EXPERIMENTAL RESEARCH ON A WIDEBAND PASSIVE DYNAMIC ABSORBER USEFUL TO INCREASE THE BORING BARS STABILITY

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Abstract: The self-generated vibrations in boring process (also known as chatter) are extremely undesirable because their effect concern the quality of surface and the precision of the workpiece. Also, in some cases, these vibrations are able to destroy the cutting tool or the workpiece. Generally these vibrations occur because in cutting processes there is at least one close loop with mechanical dynamic positive feedback (e.g. a velocity-force positive feedback). Currently the boring bar is involved in chatter for the reason that it has low frequency flexural vibration modes with low natural damping (so high resonating amplification). The paper proposes a very simple technique to increase the stability of boring, based on a wideband passive dynamic absorber which is possible to be placed inside the boring bar. The absorber consists of a mass, a viscoelastic suspension and a passive damper based on the interaction between eddy currents and a permanent magnetic field. Some experimental research was performed on a simple cantilever beam setup. The paper presents several results on the efficiency of this absorber (modal attenuation up to 38 dB and 60 Hz bandwidth).

Key words: Stability, boring process, vibrations, passive damping, boring bar.

1. INTRODUCTION

Frequently in cutting processes with boring bars occur self-generated vibrations also called chatter. These self-generated vibrations produce a poor quality of surface, strong vibrations and loud noise and generally the reduction of tool lifetime. According to Merrit (Merrit, 1965) these vibrations are generated because of the dynamic instability of the close loop interaction between the tool and workpiece during the cutting process. On certain values of the speed of rotation of the boring bar (described on the stability lobe diagram) the feedback inside the close loop system becomes positive, it occurs a relative vibratory motion between tool and workpiece, on a modal frequency of the boring bar, self- excited at resonance. A positive feedback inside the close loop of the cutting tool and workpiece elastic system, signify sometime that at least in one point placed on the cutting edge of the boring bar the cutting force F is direct proportional with the absolute displacement x of that point, the force and displacement are in the same direction, $\partial F / \partial x > 0$, negative stiffness is generated inside the elastic system. This kind of feedback is rarely involved in instability with self-excited vibration.

In most any cases a positive feedback signify that on the cutting edge of the boring bar there is a point where the cutting force is direct proportional with the absolute velocity *dx/dt* and in the same direction (proportional derivative feedback). In the close loop dynamic system a negative damping is generated. This kind of positive feedback is usually involved in instability with self-excited vibration, if the negative damping is bigger than the natural damping of the system (always positive). If the damping is positive (force and velocity in opposite direction or the phase angle θ between force and velocity is: $\pi/2 < \theta < 3\pi/2$) then a negative modal power P_n is generated. This mechanical power is converted in heat and eliminated from the system (Horodinca et al., 2011). If the damping is negative (force and velocity in the same direction, or the phase angle θ between force and velocity is: $-\pi/2 < \theta < \pi/2$) then a positive modal power P_p is generated. This power is absorbed by the mechanical system and used to actuate the vibratory motion.

The system is stable only if:

$$P_n > P_p \tag{1}$$

According to Tobias and Fishwick (Tobias & Fishwick, 1958) the chatter mechanism is produced by a regenerative effect. As a consequence of an external perturbation, a damped relative vibration between tool and workpiece occurs (on a certain modal frequency). There is a local periodical variation of the cutting depth; a wavy surface is generated on the workpiece. For the next rotations of the tool or workpiece this local periodical variation of the wavy surface produces a variable cutting force acting as a excitation on the resonant frequency of the dynamic system, the amplitude of the relative vibratory motion and the amplitude of the local periodical variation of the wavy surface and cutting depth are increased (Stepan, 2001) at each future rotation. Both scenarios for the appearance of cutting instability presume a solution to reduce or to eliminate the chatter: *increasing the structural damping (the damping ratio ξ) of the mode which is excited*. According to Eq. (1) this implies the increasing of the amount of negative power P_n until the stability condition ($P_n > P_p$) is assured.

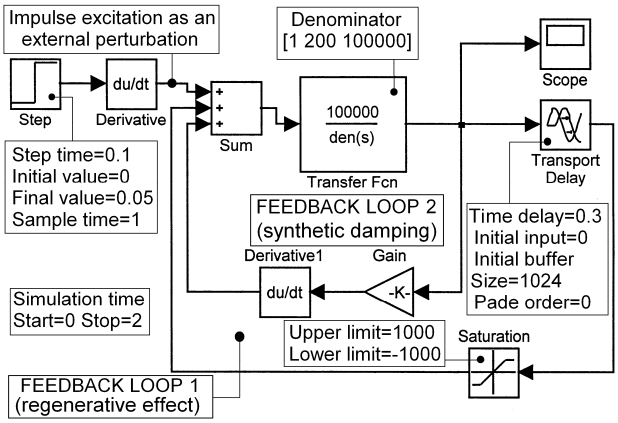

Fig.1. A Matlab/Simulink model of chatter based on regenerative effect. Chatter correction using negative feedback in feedback loop 2

In certain conditions, for the chatter scenario based on feedback, a time delay (or a phase shift as well) introduced in the feedback loop (Ganguli, 2005) increases the stability of the cutting process.

The increasing of the structural damping of a boring bar (for flexural vibration modes) can be done in active way (Sims et al., 2005), using a collocated piezoelectric sensor and actuator (Håkansson & Andrén, 2004) and a negative feedback loop between. The negative feedback loop works as a negative modal power supplying the dynamic system.

This technique assures a high stability of the cutting process but is very expensive. Also the passive damping can be done, using the inertial damping as proof mass or impact damper (Ema & Marui, 2000) and tune mass damper (Chen, 2010). This kind of dampers works as a non-optimal narrow band modal energy absorber.

Our research is focused on the possibility to use a wide band frequency passive modal absorber placed inside the boring bar, acting as a tune mass damper based on a viscoelastic suspension.

A dynamical simulation in Simulink is used to understand and to illustrate the chatter mechanism by regenerative effect and chatter correction by negative feedback.

An experimental setup based on a cantilever beam and a passive dynamic absorber placed at the free end of the beam is used to prove the high efficiency in damping of the first vibratory flexural mode.

In the future we intend to design, to manufacture and to test in boring process a low cost boring bar with a high efficiency passive dynamic absorber placed inside. Also we intend to explore a modal energy absorber based on piezoelectric transducer supplied with negative active modal power (Horodinca et al., 2011).

2. DYNAMIC SIMULATION OF CHATTER IN BORING PROCESS

This simulation was done using the Matlab/Simulink dynamic model described in Figure 1.

The dynamic model of the boring tool (as a single degree of freedom vibratory system, see Transfer Fcn block) is the main part of a model with two feedback loops. First feedback loop simulate the regenerative effect, with a proportional feedback (the gain =1) and a delay of 0.3 s. The delay is equal with the rotation time of the workpiece or tool (here the speed is 200 rpm). Second loop works as a proportional derivative feedback loop (Merrit, 1965).

According to (Håkansson & Andrén, 2004) on the real system a velocity-force negative or positive feedback with collocated sensor and actuator can be used to replace this second loop.

According to the hypothesis of regenerative effect (Stepan, 2001), the dynamic model needs only one initial impulse excitation as an external perturbation (e. g. a positive or a negative impulse generated by the variation of the mechanical properties of the workpiece material).

Using a positive gain of 0.001 in the feedback loop 2 (acting as a synthetic negative damping generator on the boring bar), the model produces an output signal on the oscilloscope described on Figure 2.

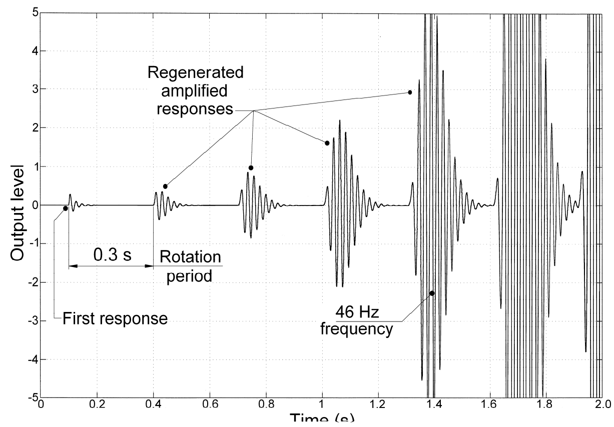

Fig.2. Dynamic response with +0.001 gain in the feedback loop 2 (negative synthetic damping)

It is clear that at each rotation, the amplitude of the response increases very quickly, and in a very sort time the output signal is a continuous vibration on the excited frequency (and constant amplitude, because of the saturation block placed in first feedback loop).

This saturation block reproduces a well known behaviour in practice; the amplitude of chatter is not infinite, because the positive modal power which supplies the vibration is not infinite. Roughly the same behaviour is obtained if the gain is zero (chatter with regenerative effect). The positive feedback on the loop 2 decreases the stability of the boring bar. If a negative gain -0.001 is used in feedback loop 2 then a synthetic, active positive damping is added in the boring bar dynamic system.

In other words a negative modal power supply actuates the system in order to assure the stability condition given in Eq. (1).

The model from Figure 1 produces an output signal on the oscilloscope described on Figure 3.

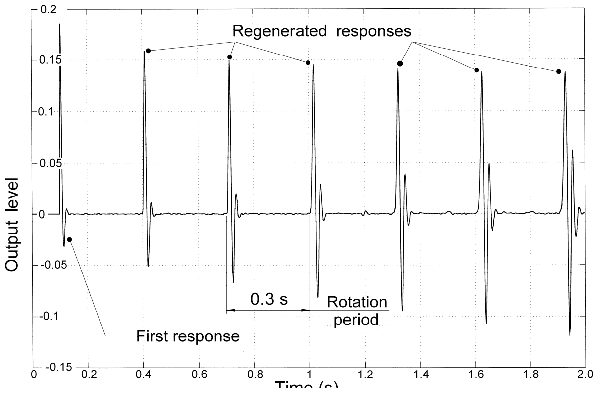

Fig.3. Dynamic response with -0.001 gain in the feedback loop 2 (active positive damping)

The stability is strongly increased (the magnitude on Figure 3 is 28 times smaller than the magnitude on Figure 2) but the system is not yet stable. The boring bar dynamic system becomes stable if a -0.002 gain (or smaller) is used in feedback loop 2.

The chatter can be eliminated also if the modal attenuation (the damping ratio ξ, as a parameter of the transfer function) of the boring bar is increased more. There are two ways to increase the damping ratio ξ:

● Using synthetic damping actively generated (collocated piezoelectric sensor and actuator placed in the node of vibration and negative feedback between). The synthetic damping actively generated supposes to actuate the boring bar with a negative modal power supply (Horodinca et al., 2011) According to the simulations (Figures 2 and 3) the active damping has high efficiency but the equipment is very expensive especially for rotating boring bars.

● Using passive techniques with dampers placed in anti-nodes of vibration, where the amplitude of flexural vibration is maximal. Unfortunately it is not possible to use a classical damper; it needs a fixed point (not available in boring). But it exist a well known solution, the passive tune mass damper (Den Hartog, 1965, 1985), acting as a *sky-hook damper* (doesn't need a fixed point) or modal energy absorber. This damper works as a narrow band passive negative modal mechanical power supply (or passive dynamic modal power absorber) if is tuned on the frequency of the excited mode of vibration.

The main challenge of this paper is to explore the resources of this type of damper placed inside the boring bar.

Nevertheless in drilling of small diameter deep holes, the tool (so called gundrill) uses damping generates by friction torque in cutting area, in order to eliminate self-generated torsional vibration of the tool.

3. MODAL ATTENUATION USING PASSIVE DYNAMIC ABSORBER

In a very short definition, a passive dynamic absorber (PDA) or tune mass damper as well, is a single degree of freedom vibratory system (spring-mass-damper). If PDA is excited on resonance frequency, it absorbs and dissipates (as heating) a big amount of modal energy because the mechanical impedance is minimal on this frequency. If PDA is placed on a structure (e.g a boring bar) excited with harmonic force, then it works as a modal energy absorber. If the structure has a low damped vibration mode and PDA is tuned to have the resonance on the frequency of the mode (co-resonance condition) then PDA act as a passive damper (modal energy absorber, or negative modal power supply as well). PDA absorbs and dissipates modal energy from the structure. Of course PDA should be placed on the structure in a place

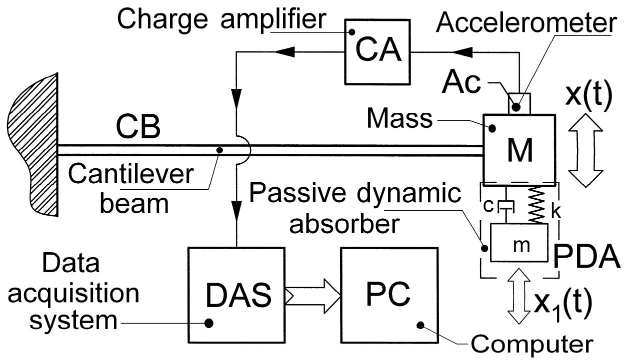

Fig. 4. A sketch of the experimental setup

where the vibration amplitude is maximal (in an anti-node). A PDA placed on a boring bar should solve the stability problem of cutting: *the force and the relative velocity between the tool and workpiece measured on cutting depth direction must be permanently in opposite direction*. In other words, the boring bar doesn't vibrates during the cutting process if there are not available any positive modal power supply. At least the damping (with negative modal power P_n) and self-excitation (positive modal power P_p created during the boring process) should satisfy the Eq. (1). In these conditions no modal energy stored in the boring bar, no vibrations occur.

3.1 Experimental setup

In most cases, in chatter are involved the vibrations on the first flexural (bending) mode of the boring bar. The easiest experimental approach on this subject can be done using a setup which consists of an equivalent mechanical structure for a boring bar: a cantilever beam **CB** with a mass ***M*** (the equivalent of the modal mass on first mode) placed on the free end of the beam, according to Figure 4.

The passive dynamic absorber PDA is placed on the mass *M*, where the amplitude of the flexional vibration ***x(t)*** is maximum). A Brüel & Kjær 4305 accelerometer **Ac**, a charge amplifier **CA**, a data acquisition system **DAS** and a personal computer **PC** are used to describe and to analyse the flexural motion *x(t)* (as free response at mechanical excitation induced by mechanical impulse excitation). The charge amplifier CA and the data acquisition system DAS are included inside of a digital oscilloscope ADC 212/50 (PicoTechnology, UK).

3.2 Passive dynamic absorber design

According to Figure 5 the mass *M* is an aluminium cylindrical part with a central hole inside. The mass *M* is fixed by screws at the free end of the cantilever beam. The mass ***m*** of PDA consist of two cylindrical parts (1, 2) and two cylindrical rare-earth NdFeB (neodymium-iron-boron) permanent magnets 3 (axial magnetic field), with a gap ***d*** relative to the mass *M*. An elastic suspension ES made from a viscoelastic polymer (Sorbothane) with spring stiffness *k* and damping coefficient *c* is used to place the mass *m* inside the mass *M*. The relative motion between the masses *m* and *M* (due to vibration) generates damping in polymer. Also the magnets 3 generate eddy-currents in the material of the mass *M*. The interaction between eddy-currents and the magnetic field generates also an adjustable mechanical viscously damping effect. If the PDA is tuned properly (in order to have a resonance frequency close to the resonance frequency of the elastic system of the cantilever beam, the co-resonance condition), then PDA absorbs a maximal amount of modal energy, acting as a narrow band damper for the cantilever beam system (Horodinca et al., 2011). Usually the PDA frequency is tuned by changing the mass *m* (the resonance frequency is inverse proportional with the mass *m*).

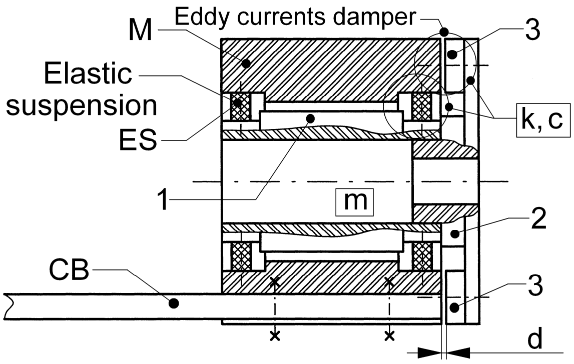

Fig. 5. Cantilever beam setup with passive dynamic absorber

If the cantilever beam is excited at resonance (first flexural mode) then there is a relative motion between the masses *m* and *M* (π/2 shift of phase between *x(t)* and $x_1(t)$ motion, according to Figure 4).

Figure 6 presents a view on the experimental setup (the part 2 and magnets 3 was removed; the length of the cantilever beam was reduced).

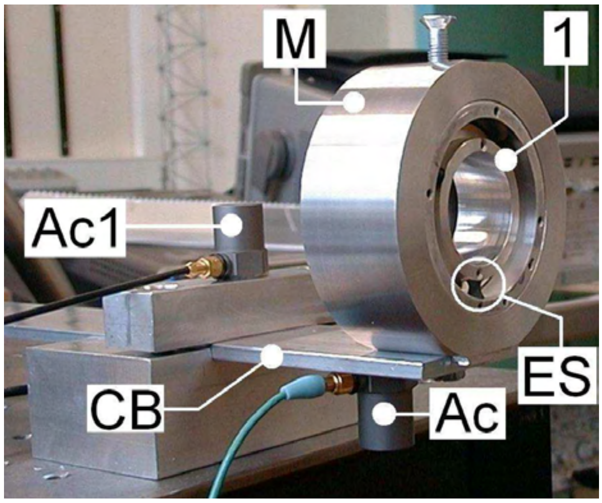

Fig. 6. A view of the cantilever experimental setup

3.3 Experimental results and discussions

Figure 7 presents the results of curve fitting of the experimental viscously damped free response of the cantilever beam with PDA (I, see a zoom-in in detail A, 45.8 Hz, $\xi=15.1\%$) and without PDA (II, see a zoom-in detail in B, 58.1 Hz, $\xi=0.18\%$). The increasing of the damping ratio ξ is very high. The frequency of vibration with PDA decreases because of the additional mass *m*.

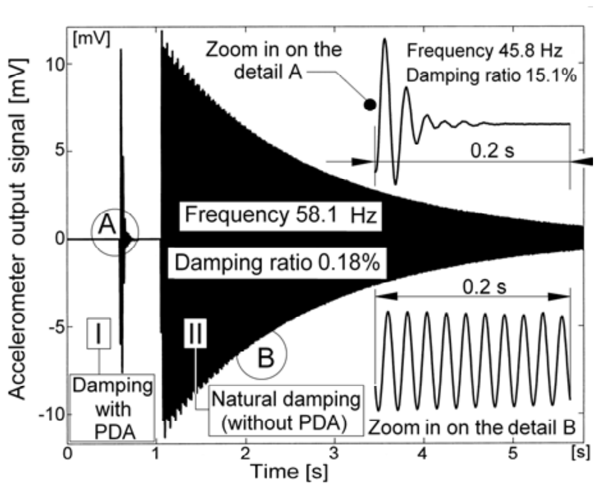

Fig. 7. The PDA efficiency (optimal tuning) in time domain, mirrored in viscously damped free response (I). Viscously damped free response of the cantilever beam without PDA (II).

The free responses are described by the electrical signal (voltage, *U*) generated by the accelerometer Ac.

The experimental free response of the cantilever beam is used to find out the frequency *f* and the damping ratio ξ, by curve fitting (signal identification).

This response is described by the following equation:

$$U(t) = u_0 \cdot e^{-\xi p t} \cdot \sin(p \cdot t + \varphi) \tag{2}$$

This equation is an image of the flexural motion of the cantilever beam:

$$x(t) = x_0 \cdot e^{-\xi p t} \cdot \sin(p \cdot t + \varphi) \tag{3}$$

The curve fitting suppose to find out the values u_0, ξ, p and φ, in order to have the best fit of the theoretical response (Eq. (1)) on the experimental response. The frequency vibration f of the free response is approximated as $f=p/(2\cdot\pi)$. For this purpose a computer program (written in Matlab) was used.

The results of curve fitting for the free response of the cantilever beam with PDA (optimal tuning, see the region I from Figure 7) is described in graphical terms in Figure 8. The identified signal fit on the experimental signal with reasonable precision.

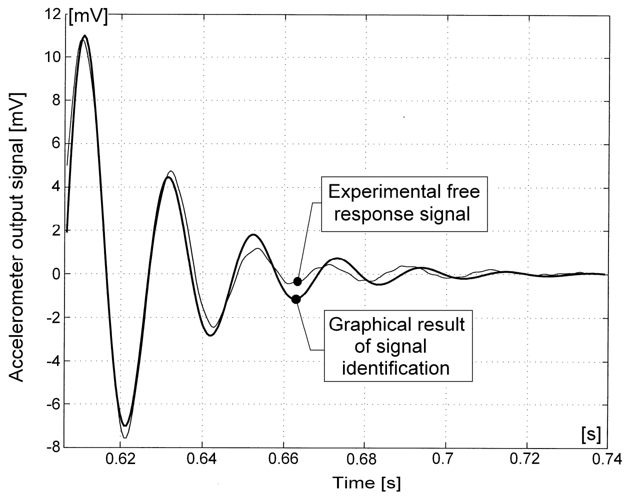

Fig. 8. Graphical result for signal identification ($u_0=3.81\cdot10^9V$, $\xi=15.1\%$, $f=45.8$ Hz, $\varphi=5.0781$ rad)

Based on these two parameters (frequency and damping ratio), the theoretical evolution of the relative resonant amplification of the free end of the cantilever beam in frequency domain can be deduced. The relative amplification X_o/X_i is the amplitude of the response X_o, measurable with the accelerometer Ac, divided by the amplitude of the excitation X_i, measurable with the accelerometer Ac1, see Figure 6. X_o/X_i is also called resonant amplification factor, if we suppose that a harmonic excitation is used.

The relative resonant amplification X_o/X_i is described by the following equation:

$$\frac{X_o}{X_i} = \sqrt{\frac{1+(2\cdot\xi\cdot\eta)^2}{(1-\eta^2)^2+(2\cdot\xi\cdot\eta)^2}} \tag{4}$$

Here $\eta=\omega/p$ is the relative angular frequency, ω is the angular frequency of the excitation, p is called natural angular frequency. The efficiency of PDA in frequency domain is described in Figure 9. The free response analysis was done in order to find out the frequency and damping ratio, for plotting (using Eq. (4)) the evolution of resonant amplification factor (transmissibility), for different values of modal frequency (CB with variable length).

The curve Af_{1n} describes the evolution of the resonant amplification factor without PDA; the curve Af_{1d} describes the same evolution but with PDA, optimal tuned (co-resonance). A maximum attenuation of 38 dB (or 78.4 in linear magnitudes) is assured. The PDA works as a wide band absorber; even if the modal frequency of the cantilever beam is changed it keeps a high efficiency (according to the others curves, Af_{2n} and Af_{2d}.... up to Af_{5n} and Af_{5d}).

This mean that for a boring bar with a PDA placed inside, the tuning of the damper (co-resonance condition) is not so critical. The modal frequency of the first flexural mode of the boring bar is sometimes slightly changed during the boring process because the contact with the workpiece.

4. CONCLUSIONS

The passive dynamic absorber based on viscoelastic suspension is an important challenge which is able to solve the dynamic instability of the cutting processes with boring bars, due to the first flexural vibration mode. The paper illustrates in experimental terms on an equivalent setup the efficiency of a low price damper technique in time and frequency domain. Simple scientific research equipment for data processing was used. According to Figure 9, a maximum attenuation of 38 dB was proved (the resonant amplification factor decreases by a factor of 74.8). The PDA assures more than 20 dB attenuation

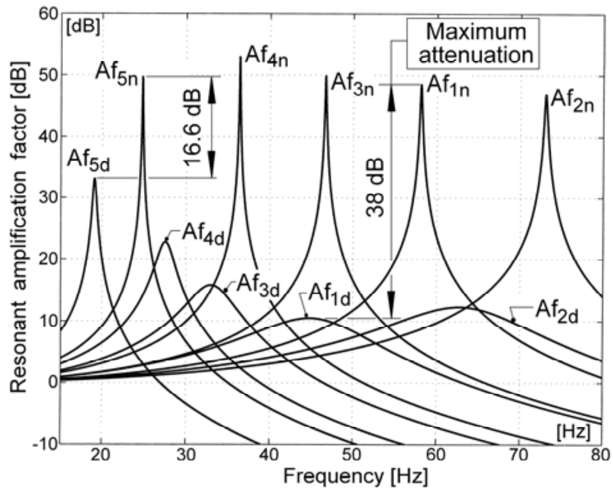

Fig. 9. The PDA efficiency in frequency domain, mirrored in the evolution of the resonant amplification factor (the magnitude in decibels)

(the resonant amplification factor decreases by a factor of ten) even if the modal frequency changes with ± 30 Hz around the optimal frequency. It works as a wideband passive dynamic absorber.

The passive dynamic absorber should be placed inside the boring bar. It is just a matter of design, manufacture and a preliminary experimental study of optimal tuning (of the parameters *m*, *k* and *c*, for co-resonance condition). According to Figure 5 the parts

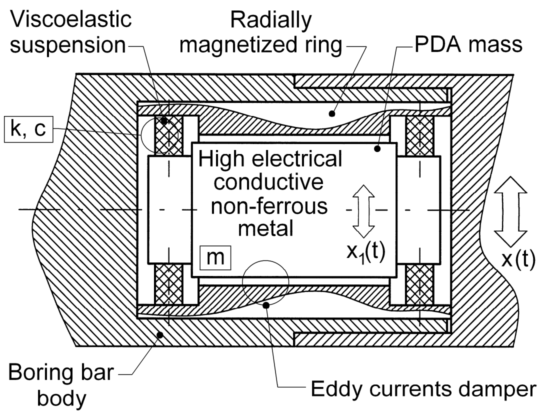

Fig. 10. The PDA setup inside the boring bar

1, 2, 3 and the elastic suspension ES should be placed inside the boring bar, close by the free end. The part 2 can be eliminated if the magnets 3 are placed inside the mass *M* (made of mild steel), close by the part 1 which is manufactured using a heavy material with good electrical conductivity (copper or aluminium and lead). Figure 10 describes (as an idea) how to build and to place the PDA inside the boring bar. The mass *m* is a cylindrical piece made of a high electrical conductive non-ferrous metal. It is placed inside a radially magnetized ring (neodymium-iron-boron permanent magnet). Few cylindrical pieces made of viscoelastic polymer are used as elastic suspension (and dissipative elements as well). One side of each piece is glued on magnetized ring; the other side is glued on the PDA mass *m*. The relative motion between the mass *m* and the magnetic ring provides a high damping ratio (eddy currents damping), extremely important for PDA efficiency. The intensity of magnetic field increases because the boring bar is made by iron or mild steel. The PDA (as a single body consisting of the radially magnetized ring, viscoelastic suspension and the mass *m*) should be placed inside the boring bar as close as possible by the free end. The boring bar is a cylindrical body with a rigidly fixed end. At free end the cutting part the PDA is placed. The rotational symmetry of the PDA assures a high efficiency in damping of flexural vibration even if the radial direction of the vibration of free end is variable, or if the boring bar vibrates simultaneously on two direction (e. g., according to Figure 10, one direction is *x(t)* the other is perpendicular to *x(t)*). This damping technique is useful for many other mechanical structures (tall buildings, towers, flexible bridges, cranes, etc.).

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