principle which could be demonstrated is essential.

Static behaviour

First of all the dependence of the inductance from the gap width was measured using a LCR-meter. As it can be seen in Fig. 3, the inductance decreases about 9% from the beginning with a rate of approximately $-0.3\,\mathrm{H/m}$.

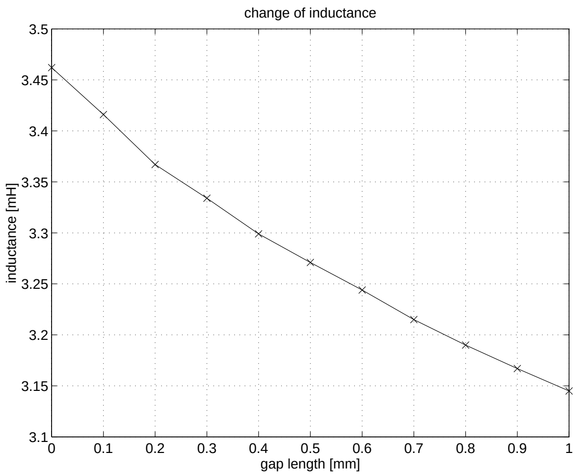

FIGURE 3: *The change of inductance depending on the gap width.*

At the voltage amplitude E and the excitation frequency ω_e the transfer behaviour of the circuit with respect to the coil current was measured in dependence of the air gap width. Figure 4 shows a maximum voltage to current rate of $6800\,\mathrm{A/m}$ for the given excitation voltage and a maximum amplitude of $4\,\mathrm{A}$.

The magnet force depending on the gap width can be seen in Fig. 5. As it is obvious, the maximum is reached, if the tuned LC circuit becomes resonant, which is the case for a gap width of about $l = 0.6 \cdot 10^{-3}\,\mathrm{m}$. For a given magnet this point is determined by the capacitor and the excitation frequency as shown in Eqn. (10). With these parameters any slope and any maximum can be realised.

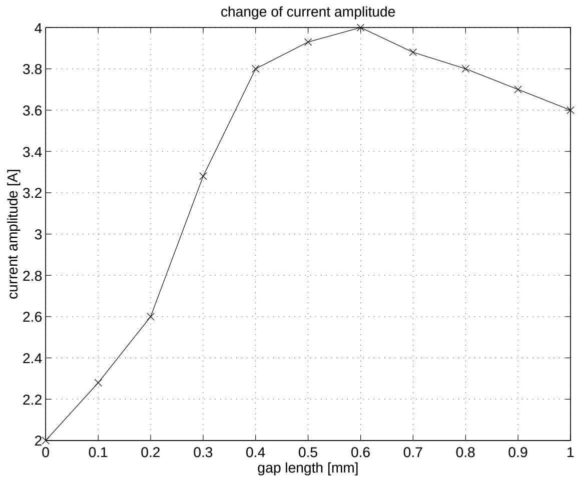

FIGURE 4: *The change of current amplitude depending on the gap width.*

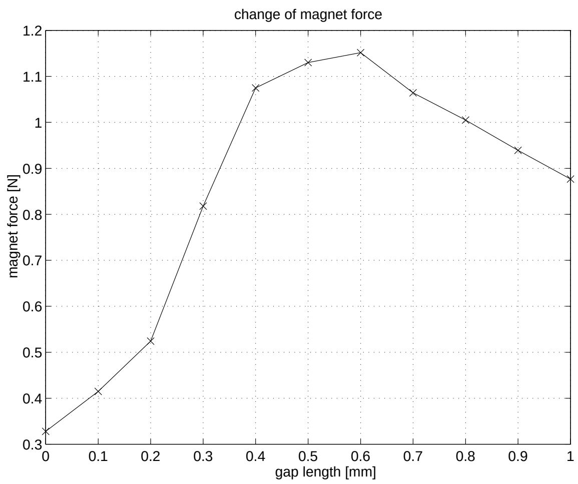

FIGURE 5: *The change of magnet force depending on the gap width.*

Since the amplitude E of the sinus generator was chosen to be $3\,\mathrm{V}$ for the magnetic suspension of the object in Fig. 2, the point of operation l_0 depends on the object mass and the fluid density. The external force on the object $F_0 = 0.671\,\mathrm{N}$ results in a static equilibrium at the gap width of $l = 0.250 \cdot 10^{-3}\,\mathrm{m}$.

With all static measurements the coefficients for the point of operation could be calculated as can be seen in the following table.

TABLE 7: Static parameters

static gap width	l_0	$0.25 \cdot 10^{-3}$	m
static force	F_0	0.6708	N
inductive parameter	k_L	-0.33	H/m
amplitude parameter	k_A	2267	A/Vm
stiffness parameter	k_l	979	N/m

Dynamic behaviour
Using the stiffness parameter and the object mass, the natural frequency of the magnetic suspension system is $92.62\,\mathrm{s}^{-1}$ which corresponds to 14.74 Hz. With a relative damping of $\zeta = 0.7$ the necessary damping c has to be more than 14.78 Ns/m according to Eqn. (13). Using Eqn. (12) with the scaling factor for circular gaps the fluid damping factor can be computed for the point of operation according to Eqn. (11) as $c(l_0) = 251.88\,\mathrm{Ns/m}$.
As it is obvious the damping factor is more than sufficient for the point of operation. The largest gap possible for the desired relative damping of $\zeta = 0.7$ is according to Eqn. (13) at a gap width of $l = 0.6 \cdot 10^{-3}$ m.

TABLE 8: Dynamic parameters

natural frequency	$\omega_n(l_0)$	92.62	s^{-1}
damping factor	$c(l_0)$	251.88	Ns/m
relative damping	$\zeta(l_0)$	11.92	-

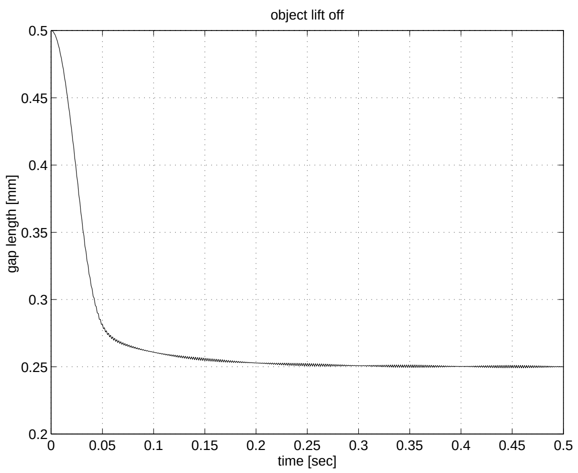

FIGURE 6: *Lift off of the object to its nominal gap width.*

Starting at a gap width of $0.5 \cdot 10^{-3}$ m a lift off could be 'measured' with the time plot in Fig. 6, which is of course a derived signal from the measured current amplitude.

In the first step of the lift off, damping is low which causes a fast response. Due to the increasing damping with the decreasing gap width, the lift off is well damped at the end. Additionally, a small ripple corrupts the gap width signal which is of course generated by the tuned LC-circuit, because the magnet force is harmonic. Since the magnet force is the square of the sinusoidal current signal, the frequency is twice as high, i.e. approximately 1kHz.

CONCLUSION

In the presented paper it could be shown by experiment that a simple and cheap concept is capable of the magnetic levitation of an object in a fluidal environment. This concept relies on the combination of an active magnetic bearing using tuned LC-circuits and system inherent squeeze film damping. Of course, this concept can only be employed in the presence of any fluid and is restricted to pumps and similar applications. However, it is obvious that for these types of plants this simple approach is sufficient and in particular, straight forward, cheap and still reliable.

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