Active Vibration Control of Gradient Coils to Reduce Acoustic Noise of MRI Systems


Abstract—Lorentz-force induced vibrations in Magnetic resonance imaging (MRI) systems cause significant acoustic noise levels during scanning, the main acoustic noise source being the vibrating gradient coil. In this paper a novel active vibration control technique is presented to reduce vibrations of the gradient coil and hence achieve a reduction of acoustic noise during scanning. The active vibration control technique uses seismic masses that are actuated by means of piezo actuators to create forces on the gradient coil counteracting its vibrations. Using 4 seismic mass actuators, a vibration reduction of 3 to 8 dB at resonance frequencies is achieved, giving an overall vibration reduction of 3 dB for a typical FE-EPI gradient sequence, as substantiated by measurements. Using 8 actuators, an overall vibration reduction of 5 dB is predicted for this sequence.

Index Terms—Acoustic noise, Active vibration control, Magnetic Resonance Imaging.

I. INTRODUCTION

MagNetic resonance imaging (MRI) scanners are widely used in hospitals for both medical diagnosis and clinical research. Nowadays magnetic field strengths as high as 3 Tesla are common, allowing a high spatial imaging resolution using fast scanning sequences. However, with increasing magnetic field strength, the acoustic noise generated during scanning increases also.

The acoustic noise produced by MRI systems is a cause for growing concern. Price et al. [1] give an overview of the acoustic noise produced by commercially available MRI systems ranging in field strength from 0.2T to 3T. The acoustic noise levels reported varied from 85dBA for 0.2 - 0.5T systems to 115dBA for 3T systems in the case of fast pulsed sequences. They also point out that the increasing gradient field levels and slew rates are pushing these values even higher, and that levels up to 130dBA have been reported for 3T systems. Needless to say, a reduction of the acoustic noise levels is desirable to avoid discomfort or even hearing loss of both medical personnel and patients.

Various noise reduction techniques have been proposed in literature. Mannsfield et al. [2] proposed a revolutionary gradient coil design, using additional coils to counteract the vibrations of the gradient coil carrier. Special care has to be taken to stiffen the gradient coil carrier, to avoid adverse effects. Current state of the art of this technology is that a significant acoustic noise reduction can be obtained of an unprecedented 50 dB within a relatively narrow frequency range for a given coil carrier material [3].

Edelstein et al. [4] as well as Katsunuma et al. [5] propose the design of a vibration-isolated gradient coil assembly contained in an airtight enclosure. Katsunuma et al. [5] claim an acoustic noise reduction of more than 10 dB, with noise reductions up to 23 dB if the coil is independently supported. Edelstein et al. [4] also used a low-eddy-current radio frequency (RF) coil and a non-conducting inner bore cryostat to reduce eddy current induced vibrations.

Moelker et al. [6] used passive sound insulation to reduce the acoustic noise radiating from the patient bore covers, leading to noise reductions up to 18.8 dB.

Active noise cancelation was first proposed by McJury et al. [7], followed shortly by Goldman et al. [8] and later on by Chen et al. [9]. The proposed technique uses anti-phase acoustic waves to create a zone of destructive interference. In order to reduce electro-magnetic interference (EMI) and radio-frequency interference (RFI), the use of metal parts such as copper leads, even small ones, need to be minimized. A recent development is the use of an opto-acoustical transducer that operates on the principle of light modulation [10]. These are immune from, and do not create, EMI and RFI, which is especially important for functional MRI.

Usually feedback controllers are used for active noise cancelation. The time required for the control signal to propagate from the loudspeaker to the subject, and the necessity for maintaining stability, means that the upper frequency for feedback systems is commonly limited to a few hundred Hertz [10]. For this reason noise cancelation systems are combined with passive hearing protection devices which give a good attenuation at high frequencies, which is complementary to the noise reductions that can be obtained by active noise control systems at low frequencies.

The effectiveness of present active noise cancelation systems in conjunction with hearing protection devices that reduce sound transmission through the ear canal is limited by noise conducted through the head and body that bypasses these treatments [11].

An alternative approach is to use active structural acoustic control, in which the sound radiating structure is controlled in an active manner, leading to acoustic noise reduction in a global sense, and thus not suffering from the limitations due to head and body conduction. Though there is a vast literature on this subject in general, only a few research groups have published on this in relation to an MRI scanner. Trajkov et al. [12], [13], [14] report on a study to control the vibrations of funnel-shaped covers, such as those of an MRI system,
by means of piezo-electric patches. Koevoets et al. [15], [16] have reported on measurement based techniques for improved modelling of piezo-electric patches in similar applications.

Roozen et al. [17] implemented an active isolation system to support a gradient coil assembly. The goal of this work was to reduce the acoustic noise radiated by the main magnet structure that is excited via a structure-borne path by the vibrating gradient coil, reporting reductions of 10 dB of said structure-borne noise transmission path.

This paper describes a novel active vibration control system to suppress the vibrations of a gradient coil assembly, leading to a reduction of the radiated acoustic noise levels in a global sense.

The paper is organized in the following manner: Sec. II describes the main principles of structural excitation in MRI systems. Analytical and experimental analysis of the structural dynamical behavior of the gradient coil assembly is used to identify the most relevant resonances of the gradient carrier. In Sec. III, a novel MRI compatible actuator is presented that is used to reduce the vibrations of the gradient coil. A feed-forward control design strategy based on measured frequency response function (FRF) data is presented that is able to achieve vibration reduction in a global manner. Experimental results presented in Sec.III-C show that a reduction of the structural vibration levels of the gradient coil up to 10 dB at the structural resonances of the gradient coil carrier can be achieved with the proposed approach.

II. GENERATION OF ACOUSTIC NOISE

To understand how acoustic noise is generated in a MRI system, some basic knowledge about a MRI system is required. Basically, a MRI system consists of a main magnet system, a gradient coil system and a RF-coil system. These main components will be discussed shortly.

The purpose of the main magnet is to create a static magnetic field. Extreme care is taken to ensure the uniformity of this static field in the scanning volume.

The gradient coil system (shown in Fig. 1) creates a highly linear magnetic gradient field. Three gradient fields can be created, i.e. the X-, Y- and Z-gradient fields, which vary linearly across the X-, Y- and Z-direction, respectively. The X-gradient coil is basically a three-dimensional saddle coil. Fig. 2 shows the windings of an X-gradient coil without carrier structure. The Y-gradient coil is similar to the X-gradient coil, rotated 90 degrees. The X- and Y-gradient coils cause magnetic gradients in the X- and Y-directions, respectively. The Z-coil is a rotationally symmetric coil, causing gradients in the Z-direction. By means of these gradient fields, specific slices of the human body can be selected and the magnetic resonance signal can be localized [18].

Finally, the RF-coil system creates a radio frequency signal in the order of tens of MHz (e.g. 64 MHz for a 1.5 Tesla MRI system, its value being dependent upon the static magnetic field of the system), which excites the atomic nuclei of the selected slice at their resonance frequency.

During scanning the magnetic gradient field is switched on and off by means of alternating gradient coil currents, called gradient sequences. As the main magnetic field is in the order of a number of Tesla’s (currently 3T for most machines), and the gradient coil currents in the order of 600 Amps, the Lorentz forces acting upon the gradient coil windings are in the order of 2000 N per meter coil winding. Knowing that the total...
Fig. 5. Noise transmission paths

winding length of an X- or Y-coil amounts approximately hundred meters, the Lorentz forces are huge, resulting in vibrations with acceleration levels in the order of \(100 \, \text{m} \, \text{s}^{-2}\). As a result acoustic noise is caused during scanning.

A. Noise transmission paths

The gradient coil vibrations cause acoustic noise to be radiated by the gradient coil system directly, via the air borne noise transmission. The direct transmission path is shielded by the covers of the system, which is in most cases insufficient.

Besides the direct radiation of the gradient coil also indirect transmission paths exists. Via structure borne paths the cryostat structure that supports the gradient coil is brought into vibration, which then radiates acoustic noise. Attenuation of this transmission path can be obtained by a compliant mounting of the gradient coil, which is not always possible because of the significant mass of the gradient coil and requirements with respect to the position of the gradient coil relative to the patient and cryostat. An alternative approach is to use active vibration isolation control [17].

A third transmission path exists in MRI systems, which is a magnetic borne path. The main function of the gradient coil is to create time varying gradient fields. These gradient fields will cause Eddy currents in electrically conducting parts of the MRI system, which can cause significant Lorenz forces due to the presence of a very strong static magnetic field emanating from the main magnet. Especially the vibrating cryostat structure, encompassing the main magnet, will vibrate significantly because of this transmission path. Tackling the magnetic borne path was done by Biloen et al. [19] and Edelstein et al. [20] using a copper screen wrapped around the gradient coil to shield the cryostat structure from magnetic fields radiated by the gradient coil. The effectiveness of the magnetic shielding was about 20dB for the cryostat bore structure, and about 10dB for the front and back sections of the cryostat structure. In Fig. 5 the three transmission paths are summarized.

The relative importance of the transmission paths changes with system design. Whilst for systems with an aluminium magnet housing the magnetic transmission path can be significant, for systems having a stainless steel magnet housing this transmission path is less strong. The direct acoustic radiation of the gradient coil strongly depends upon the central bore covers. For most MRI systems the direct air borne transmission path is the most dominant path as the relatively thin covers used in the patient bore can not easily reduce the acoustic noise. Therefore the focus of this paper is on the air borne transmission path. The aim is to reduce the vibrations of the gradient coil itself by means of an active vibration suppression system, so as to reduce the usually dominant air borne noise transmission path.

B. Gradient coil dynamics and acoustic noise

In order to design the active vibration suppression system, the excitation of the structural dynamics of the gradient carrier is analyzed. This knowledge will be used in Sec. III-A for actuator placement.

For analysis of the structural dynamics of the gradient carrier, the concepts of structural mode-shapes is used [21]. In order to apply this technique, it is assumed that the structural dynamic behavior is lightly damped or satisfies Raleigh-damping. As a result, the dynamic response can be written as a super-position of structural mode-shapes:

\[
Y_i(j\omega) = \sum_k \frac{\phi_i\phi_j}{s^2 + 2\xi_k\omega_n + \omega_n^2} U_j(j\omega) \quad (1)
\]

where \(\phi_j\) and \(\phi_i\) represent respectively the mode-shape at the location with index \(j\) where force \(U_j(j\omega)\) is applied and the location with index \(i\) where the displacement \(Y_i(j\omega)\) is measured. \(\omega_n\) and \(\xi_k\) represent respectively the modal
Fig. 8. Gradient radial mode shape

Fig. 9. Gradient coil square mode shape

eigenfrequency and relative damping of mode \( k \). It can be observed from Eq.(1) that the frequency response of a certain point depends on the inner-product between a force distribution and the mode-shape, and the inner-product between the observation point and the mode-shape. Since we are interested in vibration suppression over the entire structure, we will mainly focus on the first part, i.e. the inner-product between force distribution and structural mode-shape. Though a large number of structural modes exist, only a few modes of the gradient coil structure are excited by the Lorentz forces due to the specific spatial distribution of these forces. The modes that are excited most dominantly will be discussed shortly. This analysis is based on numerical simulations using the finite element method (FEM).

The Galerkin finite element method is used to predict the dynamic behavior of the gradient coil structure. Half the structure is modeled with 4-noded isoparametric tetrahedral elements (ANSYS element type “Solid92”), exploiting symmetry in Y-direction (see Fig. 6). Using tetrahedral solid elements, the high stiffness of the gradient coil structure is taken into account correctly. As the gradient coil structure consists of many different materials, each material layer is modeled separately, having appropriate material properties assigned to it. Some of the structural eigenmodes that were extracted with this model are shown in Fig. 7, Fig. 8 and Fig. 9.

The Lorentz force distribution being developed in the X- and Zcoil windings are schematically depicted in Fig. 3 and Fig. 4, respectively. This explains the relative effective excitation of the mode as shown in Fig. 7 by the Lorentz force distribution of the X- and Y-coils since the inner product of Lorentz forces for the X- and Y-coil and this bending mode is quite high. Two bending modes exists for a gradient coil structure, one which is vibrating predominantly in the X-direction and the other which is vibrating predominantly in the Y-direction. Besides the bending modes also other modes are excited by the X- and Y-coils. One such mode is shown in Fig. 9, which has a mode shape which shows both bending and radial movements.

The Lorentz force distribution of the Z-coil typically excites the radial mode as shown in Fig. 8. This mode is an axi-symmetric mode, which is one of the reasons why this mode is excited by the axi-symmetric Lorentz forces created by the Z-coil. The Lorentz forces acting upon the Z-windings are shown in Fig. 4. It is again the inner product of the Lorentz forces of the Z-coil and this radial mode which is quite high, resulting in a strong excitation of this specific mode by the Z-coil.

Experiments were carried out to allow a comparison of the numerical results to experimental results. Frequency response function estimates (FRF’s) were measured from the X-, Y- and Z- gradient coil currents to the structural velocity at 120 positions equally distributed on the inner wall of the gradient coil carrier. The measurements were performed using a 4 channel Siglab digital signal analyzer, a Bruel and Kjaer conditioning amplifier type Nexus and three Bruel and Kjaer accelerometers type 4393V. The FRF’s are presented in arbitrary units (a.u.) for both the frequency axis and the response axis.

The dominance of the bending mode and combined bending-radial mode for the X- and Y-coil, as shown in Fig. 7 and in Fig. 9 respectively, as well as the dominance of the radial mode for the Z-coil excitation as shown in Fig. 8 is clearly visible in the measured structural response of the gradient carrier. Fig. 10 shows the measured FRF’s from the Y-gradient coil current to the structural velocity at the 120 positions on the gradient coil. The spectra for the X-coil excitation are quite similar. Fig. 11 shows the FRF from the Z-coil input to the gradient coil acceleration. In Fig. 10 and Fig. 11, a number of dominant peaks in the spectrum are marked with the letters A through E, which correspond to the operational deflection shapes (ODS) given in Fig. 12 and Fig. 13. These figures show the response of the gradient coil at the specified resonance frequencies A through E as measured at the 120 postions on the inner side of the gradient coil carrier.

The resonance peaks A and B correspond to bending modes, as shown in Fig. 7. The asymmetry caused by the gradient support structure, which was not incorporated in the analytical model, produces two bending modes having different eigenfrequencies. Their operating deflection shapes for Y-gradient coil actuation are shown in Fig. 12.

Resonance peak C corresponds to the combined bending-radial mode shape, its operational deflection shape shown in Fig. 13 and the corresponding numerical mode shown in Fig. 9. The symmetry of the combined bending-radial mode shown in Fig. 9 suggests that this mode will hardly be excited by the X- or Y-coils due to reasons of symmetry, causing the inner product of Lorentz forces for the X- and Y-coil and the
square mode to vanish. In practice it is excited, again due to the asymmetry caused by the gradient support structure. Note that peak C is less significant than peak B or D.

Resonance D corresponds to a second order bending mode, having a higher spatial wavenumber in the axial direction, as shown in Fig. 13.

Resonance E (Fig. 11) corresponds to the radial mode of which its shape does not vary as function of the circumferential coordinate. Its operation deflection shape for Z-coil excitation is given in Fig. 13. The corresponding mode shape obtained via FEM analysis is shown in Fig. 8.

III. ACTIVE VIBRATION CONTROL GRADIENT COIL

The design of an active vibration control system needs to fulfill a number of requirements for application in an MRI system. First of all the actuator needs to be MR compatible, meaning that it may not disturb the magnetic field as this would degrade image quality. This requires the use of non-magnetic materials, such as aluminum or specific, non-magnetic, stainless steel materials and plastic materials if possible.

In addition, the currents that are possibly used for actuation may not disturb the magnetic field as well. Bearing in mind that the forces required to counteract the vibrations of the gradient coil are in the order of a few hundred Newtons, this will not be an easy task. Lorentz actuators are for this reason not permitted. Piezo actuators, however, are MRI compatible, as the currents that are used to actuate the piezo are very small. Piezo’s have a very high electrical impedance, and it is only the electrical charge that is actuating the piezo. The PZT piezo material as such also appears to be MRI compatible.

Furthermore the actuator needs to be compact. The first reason for this requirement is a practical reason: there is little space available. Covers are often close fitted around the gradient coil and cryostat, limiting the space to centimeters. Another reason for this requirement is that a compact design will not suffer from parasitic resonances in the frequency range of operation as these resonances will degrade actuator efficiency.

The actuator should be able to deliver a force of sufficient magnitude over a broad frequency range. As argued earlier the actuation force is in the order of a few hundred Newtons. This force should be delivered at frequencies between 500 and 1500 Hz. In practice the strongest acoustic noise levels occur within the mentioned frequency range.

Finally, the design should be such that it is "force balanced", meaning that the forces that are created by the actuator are actuating upon the gradient coil only, and not upon other parts of the MRI system. The reason for this requirement is that significant vibrations in the "other parts" of the MRI system could occur otherwise, resulting in a significant contribution to the acoustic noise levels, which is obviously not desired.
All the above requirements led to the design of a seismic mass piezo actuator, which is described in the next section.

A. Seismic mass piezo actuator design

The basic principle used in the seismic mass piezo actuator design is the principle of inertia. A piezo actuator is mounted between the gradient carrier and a seismic mass, its acceleration causing an inertial force. This design is compact, force balanced and MRI compatible.

Fig. 14 shows a CAD drawing of the seismic mass piezo actuator. The base plate is mounted on the gradient coil ends. Actuating the piezo will cause a force being transferred to the gradient coil through the base plate. Guidance plates are used to guide the seismic mass during its movements. Fig. 15 shows the individual parts of the actuator.

The locations of the actuators on the gradient carrier structure are chosen such that all dominant modes, as discussed in Sec. II-B are controllable with the actuators, i.e. the inner-product between the mode-shape and the force generated by the actuators is relatively large (see Eq.(1)). However, practical considerations constrain the placement of the actuators. Due to availability of construction space in axial direction, placement of the actuators in the Z-direction is preferred.

Force actuation in axial direction seems a little bit strange at first sight. It should however be mentioned that all relevant modes of the gradient coil structure, such as the bending mode shown in Fig. 7, clearly vibrate in both radial and axial directions and are therefore controllable via forces in axial direction.

Four seismic mass piezo actuators were mounted on the gradient coil, two mounted at the front and two mounted at the back of the gradient coil, one at the left hand and one at the right hand side. Fig. 16 shows the front right actuator. The forces exerted on the gradient coil are directed in axial (Z) direction.

B. Controller design

The amount of acoustic noise radiated from the gradient coil is directly related to the velocities at the surface of the gradient coil [22]. The squared velocity integrated over the coil surface and time is therefore chosen as a relevant measure for the amount of radiated noise. Hence, the following objective is used for control design:

$$\min_{u_{\text{piezo-act}}(t)} \int_{-\infty}^{\infty} \int_{-\pi}^{\pi} \int_{z_{\text{min}}}^{z_{\text{max}}} v(\phi, z, t)^2 \, d\phi \, dz \, dt$$  \hspace{1cm} (2)

where $v(\phi, z, t)$ represents the velocity at the surface of the coil at radial location $\phi$, axial position $z$ at time $t$ and $u_{\text{piezo-act}}(t)$ is the vector of input signals to the piezo-actuators.

In order to achieve this goal, both feedback and feedforward control strategies for vibration suppression can be considered [23], [24], [25]. Due to several restrictions imposed by feedback (summarized below), it is motivated that feedforward techniques better fit the characteristics of this particular application. A summary of properties of feedback systems for this application is given.

- Analytical properties of closed-loop feedback systems, e.g. Bode’s sensitivity integral, make that disturbances are typically amplified in the high frequent region, i.e. above the bandwidth. This requires closed-loop bandwidths which are significantly higher than the frequency content of the sequences applied on the gradient coils, typically within the frequency range from a few hundred Hertz up to 1500 Hz. The need for a high bandwidth poses tight requirements on the dynamics of the equipment used in the feedback loop and is expected to result in high cost.
- Creating a sensor that represents a relevant performance measure for noise radiation is not straightforward.
- Knowledge about the main disturbance source permits non-causal control strategies which is not exploited with feedback.
• The high modal density observed in the feedback loop makes that feedback control design for performance (or even stability) is not straightforward.

Due to the complexity of the system, a data-based feedforward approach is pursuit such that modeling of the system can be omitted. To enable practical evaluation, Eq.(2) is discretized in space where the coil is subdivided in a grid of 12 segments in \( \phi \) direction and 10 in \( z \) direction:

\[
\min_{u_{\text{piezo}}(t)} \sum_{i=1}^{12} \sum_{j=1}^{10} \int_{-\infty}^{\infty} v(\phi_i, z_j, t)^2 \, dt
\]

It is expected that the resulting distance between grid-points is small compared to the modal wavelengths in the relevant frequency region, such that all relevant modal shapes are well observed.

For control purpose, linearity of the system and especially of the piezo actuators is assumed. It will be seen later that this restriction can be relaxed for the piezo actuators. As a result, Eq.(3) can be written in the frequency domain via Parseval’s theorem:

\[
\min_{u_{\text{piezo}}(t)} \sum_{i=1}^{12} \sum_{j=1}^{10} \int_{-\infty}^{\infty} |V(\phi_i, z_j, \omega)|^2 \, d\omega
\]

where \( V(j\omega) \) represents the Fourier transformed signal \( v(t) \).

It will be derived that for repeating sequences, the feedforward signal to the piezo-actuators, \( u_{\text{piezo}}(t) \) can be computed based on the measured FRF-data and knowledge of the gradient sequences. The resulting inputs are optimal in the sense of Eq.(4).

According to the frequency separation principle (no feedback is applied such that Bode’s sensitivity integral does not hold), the objective given in Eq.(4) can be minimized per frequency:

\[
\min_{u_{\text{piezo}}(t)} \tilde{V}^\dagger(j\omega) \tilde{V}(j\omega) \quad \forall \omega
\]

where \( \tilde{V}(j\omega) \) is a vector of velocities over the grid of 120 points, \( \tilde{V}_{ij} = V(\phi_i, z_j) \) and \( .^\dagger \) represents the conjugate transpose of \( (.) \).

The output of the system \( \tilde{V}(j\omega) \) can be written as function of the inputs to the gradient coils and piezo-actuator via:

\[
\tilde{V}(j\omega) = H_{\text{coil}}(j\omega) \begin{bmatrix} U_x(j\omega) \\ U_y(j\omega) \\ U_z(j\omega) \end{bmatrix} + H_{\text{piezo}}(j\omega) \begin{bmatrix} U_1(j\omega) \\ \vdots \\ U_n(j\omega) \end{bmatrix}
\]

where \( H_{\text{coil}}(j\omega) \) and \( H_{\text{piezo}}(j\omega) \) represent the frequency response function matrix between actuator inputs and velocities at the coil surface which can be obtained from measurements. \([U_x, U_y, U_z]^T\) and \([U_1, \ldots, U_n]^T\) represent respectively the input to the gradient coils and the piezo actuators. Substitution of Eq.(6) in Eq.(5) gives:

\[
\min_{u_{\text{piezo}}(t)} \left\| H_{\text{coil}}(j\omega) \begin{bmatrix} U_x(j\omega) \\ U_y(j\omega) \\ U_z(j\omega) \end{bmatrix} + H_{\text{piezo}}(j\omega) \begin{bmatrix} U_1(j\omega) \\ \vdots \\ U_n(j\omega) \end{bmatrix} \right\|_{2}^2
\]

Since the input-signals to the coils \([U_x, U_y, U_z]^T\) are known a priori, Eq.(7) can be solved via a least-squares approximation problem of the form \( \min_x \|Ax-b\|_2 \). The solution is given by (for compactness, dependence on \( j\omega \) is omitted):

\[
\begin{bmatrix} U_1 \\ \vdots \\ U_n \end{bmatrix} = -(H_{\text{piezo}}^\dagger H_{\text{piezo}})^{-1} H_{\text{piezo}}^\dagger H_{\text{coil}} \begin{bmatrix} U_x \\ U_y \\ U_z \end{bmatrix}
\]

If the signals are repeating over time, which is most common for MRI sequences, the vector \([U_x, U_y, U_z]^T\) can be described by a finite number of frequency domain coefficients and thus has a finite length. As a result, Eq.(8) can be solved off-line. The feedforward signal that minimizes Eq.(3) is obtained by applying a fast fourier transform on \([u_x(t), u_y(t), u_z(t)]^T\) over the repeating time-interval. This data is substituted in Eq.(8) and the resulting \([U_1, \ldots, U_n]^T\) are transformed back into the time domain via an inverse fast fourier transform to obtain \( u_{\text{piezo}}(t) \).

1) Remarks: In this work we chose to minimize the quadratic sum of the structural velocities as this is a reasonable measure for the acoustic noise radiated by the gradient coil. A better acoustic performance could result by weighing the velocities by the acoustic radiation modes, a concept that was introduced by Cunefare et al. [26],[27],[28], and first applied to the acoustic radiation from MRI gradient coils by Kuijpers et al. [29].

Active suppression of gradient coil vibrations during typical operation using gradient coil currents of 100 Amps requires significant actuator forces. Depending on the frequency of actuation, these forces demand piezo displacements in the order of 10 \( \mu m \). Such high levels of displacements can be realized by means of piezo stacks, but generally give rise to non-linear behavior of the piezo. In [30] a lifted Iterative Learning Control approach was introduced to cope with this problem. This technique suppresses the super-harmonic response of the seismic mass piezo actuator by exploiting the repetitive behavior of the gradient coil sequences. Using this iterative learning control approach, it can be assumed that the response of the piezo stack behaves linearly after a few trials. Hence, even under high excitation conditions of the piezo’s, Eq.(8) can be applied.

C. Experimental results

Four seismic mass piezo actuators were used in the experiments to reduce the vibration levels of the gradient coil. The front right seismic mass piezo actuator is shown in Fig. 16. Similar actuators were implemented at the front left, rear left and rear right of the gradient coil.

In order to apply Eq.(8), the FRF’s from \([U_x, U_y, U_z]\) to \( \tilde{V} \) and \([U_1, \ldots, U_n]\) to \( \tilde{V} \) are measured [31]. This is performed using white noise excitation on the input of the piezo actuator while measuring the velocities with an accelerometer. A dSpace control system combined with high-voltage piezo amplifiers is used to generate the feedforward signals and drive the piezo actuators. In Fig. 17 the FRF’s from one piezo actuator to the accelerations in radial direction of all
120 grid points are plotted in arbitrary units (a.u.) for both the frequency axis and the response axis. It can be observed that $H_{\text{piezo}}(j\omega)$ increases with frequency. This is caused by the fact that the piezo stack can be seen as position actuator due to its high stiffness. As a result, the inertia forces created by the seismic mass increase with frequency for a fixed maximum displacement of the piezo actuators. This property suits the application well since $H_{\text{coil}}(j\omega)$ also shows an increase as function of frequency (see Fig. 10 and Fig. 11).

Minimizing the objective function as described by Eq. 5, using four seismic mass piezo actuators simultaneously, reduces the spatially averaged response of the gradient coil vibrations for Y-gradient fields by 3 to 8 dB at the dominant frequencies (see Fig. 19). For Z-coil gradients the reductions are in the order of 3 to 4 dB at the dominant frequencies, as shown in Fig. 21.

The reduction of the structural vibrations that can be achieved by means of active vibration control is limited for two reasons: the limited stroke of the actuators and the fact that the actuating forces are discrete in nature. By means of the four seismic mass actuators discrete forces are introduced, which will excite other structural modes of the gradient coil carrier than the gradient coil current induced Lorentz forces do. In general, more modes will be excited by the point forces introduced by the seismic mass actuators. These additional modes have a higher spatial wavenumber, compared to the gradient coil current induced vibrational modes. This is illustrated in Fig. 18, which shows the structural response of the gradient coil carrier for Z-gradient current excitation and for piezo actuator excitation. From this figure it can be seen that the gradient current induced Lorentz forces excite a limited number of eigenmodes, whereas the piezo actuators excite many more modes, having high spatial wavenumbers. Whereas in theory the modal contribution of a specific mode can be canceled completely, in practice this is of little value as other modes will then more strongly contribute to the structural response. In effect, the reduction of the spatially averaged structural response is limited, as shown in Fig. 19 and Fig. 21.

Increasing the number of seismic mass piezo actuators will result in higher reductions of the spatially averages structural gradient coil response. To quantify this effect a number of simulations, based on measurement data, have been performed using virtual piezo actuators. To this end the FRF’s between the force exerted by a virtual piezo actuator and the structural response was experimentally determined by means of an impact hammer. Using these FRF’s, combined with the actual data of the four physically present seismic mass piezo actuators, the vibration reduction that can be achieved using eight actuators was computed. The results are shown in Fig. 20.
and Fig. 22 for Y- and Z-coil excitation, respectively. From these figures it can be seen that the achievable reductions range from 7 to 10 dB for Y-coil excitation and from 3 to 7 dB for Z-coil excitation.

In the preceding text the reduction of the gradient coil structural vibrations was discussed in terms of frequency response functions. To estimate the reduction of the gradient coil structural vibrations in practice, the response of the gradient coil was calculated for a typical gradient coil current as function of time. For this purpose an EPI sequence was employed (see Fig. 23) which is known for its high acoustic noise levels [5]. The resulting response of the gradient coil structure in the frequency domain is shown in Fig. 24, without and with control, using 4 actuators. The cumulative power spectrum shows a reduction for the entire frequency range studied of 3 dB. Using 8 actuators 5 dB reduction is obtained.

**IV. CONCLUSION**

Noise generation in MRI systems is a growing cause for concern. As magnetic field strengths and gradient slew rates increase, so does the noise produced by the system. The root cause of the noise generation lies in the interaction between the alternating currents in the gradient coil system and the static main magnet field. The Lorenz forces exerted on the gradient coils lead to vibrations and, via various transmission paths, to noise radiation. In most MRI systems the acoustic noise radiated by the gradient coil carrier itself, the so-called direct transmission path, is most dominant.

In this paper, the design of an active vibration suppression system is proposed which is applied directly to the gradient carrier and hence reduces acoustic noise emission in a direct manner. As a first step, the structural dynamics of the gradient coil carrier are analyzed to identify the dominant structural dynamics and hence determine the optimal actuator placement. In order to generate the forces that counteract the gradient vibrations, a MRI compatible actuator is proposed which consist of a piezo actuator mounted between an inertial mass and the gradient coil. This setup is compact and well suited to deliver forces in the relevant frequency region between 500 Hz and 1500 Hz.

A feedforward control design strategy is described that exploits the repetitive behavior of MRI sequences to generate an optimal input signal for the seismic mass piezo actuators based on measured frequency response data solely such that modeling can be omitted.

Experimental results using 4 actuators show that significant reduction of the vibration levels can be achieved ranging from 3 to 8 dB at the resonance peaks, resulting in a reduction of 3 dB for the entire frequency range studied, using a typical FE-EPI input sequence. The proposed approach however has the potential to achieve even better performance if more actuators or more distributed force patterns are applied. More seismic actuators will represent the Lorenz force distribution better, and therefore counteract the modes excited by the Lorenz forces more effectively while exciting less other modes. A
simulation study with 8 actuators showed reductions ranging from 7 to 10 dB at the resonance peaks, resulting in a reduction of 5 dB for the entire frequency range.

ACKNOWLEDGMENT

The authors would like to thank Philips Medical Systems, and in particular Kees Ham, Hans Tuithof and Peter van der Meulen, for facilitating and financing the research described in this paper.

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