FROM OIL TO MAGNETIC FIELDS: ACTIVE AND PASSIVE VIBRATION CONTROL

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Abstract: When a reduction of vibration amplitude usually designers resort to dampers based either on dry friction, internal material damping or fluid viscosity, each one of these mechanisms having its peculiar field of application. To improve performance while at the same time reducing costs and environmental load, electromagnetic damping devices are finding new applications, beyond the fields where they have a long history. Moreover, electromagnetic dampers can be easily controlled, obtain active or even 'intelligent' vibration control. Three examples from the internal combustion engines, automotive and gas turbine fields are discussed in some detail to show the potential advantages of this technology.
Keywords: Vibration control, eddy current dampers, active damping.

1. INTRODUCTION

Mechanical systems (actually not only mechanical) are prone to vibrate if they can store energy in two different forms, usually potential and kinetic, in a way that energy can flow from one form to the other. Vibration is the process in which this energy exchange takes place.

Since every time energy is transformed from one form to another some of it is dissipated (usually converted into low temperature thermal energy, from which it cannot be transformed back), any vibrating system is bound to come to rest eventually, unless it is connected to some energy source, providing to refurbish its energy level. This process of energy dissipation is usually referred to as damping.

The designer of any machine has usually to deal with vibration. Although there are cases where vibration is a desired effect (vibrating sieves, vibration welding machines, etc.), usually the task of the designer is to minimize, or at least to control, it.

When it is not possible to act on what excites vibration or to insulate the relevant element from it, the traditional approach to keep vibration under control is to act on the elastic and inertial characteristics of the system to modify the frequencies at which free vibration takes place (these modifications may include the addition of a further mechanical system operating as a vibration absorber) or to increase the damping properties of the system. Often both actions are required, like when using damped vibration absorbers.

Several mechanisms can be used to dissipate energy during vibration. Those traditionally employed are:

– Dry friction between two surfaces moving in contact with each other;
– Internal damping of some materials;
– Viscous forces in a fluid.

Dry friction is today relied upon only for small quantities of energy, and in very simple machines. Its major disadvantage, introducing nonlinearities into the system, is considered a serious drawback, both for its performance and owing to difficulties in modeling its behavior.

Internal damping of most engineering materials is too low to be used to dissipate large quantities of energy, so that dampers of this kind have some restricted fields of application. Elastomeric materials may be tailored to have the required damping characteristics, but their low thermal conductivity and poor high temperature characteristics limit their applications. Nevertheless many small automotive diesel engines use torsional vibration dampers based on elastomeric elements.

Viscous dampers are widely used in many applications, from torsional vibration dampers in reciprocating engines to squeeze film dampers in turbines, from automotive shock absorbers to large dampers used in large buildings, just to name a few examples. In their basic form they are linear devises, since the force they supply is proportional to the velocity

$$F = c\dot{x}\,, \qquad (1)$$

to the point that viscous damper has become a synonymous of linear damper. It is however possible to obtain different law $F(\dot{x})$ by adequately designing the system.

Another way for dissipating energy in a vibrating system is transforming the (mechanical) vibration energy into electric energy and then dissipating it through Joule effect. This can be done both in a solid conductor, by generating eddy currents, or through purposely built electric circuits in which a current can flow. All these electromagnetic devices need a magnetic field to operate: if this is produced by permanent magnets no external power is required, and a passive system is obtained. If, on the contrary, an electromagnet is used, some external power is required and then the system must be considered as active (Some confusion about this terminology is sometimes found. Here a passive system is intended as a system requiring no external energy to work, while an active system is a system that needs to receive power. This has nothing to do with whether the system is controlled or not: an eddy current damper in which the magnetic field is produced by an electromagnet fed by a constant current is active but not controlled. The term semiactive is also used, often with the meaning of a controlled system needing a limited amount of energy. What limited means is clearly arbitrary). In principle, the vibration energy extracted from the system may be converted into some useful form of energy to be used somewhere else: this require a transducer working on four quadrants instead of a simple energy dissipator (usually a resistor).

Electric and magnetic circuits may be coupled to the mechanical system through transducers of various kind, so that to modify the dynamics of the system. Electromagnetic damped vibration absorbers may use electric damped oscillators instead of springs, masses and dampers.

Active devices, based on actuators exerting forces on the system with the aim of reducing vibration, can nowadays be added to these more or less classical ways of controlling vibration. Since they supply energy to the system, the actuators must be carefully controlled to avoid instabilities. While passive systems based on energy dissipators are intrinsically stable, the stability of active systems must be checked carefully in each case.

2. OIL AND RUBBER VS. MAGNETIC FIELDS

The trend towards electromagnetic dampers, be they active or passive, controlled or not, is clear. It is due to both new needs and new opportunities.

Viscous and elastomeric dampers have basically 3 problems.

The first is thermal stability. The viscosity or oil and the internal damping of rubber are strongly dependent of temperature. Specific problems are then cavitation in oleodynamic dampers (cavitation is an important problem in automotive shock absorbers) and overeating with subsequent failure in elastomeric dampers. Elastomeric materials show also a large dependence of their damping characteristics on frequency.

The second problem is linked with manufacturing and disposal. Environmental laws cause the cost of dealing with oils and above all to dispose of used oil to increase steadily. Strict rules are also imposed on the manufacture and disposal of rubber machine components. These rules are bound to become more strict in the future with further increasing costs.

A third point, linked with viscous damping devices, is the difficulty of controlling the characteristics of the fluid. Electrorheological and magnetorheological fluids have been investigated showing attractive potentialities for tuning damping forces according to the operating conditions. However they too have problems related to the ageing of the fluid and to the tuning required for the compensation of the temperature and frequency effects.

Electromagnetic dampers improve much this situation. Although the resistivity of conductors (a property that determines the characteristics of electromagnetic dampers) is a function of the temperature, this dependence is much weaker that that of the viscosity of oils or the internal damping of rubbers. Their manufacture and disposal is much less subject to restrictions and they are easily controlled by electronic, possibly digital, microprocessor-based, devices.

Electromagnetic dampers were often discarded in the past for many applications because they were considered too heavy, bulky and costly. Recently the situation has much improved.

New opportunities have opened with the introduction of high-performance rare-earths permanent magnets. They allow generating magnetic fields much more intense than those due to traditional magnets for a given quantity of magnetic material. The mass and bulk of electromagnetic dampers is thus reduced a great deal. After the expiry of the original patents, their cost started decreasing and today it is possible to build electromagnetic passive dampers that are competitive with traditional dampers.

Even if slower, there has been also a progress in the field of soft magnetic materials, which helps in containing the size and weight of electromagnetic dampers.

To describe the potentialities of electromagnetic dampers, three applications will be described in detail. They deal with vibration in widely different frequency range, and are based on different layouts.

Electromagnetic dampers can be built following two different schemes: vibrational motion can cause electric currents to be generated, which flowing either n a solid conductor or a resistor of various type produces the required energy dissipation. Alternatively, electromagnetic forces can be produced by controlled actuators, either of the Lorentz or the Maxwell type. While in the first case damping is automatically produced, in the second case the controller must ensure that the forces oppose vibrational motion, so that energy is dissipated and vibration is quenched.

3. MOTIONAL EDDY CURRENT DAMPERS FOR TORSIONAL VIBRATION CONTROL

The electromechanical dynamics of a torsional eddy current damper of motional type can be studied using the model shown in Fig. 1. Such a configuration is characterized by a single magnetic pole pair (Tonoli, 2007; Graves et al., 2000). The rotor is made by two windings 1, 1’ and 2, 2’ installed on orthogonal planes. It is crossed by the constant magnetic field (flux density Bs) generated by the stator. The analysis is performed under the following assumptions:

– The two rotor coils have the same electric parameters and are shunted,
– The reluctance of the magnetic circuit is constant. The analysis is therefore only applicable to motional eddy current devices and not to transformer ones (Graves et al., 2000; Kamerbeek, 1973),

– The magnetic flux generated by the stator is constant as if it were produced by permanent magnets or by current driven electromagnets,
– All quantities are assumed to be independent from the axial coordinate,
– Every electric parameter is assumed to be lumped.

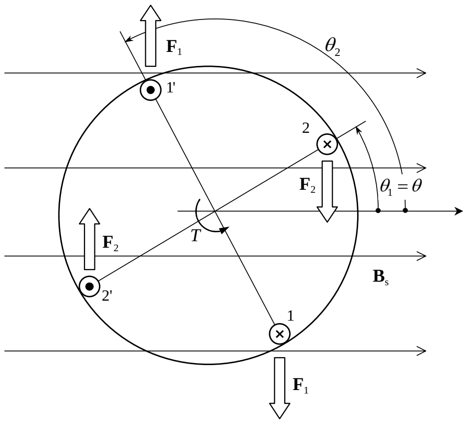


Fig. 1. Sketch of the eddy current damper

Angle $\theta(t)$ between the plane of winding 2 and the direction of the magnetic field indicates the angular position of the rotor relative to the stator. When currents $i_{\gamma 1}$ and $i_{\gamma 2}$ flow in the windings, they interact with the magnetic field of the stator and generate a pair of Lorentz forces $F_{1,2}$ in Fig. 1. Each force is perpendicular to both the magnetic field and the axis of the conductors. Their magnitudes are:

$$F_1 = Nl_r i_{r1} B_s, \quad F_2 = Nl_r i_{r2} B_s, \tag{2}$$

where N and l_γ indicate the number of winding in each coil and their axial length, respectively.

The total torque T acting on the rotor is:

$$T = \phi_{rs0}\left[\sin(\theta) i_{r1} + \cos(\theta) i_{r2}\right]. \tag{3}$$

For constant rotation speeds ω, the torque to speed characteristics is found:

$$T = \frac{c_0}{1+(p\omega)^2/\omega_p^2}, \text{ with } c_0 = \frac{\phi_{rs0}^2}{R_r}, \ \omega_p = \frac{R_r}{L_r}, \tag{4}$$

where, c_0, p and ω_p are respectively the damping at low frequencies, the number of magnet poles and the frequency of the electric pole of the system, and $\phi_{\gamma s0}$ is the equivalent crank radius. R_γ and L_γ are the resistance and inductance of the rotating coils. The mechanical impedance can also be obtained using the Laplace transform as:

$$Z_m(s) = \frac{T(s)}{\dot{\theta}(s)} = \frac{c_{em}}{1+s/\omega_p} = \frac{c_{em}}{1+s\left(k_{em}/c_{em}\right)}. \tag{5}$$

This impedance corresponds to a viscous torsional damper and a torsional spring connected in series, whose parameters are:

$$c_{em} = \frac{p\phi_{rs0}^2}{R_r} \text{ and } k_{em} = \frac{p\phi_{rs0}^2}{L_r}. \tag{6}$$

The model has been validated experimentally (Tonoli and Amati, 2008) using a test rig based on a four pole pairs axial flux induction machine (Fig. 2, steady state tests).

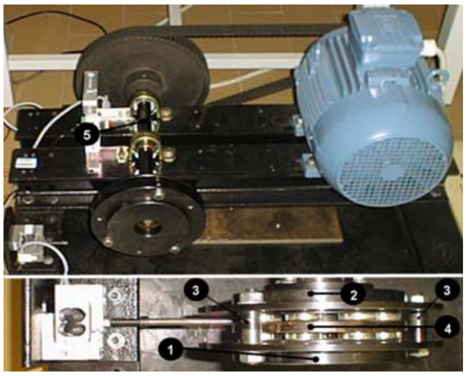


Fig. 2. Test used for the identification of the induction machine at steady state

In Fig. 2, the magnetic flux is generated by permanent magnets while energy is dissipated in a solid conductive disk. The first array of 8 circular permanent magnets is bond on the iron disk (1) with alternate axial magnetization. The second array is bond on the disk (2) with the same criterion. Three calibrated pins (3) are used to face the two iron disk - permanent magnet assemblies ensuring a 1 mm airgap between the conductor and the magnet arrays. The latter are circumferentially oriented so that the magnets with opposite magnetization are faced to each other. In the following such an assembly is named "stator". The conductor disk (4) is placed in between the two arrays of magnets and is fixed to the shaft (5). It can rotate relative to the stator by means of two ball bearings installed in the hub. The magnetic circuit has been designed using a simplified one-dimensional analysis with the aim of avoiding saturation in the iron parts. The main features of the induction machine are summarized in Tab. 1.

Tab. 1. Main features of the induction machine used for the tests

Number of pole pairs	-	4
Diameter of the magnets	mm	30
Thickness of the magnets	mm	6
Magnets' geometry	-	Circular
Magnets' material	-	Nd-Fe-B(N45)
Residual magnetization of the magnets	T	1.22
Thickness of conductor disk	mm	7
Resistivity of conductor (Cu)	Ohm m	$57\ 10^{-6}$
Airgap	mm	1

Experimental tests at constant speed have been carried out to identify the slope c_0 of the torque to speed characteristic at zero or low speed and the pole frequency ω_p. While the former has been identified in quasi-static tests, the latter has been identified as best fit of the experimental points reported in Fig. 3 (Tonoli and Amati, 2008). The identified parameters are c_0 = 1.24 Nms/rad and ω_p = 51.1 Hz, from which the induction machine characteristics can be obtained.

An interesting field of application of this technology is the vibration damping of the crankshafts in internal combustion engines. In fact, technological issues make active solutions difficult to be implemented, which naturally lead to passive solutions. Nowadays, the common solutions involve either elastomers or viscous fluids. However, they undergo a large number of stress-strain cycles, and consequently their working life is reduced to approximately half the working life of a motor. Instead, eddy current damping does not imply the deformation of the material of the actuator, Therefore, the working life is increased, and the efficiency of the apparatus is improved (Fig. 4).

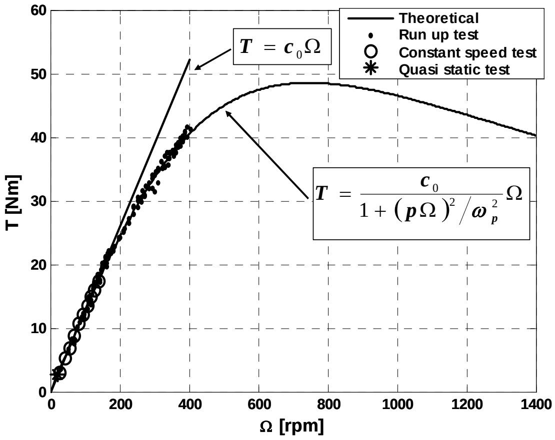


Fig. 3. Experimental results of the induction machine characterization at steady state

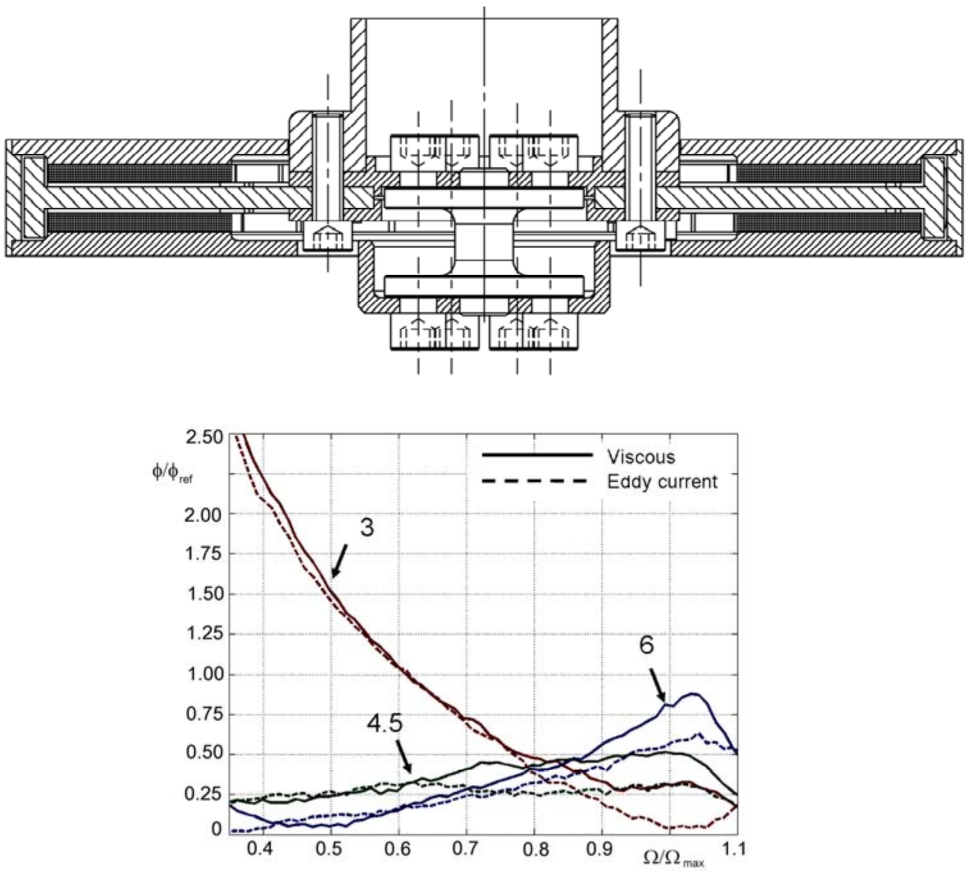


Fig. 4. Torsional damper: schematic cross section and comparison of the amplitude of three harmonics in the nondimensional torsional response of the engine using a viscous and an eddy current damper

4. ELECTROMAGNETIC AUTOMOTIVE SHOCK ABSORBERS

Working principle. Main tasks of automotive shock absorbers are the capability of reducing transmission of vibration and controlling the applied load. Active solution can be used to achieve these goals, but very good results may be obtained also from semi-active configurations, allowing to change the value of damping, depending on the driving conditions. Conventional solutions are nowadays based on hydraulic dampers, whose characteristics may be variable as a result of modifications in the hydraulic circuit or in the oil physical properties (magnetorheologic fluids). Solution based on linear electric motors were proposed to solve the typical problems related to the use of fluids, but they lead to an increase of size and mass (Karnopp, 1989). This is due to the fact that, while in rotary electric motors all the magnetic and conductor material is always active, in linear motors only a relatively small part of it is working at each time. Moreover, rotating electric motors are based on a more consolidated design practice and on a higher technology background.

The electromagnetic technology based on brushless motors allows obtaining a tunable suspension damper with many advantages and limited disadvantages. The capability of being tunable, and regenerating energy (four-quadrant operation) with limited increase of mass and size using an actuator that can become fully active are considered important features.

The tunable electromechanical damper described in the following section is based on a DC electric motor whose electric terminals are shunted on a resistive load instead of being connected to a converter: the torque needed to rotate the shunted motor can be computed from the characteristic equations of the electric machine. They link the back electromotive force V_{emf} and the electromechanical torque T_{em} to the rotating speed ω and the electric current i

$$V_{emf} = K_e \cdot \omega \; ; \; T_{em} = K_t \cdot i . \tag{7}$$

K_e and K_t indicate the back electromotive force constant and the torque constant, respectively. If the electric terminals of the motor are shunted by a resistance R, the current i (eddy current) is induced by the back electromotive force V_{emf}, so that (in case of a constant speed ω)

$$V_{emf} = R \cdot i . \tag{8}$$

Substituting the Ohm equation (8), into the motor characteristic equations (7) allows to get rid of the back electromotive force and of the eddy current. The back electromotive force and the torque constants are the same, both are indicated as K_m. The electromechanical torque T_{em} is then related to the angular speed ω as shown in Fig. 3a:

$$T_{em} = c_\omega \cdot \omega \text{ , where } c_\omega = \frac{K_m^{\;2}}{R} . \tag{9}$$

From the mechanical point of view the resistively shunted electric motor behaves as a torsional viscous damper (Graves et al., 2000; Karnoppm 1987, Amati et al., 2006). The torsional damping coefficient c_ω can be tuned by acting on the shunt resistance. The lower is the resistance, the higher is the damping. The solution in

Fig. 5 shows a linear electromechanical damper including a rotary motor and a ball screw transmission.

From the electrical point of view, the motor of

Fig. 5 is represented as a universal motor with two electric terminals (such as a brush motor). R includes the resistance of the motor windings (R_m) and the external one (R_{ext}) that allows to tune the damping coefficient ($R = R_m + R_{ext}$). As the effect of the inductance of the motor coil L is unwanted, no external contribution is usually added to it. From the mechanical point of view, the contribution of the masses and inertias on the dynamic performances of the damper cannot be neglected.

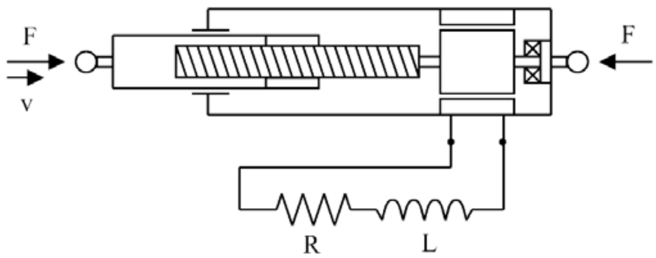


Fig. 5. Sketch of the electromechanical shock absorber

The equivalent mechanical model of the device shown in

Fig. **5** is reported in Fig. 6. It takes into account the contribution of the electric and mechanical part of the motor written at the level of the damper. A detailed description of the model, that allows to take into account its frequency dependency, is reported in Amati et al (2006). To this end, the effect of the rotor inertia J, the motor inductance (L) and resistance (R) can be written as:

$$m_{eq} = \tau^2 \cdot J \;\; , \;\; c_{eq} = c_s + \tau^2 \cdot c_\omega \;\; , \;\; k_{em} = \tau^2 \cdot k_\omega \; , \tag{10}$$

where $c_\omega = K_m^2/R$ and $k_\omega = K_m^2/L$, are the torsional damping and stiffness produced by the shunted motor.

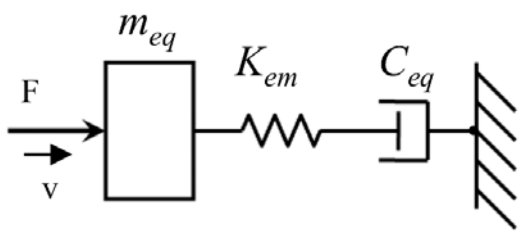


Fig. 6. Mechanical analogue of the electric motor

The dynamic behavior of the damper can therefore be characterized by the mechanical impedance given by the following equation:

$$\frac{F}{v} = m_{eq} \frac{s^2 + 2\zeta_0 \omega_0 s + \omega_0^{\;2}}{s + \omega_p} , \tag{11}$$

where s is the Laplace variable. The pole frequency ω_p, the zero frequency ω_0, and damping factor ζ_0 are

$$\omega_p = \frac{k_{em}}{c_{eq}} \;,\; \omega_0^2 = \frac{k_{em}}{m_{eq}} \;,\; \zeta_0 = \frac{\sqrt{m_{eq} k_{em}}}{2c_{eq}} \tag{12}$$

Fig. 7 holds in the case where ω_p is lower than ω_0 (0 ζ_0 1 2). Between the pole and the zeros the system behaves as a spring of stiffness k_{em} as shown by the -20 dB/dec slope of the mechanical impedance. A detailed study of the frequency behavior is reported in Amati et al (2006).

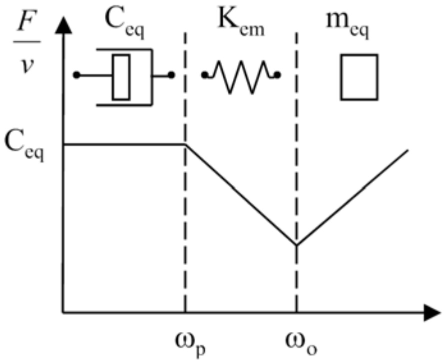


Fig. 7. Mechanical impedance of the damper

4.1. Electromechanical design

The rather typical design situation is strive to obtain the specified damping coefficient while keeping the equivalent mass (Equation 10) as small as possible. Since the equivalent mass is related to the moment of inertia of the rotor, the aim of this section is to find a relationship between the electromechanical damping coefficient c_ω in equation 9 and the moment of inertia J of the rotor. The analysis is performed assuming a conventional brushless motor with permanent magnets on the rotor surface running in a the toothed and slotted stator.

The analysis performed in Amati et al (2006) leads to the following expression of the damping coefficient

$$c_\omega = \Gamma_c r^4 l \,, \tag{13}$$

where parameter Γ_c is a function of the motor shape and technology.

From the mechanical point of view it is worth to notice that the torsional damping coefficient is proportional to the moment of inertia of the rotor J_{em}. Under the assumption that the rotor is an homogeneous cylinder made of an material with density ρ this is

$$J_{em} = \frac{\pi}{2} \rho \cdot r^4 \cdot l \,. \tag{14}$$

By introducing Eq. (13) into Eq. (14), it follows that

$$J_{em} = \frac{\pi}{2} \rho \cdot \frac{c_\omega}{\Gamma_c} \,. \tag{15}$$

Coming back to the layout of

Fig. **5**, the equivalent mechanical parameters of equation (10) are linked together similarly to torsional damping and rotor inertia. The ratio between the equivalent mass and damping due to the electromechanical effects is constant, its value being a function of the motor shape and technology

$$\frac{m_{eq}}{c_{eq} - c_s} = \frac{J_{em}}{c_\omega} = \frac{\pi}{2} \frac{\rho}{\Gamma_c} \,. \tag{16}$$

This is a very important result from the design point of view. Once the equivalent damping is specified, the equivalent mass to be minimized is just a function of the motor shape and technology (ρ Γ_c), and not of the transmission ratio τ. The only way to reduce it is to improve constant Γ_c.

In addition to the rotor inertia, the overall mass of the electric motor m_{em} is the other important mechanical feature of the damper.

For what is concerned to the total mass, in Amati et al (2006) with reference to the shock absorber of

Fig. **5**, the mass of the electrical motor is expressed as function of the equivalent damping of Eq. (10) as

$$m_{em} = \frac{\Gamma_m}{\Gamma_c} \frac{(c_{eq} - c_s)}{(\tau r)^2} r^4 l \,. \tag{17}$$

Equation (17) shows it is related to technology and $(\tau\, r)^2$ once the desired damping coefficient is chosen. For a given equivalent damping (c_{eq}), motor shape and technology (this gives Γ_m and Γ_c) the larger the rotor radius r and the transmission ratio τ, the smaller the mass of the electric motor. The apparently unreasonable lower mass corresponding to a larger radius is due to the smaller amount of conductor necessary to obtain the desired damping c_{eq}.

To conclude the present section, equations (10), (13), (16), and (17) outline a design procedure of the electromechanical damper.

4.2. Application example

To analyze the potentialities of electromechanical dampers in automotive applications, the previously outlined procedure has been applied to design the damper of a C-segment vehicle front suspension. The application example was done on a Mc Pherson suspension using the same mechanical interfaces as the original system, and using the inner space of the coil spring with an appropriate allowance (device data are reported in Tab. 2). With reference to the layout of Fig. 8 the electromagnetic damper is represented in shade.

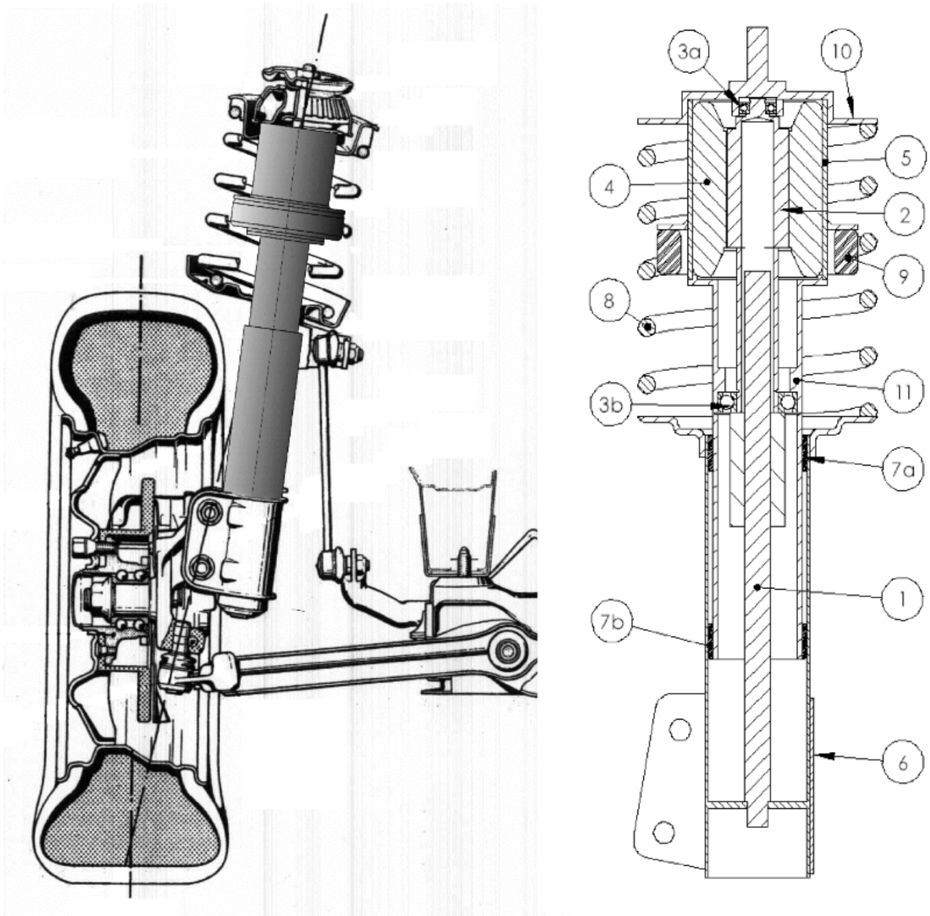


Fig. 8. Electromechanical damper integrated in a Mc Pherson suspension and cross section of the damper

The electric motor (2) (4) (5) is housed in upper part of the cylinder close to the upper strut mount (10). This choice allows to exploit the relatively large diameter inside the spring (8) while keeping a small diameter close to the wheel. The screw (1) is rigidly connected to the moving piston (6), bolted to the hub. The motor, the nut of the ball screw, and the guiding tubes (11) are connected to the sprung mass. The rotor of the electric motor and the nut of the ball screw are rotating on ball bearings (3a) (3b) and connected together, this reduces considerably the axial length of the device as it exploits the room inside the rotor of the motor.

The total mass of the electric damper is shown in Fig. 8 is 5 kg. This mass has to be compared to the 4.1 kg of a hydraulic solution with continuously variable damping. The proposed damper is still heavier than the hydraulic one but is much lighter than the configurations found in the literature with comparable performances. A passive electromechanical damper based on a linear electric motor was designed using the same specifications (Karnopp, 1989) and lead to a device of more than 15 kg of mass.

The effects of the damper in the car suspension have been investigated by integrating the model in a simple quarter car model. Fig. 9 shows the transfer function between the displacement of the contact point of the tire to the ground and the acceleration of the sprung mass.

Tab. 2. Damper specifications and parameters of the single corner suspension

Max damp. Coeff. (rebound)		>10	kNs/m
Max speed		1	m/s
Max stroke (peak to peak)		150	Mm
Spring stiffness	K_s	18	kN/m
Tire radial stiffness	K_p	150	kN/m
Tire radial damping	C_p	50	Ns/m
Sprung mass	M_s	450	kg
Unsprung mass	M_n	30	kg
Axial length at midstroke		450	Mm
Max force		1.8	kN

The undamped response curves is that of the open circuit damper ($R_{ext} = \infty$: no electro-mechanical damping); the other ones to some values the external resistance R_{ext} . The equivalent mass (m_{eq} q. 10) due to the rotor inertia introduces a sort of inertial coupling between the sprung and unsprung masses. The effect is, at any rate, not large as demonstrated by the fact that the two undamped natural frequencies of the suspension go from 0.95 Hz and 11.91 Hz (no electromechanical damper installed) to 0.94 Hz and 11.12 Hz (open circuited electromechanical damper).

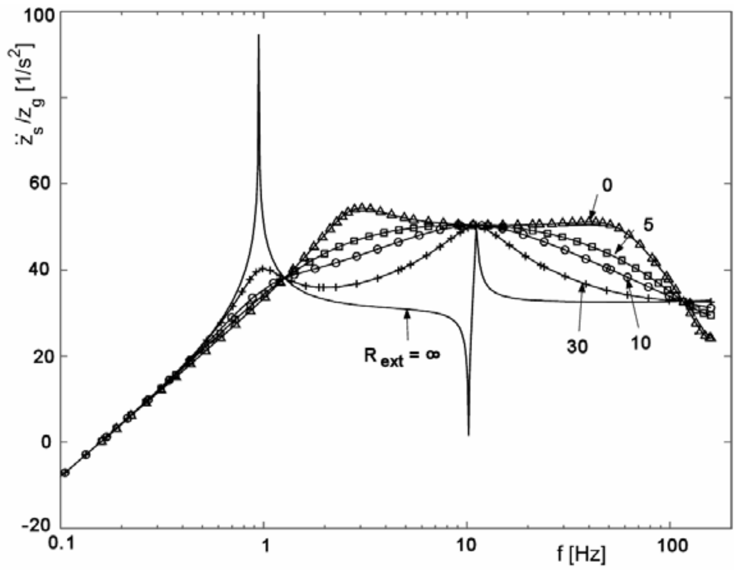


Fig. 9. Transfer function between displacement of the ground (input) and acceleration of the sprung mass (output)

5. ACTIVE AND PASSIVE DAMPERS FOR ROTATING MACHINES

5.1. Active magnetic damper (AMD)

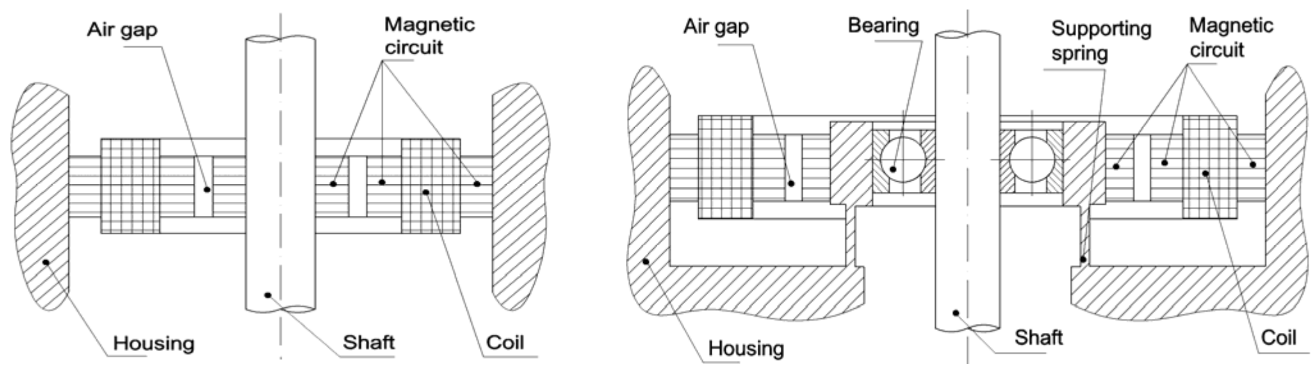
In the present section the use of active magnetic bearings (AMB, Fig. 10a) for vibration damping (active magnetic damper referred to as AMD, Fig. 10b) (Amati et al., 2006) is introduced. Those two configurations differ only by the function of the force generated by the actuator. While for the AMB, the actuator is used for both suspension and damping, the AMD is used only to introduce damping into the system and the of the suspension must be insured by mechanical means. The static stability of the suspension is thus guaranteed if the mechanical stiffness is greater than the open-loop negative stiffness of the AMD.

The AMD technology is particularly well suited in the field of rotating machines, where they can replace with advantages squeeze film dampers.

A model of the actuator is needed for the design of the control law of the AMD. To this aim, the usual expression of the force generated by one electromagnet is used:

$$F = \frac{\mu_0 N^2 A}{4}\left(\frac{i(t)}{q_0 + q(t)}\right)^2, \tag{18}$$

where N is the number of turns of each winding, A is the area of the magnetic circuit at the air gap, μ_0 is the magnetic permeability of vacuum, i is the current, q_0 is the nominal air gap and q is the displacement.



a) b)

Fig. 10. Sketch of an active magnetic bearing (a) and of an active magnetic damper (b) operating also as an elastic support

5.2. Transformer eddy current dampers

A "transformer" eddy current damper may use the same configuration as an AMD, although in this case the coils are supplied with constant voltage through which the magnetic field is generated. Damping is thus provided by eddy currents instead by control forces as in AMDs. Both AMD and transformer operation is possible in the case shown in Fig. 10b. However, in the latter case, the absence of a control law can make the implementation easier.

The basic principle of transformer eddy current dampers is the following: the displacement with speed $\dot q$ of the anchor changes the reluctance of the magnetic circuit causing the flux linkage to change in time, which generates a back electromotive force in the coils, and consequently eddy currents in the coils. The current in the coils has thus two contributions: a fixed one due to the voltage applied, and a variable one induced by the back electromotive force. The first contribution generates a force that increases when the air-gap decreases, which produces a negative stiffness. The damping force is generated by the contribution due to the anchor speed $\dot q$, that acts against the motion of the moving element. In these terms, this configuration can be called semi-active, as power is initially required to produce the initial magnetic field, but damping is introduced without a traditional feedback law (i.e. using sensors).

The transfer function between the speed $\dot q$ and the electromagnetic force F shows a first order dynamics with pole frequency ω_{RL} due to the R-L nature of each circuit

$$\frac{F}{\dot q}=\frac{1}{s}\frac{K_{em}}{\left(1+s/\omega_{RL}\right)}, \qquad (19)$$

where $K_{em}=-\frac{2V^2/R}{q_0^2\omega_{RL}}$, $\omega_{RL}=\frac{R}{L_0}$, $L_0=\frac{\mu_0 N^2 A}{2q_0}$.

The mechanical impedance has the form of a band limited negative stiffness. The value of the negative stiffness is proportional to the electrical power (V^2/R) dissipated at steady state by the electromagnet. For given values of the number of turns (N), of the air gap area (A), and nominal air-gap (q_0), the mechanical impedance and the pole frequency are functions of the voltage applied V and the resistance R.

As mentioned earlier, a mechanical spring must be located in parallel to the electromagnets to compensate their negative stiffness. For static stability, the additional stiffness K_m must be larger than the negative stiffness of the electromagnets ($K_m \geq |K_{em}|$). For the analysis, the mechanical spring in parallel to the transformer damper can be considered as a part of the damper, and from now on the whole "electromagnet + mechanical spring" will be referred to as "transformer electromechanical damper" (TEMD). The mechanical impedance of the damper in parallel with the mechanical spring, i.e. of the TEMD, is studied:

$$\frac{F}{\dot q}=\frac{1}{s}\left(\frac{K_{em}}{\left(1+s/\omega_{RL}\right)}+K_m\right)=\frac{K_{eq}}{s}\frac{1+s/\omega_z}{1+s/\omega_{RL}}, \qquad (20)$$

where $K_{eq}=K_m+K_{em}$; $\omega_z=\omega_{RL}\frac{K_{eq}}{K_m}$.

Apart from the pole at null frequency, the impedance shows a zero-pole behavior. To ensure stability, the frequency of the zero must to be smaller than the frequency ($0<\omega_z<\omega_{RL}$) of the pole. As shown in Fig. 11a, it is possible to identify three different frequency ranges.

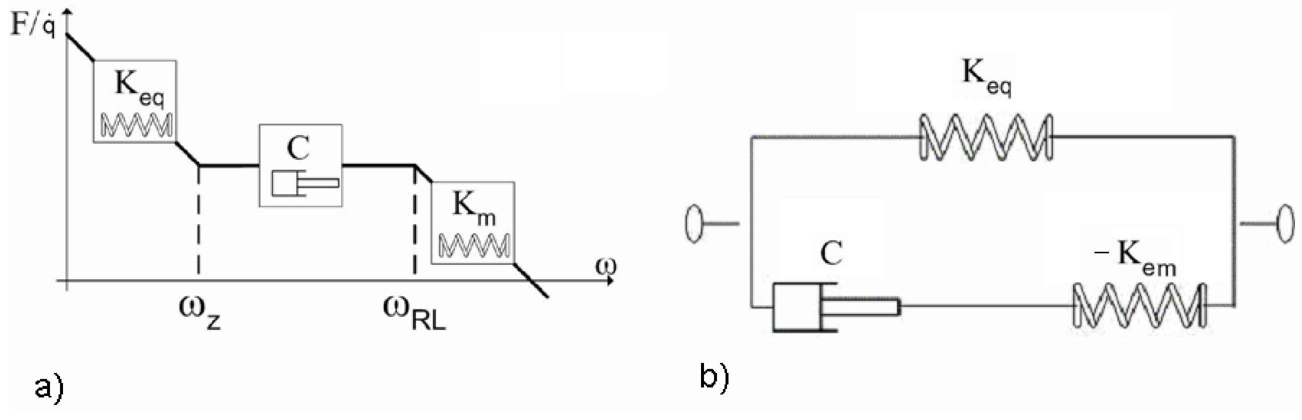


Fig. 11. "Transformer" eddy current damper. A) Mechanical impedance of a transformer damper in parallel to a spring of stiffness K_{eq} and b) mechanical equivalent

– Equivalent stiffness range $\omega << \omega_z < \omega_{RL}$: the system behaves as a spring of stiffness $K_{eq} > 0$.
– Damping range $\omega_z < \omega < \omega_{RL}$: the system behaves as a viscous damper with coefficient C defined as:
– $$C = \frac{K_m}{\omega_{RL}}. \tag{21}$$
– Mechanical stiffness range $\omega_z < \omega_{RL} << \omega$: the transformer damper contribution vanishes and the remaining stiffness contribution is provided by the mechanical spring (K_m).

It can be concluded that the association of electromagnets with a mechanical suspension opens interesting perspectives for the vibration control of rotating machines. As the AMD and TEMD configurations are identical, it is possible to switch from one strategy to the other while the machine is running.

5. CONCLUSIONS

'More electric' or even 'All electric' are more than fashionable catch phrases. They synthesize a tendency gaining momentum in many fields of technology towards a larger use of electric machines to control motion and to distribute power, while trying to do without lubricating or damping fluids. Even in cases where it is impossible to avoid hydraulic transmission, the electrohydraulic approach allows to mate the great compactness of hydraulic machinery and transmission lines with the ease of control of electric machines.

While the road to build an 'all electric' turbojet or power turbine (to quote two examples) is still long and full of obstacles, the use of electromagnetic devices to control vibration and to manage power is in many cases mature for applications. Passive, semiactive and fully active dampers find increasing applications not only in those high technology fields in which they were first used, but also in low cost consumer markets.

The automotive shock absorber and the internal combustion engine torsional dampers here shown are two examples of this trend. In eddy current dampers, usually passive or semiactive, the substitution of magnetic fields instead of oil or elastomers is straightforward and the only difficulties are costs and sometimes still mass and bulk.

Fully active devices on the contrary, apart from being more complex and usually costly, require accurate studies since their behavior is not intrinsically stable. As usual with active systems, their control laws must be designed with both cost and performance and stability in mind.

Although being still far in the future, the vision of machines with no part in contact and hence requiring no lubricant and displaying no wear, and with vibration control that does not involve energy dissipation, is a clear goal that can guide designers and scientists.

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