Optimisation, design and characterisation of a piezoelectric micro suspension

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(Received 28 October 2011; final version received 15 December 2011)

A piezoelectric micro active suspension device has been developed for the application of active isolation of sensitive electronic devices such as frequency generators or inertial sensors. The developed strategy is based on a classical skyhook active damper but adapted and optimised to allow robust implementation onto an adaptive micro-electromechanical systems (MEMS). The micro suspension is a silicon beam structure etched in a silicon on insulator (SOI) wafer and is equipped with a pair of piezoelectric transducers obtained from a lead zirconate titanate thin film sandwiched between pairs of electrodes. High performance transducers allow application of the active isolation strategy and demonstrate the possibility to implement skyhook damping with low-voltage control level.

Keywords: adaptative MEMS; micro active suspension; piezoelectric thin film; integrated piezo-MEMS technology

1. Introduction

Transport system evolution is following two major trends motivated by economic, ecological and safety requirements. Lightweight materials are widely used to reduce fuel consumption and increase available space in transportation systems. At the same time, miniaturised electronic components are integrated onto these structures to carry out strategic functions, such as communications, navigation, power train performance management, passenger safety or structure health monitoring. But transport also results in intense vibrating environments that strongly affect the reliability and service life of embedded microelectromechanical systems (MEMS) such as frequency generators or inertial sensors used for strategic functions such as steering assistance, communication, engine performance management, structure health monitoring, guidance and navigation [1–4].

Stabilisation techniques currently meet fundamental limits. Addition of passive materials with damping capabilities such as elastomeric or viscoelastic materials drastically increases the mass and the size of the device’s packaging, contradicting the technical requirements [5]. Another solution consists of adding damping in the sensitive component itself to prevent its internal dynamics from reaching critical values. Most of the time, a dissipative mechanism is introduced by fostering interaction between the mechanical structure and a gas enclosed in a sealed (or hermetic) package [6,7]. This solution, commonly used in...
capacitive or piezoresistive MEMS accelerometers, still remains too intricate to generalise to other MEMS applications. For these reasons, using a mechanical suspension between a sensitive component and its bearing structure becomes very attractive. The transmissibility of a passive mechanical suspension is depicted in black in Figure 1. It naturally filters high-frequency accelerations and prevents them from reaching the sensitive component located on the terminal payload. Low-frequency acceleration, corresponding to rigid body movements of the support, is transmitted to the isolated component that keeps its configuration in the inertial frame. Undamped suspensions (dashed line) have the best isolation performances but cannot be used due to the transfer magnitude amplification occurring at the resonant frequency of the isolator at $\omega_0 = (k/m)^{1/2}$. Passive damping limits the amplification at resonance but also reduces the high-frequency attenuation of the isolation system. Consequently, it is necessary to lower the cut-off frequency of the passive isolator to meet isolation requirements. This is traditionally performed by using soft mountings to isolate the whole card or carter containing the component. Simultaneously, this results in large low-frequency relative displacement and particular attention must be paid to assess risks of connections failure or contacts between the support and the isolated element.

Among the existing solutions to provide vibration isolation, active suspensions based on the skyhook system appear really attractive because of their achievable performance. As illustrated in grey in Figure 1, the transmissibility of such suspensions can be tuned to reach ideal vibration isolation goals [8,9]:

1. the transmission of low-frequency support acceleration;
2. the critical damping of the suspension eigenmode;
3. the preservation of the attenuation in the high frequency domain.

Successful applications of various skyhook isolators have been broadly proposed in the literature, featuring their isolation efficiency and robustness properties [9]. But such strategies are still applied to macro-scale suspensions and control energy cost is currently the main restriction that limits their implementation. On the one hand, transducer integration is complex and remains a topic dedicated to smart material specialists and, on the other hand, final users are not ready to pay the control energy cost related to the amount of mass to isolate. Some integrated electromechanical systems can be used such as integral resonant systems [10,11], but by reducing the frequency band of efficiency of the proposed solution.
To deal with all the above-mentioned problems, the micro active suspension (MAS) concept has been proposed and realised [4,12]. It consists in implementing skyhook isolation strategies directly between sensitive equipment and its vibrating support (electronic card). MEMS technology is a serious asset for the application of the concept since small suspensions can be manufactured in clean rooms from silicon on insulator (SOI) wafers and a large variety of coupling materials or mechanisms (piezoelectricity, electromechanic/electrostatic coupling) is available to perform transducer (sensor/actuator) integration. Miniaturisation also brings several benefits for the application of active isolation strategies:

1. The amount of mass to isolate is drastically reduced so that required control energy can be provided by surrounding electrical energy sources.
2. Isolation performances of skyhook isolators are limited by natural damping. In micromechanical structures natural damping is lower than in macro-scale applications that provide better isolation performances.

In previous studies, difficulties were encountered in obtaining a reliable and integrated force sensor for applying the classical integral force feedback strategy. In [2], a piezoelectric thin film located under the isolated component was used in its compression mode. This sensor appeared much more sensitive to platform deformation than to compression and could not be used to perform isolation strategy. In [13], design of two stages suspensions was studied in order to use piezoelectric film in transverse mode. For this kind of architecture, suspension and sensor dynamic are closed in the frequency domain. Natural damping induced by MEMS structures does not provide sufficient stability margin to avoid sensor instability in the closed loop configuration. This phenomenon, known as spillover, has to be dealt with by optimising transducer design or the strategy algorithm [14,15]. The solution for this specific isolation system is to add a second-order phase delay compensator into the controller to restore collocation property and avoid instability of the closed loop system, as proposed in [8]. This approach can appear complex to realize when a two-stage suspension is considered with internal residual modes that have to be decoupled [16].

Based on this short bibliography, it appears that significant difficulties have to be dealt with when attempting to implement classical skyhook isolation. The main one consists of the necessity to design a fully embedded force sensor without introducing residual dynamical behaviour into the suspension system. To solve this problem, a new MAS system has been designed, realised and characterised without using any internal force sensor for estimating the absolute velocity of the isolated device. This paper also presents the corresponding original MEMS prototype designed, fabricated and tested to demonstrate the feasibility of the MAS concept based on the use of a new active isolation strategy. Section 2 presents the control strategy used and its intrinsic performance and robustness properties. The technological implementation is exposed in Section 3, while its experimental characterisation is addressed in Section 4. Finally, Section 5 outlines conclusions and perspectives of this paper.

2. Integral estimated-force feedback (IeFF) isolation strategy

2.1. Active control strategy

The integral estimated-force feedback (IeFF) strategy is described in detail [17]. This control methodology allows implementation into a standard suspension of a skyhook damping, as described in Figure 2, to obtain optimal isolator behaviour [8]. It is based on the use of
an active electromechanical suspension system driven by a sensing signal proportional to
the internal relative displacement and works without information on the transmitted force
or the absolute acceleration of the isolated mass. If one considers a theoretical linear sus-
ension with negligible internal damping, the dynamic equilibrium of the isolated element
and the relative displacement measurement ($\Delta w$) are given by:

$$m \ddot{w}_{\text{p}} = -k \Delta w + e_A u,$$  (1)

$$y = e_S \Delta w,$$  (2)

where $w_{\text{p}}$ is the absolute displacement of the isolated element, $m$ its mass, $k$ is the sus-
pension stiffness; $e_A$ and $e_S$ are the coupling characteristics of the actuator and sensor used.

Let us assume $f_{\text{p}}$ is the transmitted force given by $f_{\text{p}} = m \dot{w}_{\text{p}}$. In the case of skyhook integral
force feedback (IFF) implementation the control force is calculated by the following
relation:

$$e_A u(t) = -G \int_{t_0}^{t} f_{\text{p}}(\tau) d\tau = -G m \dot{w}_{\text{p}}(t),$$  (3)

where $Gm$ stands for the prescribed suspension damping, which is linearly dependent on the
feedback gain $G$. As we try to avoid any measurement of the transmitted force, one can esti-
mate this quantity only by using a relative displacement sensor. By using Equations (1)–(2),
it easy to show that, knowing the imposed control command $u$ and the relative dis-
placement $\Delta w$, one can obtain the transmitted force $f_{\text{p}}$. The proposed estimator is also
given by:

$$\tilde{f}_{\text{p}} = e_A (-Ay + u),$$  (4)

where $A$ stands for an internal parameter closest as possible to its theoretical value
$A_{\text{th}} = k/(e_A e_S)$. Thus, the IeFF control strategy corresponds to the feedback equation:

$$u(t) = -G \int_{t_0}^{t} [-Ay(\tau) + u(\tau)] d\tau.$$  (5)
The control feedback transfer is given by:

\[ \frac{u}{y} = A \frac{G}{j\omega + G} \]

where
\[
A = a \times \frac{k}{(e_A e_S)} \\
G = 2\xi_{sky}\omega_0 = g\omega_0.
\]

This corresponds to a simple integration with a static gain given by estimator parameter \(a\) such as \(A = aA_{th}\). When the theoretical perfect estimation is obtained (e.g. \(a = 1\)), the IeFF strategy leads to implement a skyhook active isolator with an internal induced damping term \(\xi_{sky}\) given by the ratio between the integrator cut-off frequency \(G\) and the suspension natural frequency \(\omega_0 = (k/m)^{1/2}\).

### 2.2. Strategy performance and robustness analysis

The theoretical estimation is unfortunately not implementable in a realistic system due to inherent uncertainties in the electromechanical coefficients of the active suspension. This leads us to use the reduced estimator quality coefficient \(a\) as the central parameter for studying the control properties in term of performance but also for estimating robustness characteristics. The different useful transfer functions defining the system efficiency and reliability margins are obtained from the modelling of the whole active suspension into an inertial coordinate system linked to the isolator basis mass.

We also define \(T(\Omega)\) the suspension transmissibility, \(D(\Omega)\) the transfer between the internal stiffness force and the external disturbance apply to the basis mass directly linked to the suspension reliability and \(C(\Omega)\) the relation between the control force and the same external disturbance, also evaluating the control cost. These transfer functions are given by the following equations, where \(\Omega\) stands for the reduced frequency \(\omega/\omega_0\):

\[
T(\Omega) = \frac{m\ddot{w}_P(\Omega)}{m\ddot{w}_B(\Omega)} = \frac{j\Omega + g (1 - a)}{j\Omega (1 - \Omega^2) + g (1 - a - \Omega^2)},
\]

\[
D(\Omega) = -\frac{k \Delta w(\Omega)}{m\ddot{w}_B(\Omega)} = \frac{j\Omega + g}{j\Omega (1 - \Omega^2) + g (1 - a - \Omega^2)},
\]

\[
C(\Omega) = \frac{e_A u(\Omega)}{m\ddot{w}_B(\Omega)} = -\frac{ga}{j\Omega (1 - \Omega^2) + g (1 - a - \Omega^2)}.
\]

They are presented in Figure 3 for different values of parameter \(a\) for a control objective \(\xi_{sky} = 60\%\) defined in Equation (6).

The Routh–Hurwitz criterion allows determination of the stability margin of the proposed strategy in parametric space. We obtain a stable system if and only if:

\[
0 < g \quad \text{and} \quad 0 < a < 1.
\]

So a perfect estimation of the transmitted force is obtained when \(a = 1\) is on border but outside the stable domain. The IeFF strategy can also converge toward the skyhook IFF control performance but cannot realize it. The transfer functions presented in Figure 3 show that the proposed IeFF tends toward the skyhook IFF by insuring a strong attenuation of the modal resonance with a comparable high-frequency cut-off characteristic. The static transmissibility is also not affected by the strategy that guarantees mechanical stability of
the supported system. Nevertheless, the high isolation performance induced by increasing the estimation quality \(a\) leads to a large increase of the control force \(C\) and of the internal displacement \(D\). When \(a = 1\), these transfers tend toward infinite values that corresponds to unstable behaviour. The optimal design is a compromise between the system transmissibility \(T\), the cost \(C\) and the system reliability \(D\). Different approaches can be used for defining an optimal point in the parametric space (e.g. \(a, g\)). In [17] a complete study shows that the damping objective can be chosen to be around \(\xi_{\text{sky}} = 60\%\) (e.g. \(g = G/\omega_0 = 1.2\)) and the estimator quality \(a = 2/3 = 66\%\).

The performance analysis has to be completed with robustness characterisation. The robustness in stability is driven by the estimator quality uncertainty \(\Delta a\) such as \(|\Delta a| < 1 - a\). For the previously defined nominal point \((a = 2/3)\) we can accept a relative uncertainty \(|\Delta a|/a < 50\%\) without risk of instability. The robustness in performance can be evaluated by a sensitivity analysis of the transmissibility function \(T\) in front of variations of parameters \(a\) and \(g\) such as:

\[
|T|_\infty (g + \Delta g, a + \Delta a) \approx |T|_\infty (g, a) + \left[ \frac{\partial |T|_\infty}{\partial g} \Delta g + \frac{\partial |T|_\infty}{\partial a} \Delta a \right] (g, a), \tag{11}
\]

where the Jacobian matrix can be normalised to use relative uncertainties as:

\[
\frac{\Delta |T|_\infty (g, a)}{|T|_\infty (g, a)} \approx S_g (g, a) \frac{\Delta g}{g} + S_a (g, a) \frac{\Delta a}{a}, \tag{12}
\]

The sensitivity functions \(S_g, S_a\) are plotted in Figure 4. On surface \(S_g(g,a)\) the points \((g^*, a)\) insensitive to the damping objective variation \(\Delta g/g\) are also plotted in dashed line. This curve also corresponds to specific parametric couples for which the strategy performance is insensitive to mass variation of the isolated element \(\Delta m/m\). Excepted for the line \(a = 0\), no domain exists that is insensitive to parameter \(a\). The magnitude of the \(S_g(g,a)\) function increases with \(a\) and \(g\). It is also impossible to prevent performance variation from amplifier gain \(\Delta A/A\), or from transducer coupling coefficients \(\Delta e_A/e_A\) and \(\Delta e_S/e_S\). As indicated in Figure 5, any internal stiffness variation induces performance variation. A way to decrease its sensitivity is to use a moderate nominal parameter for ruling the control strategy.

The strategy analysis shows that nominal parameters such \(\xi_{\text{sky}} = 60\%\) (e.g. \(G/\omega_0 = 1.20\)) and \(a = 66\%\) lead to important performance of the active isolator \((|T|_\infty = 2)\) while
Figure 4. Normalised parametric sensitivity of the $H_\infty$ norm of the transmissibility function $|T|_\infty$ in the stable domain $0 < g$ and $0 < a < 1$. (a) Sensitivity to parameter $g$, $S_g(g,a)$. (b) Sensitivity to parameter $a$, $S_a(g,a)$

Figure 5. Normalised parametric sensitivity of the performance criteria $|T|_\infty$ toward suspension stiffness variation: $S_k(g,a) = -1/2 \times S_g(g,a) - S_a(g,a)$.

limiting the internal relative displacement ($|D|_\infty \approx 3.5$) and the control command level ($|C|_\infty \approx 2.1$). This point also allows moderate performance sensitivity and guarantees robust stability with a margin of 50% on parameter $a$. These parameters will be considered as the nominal objective for the system implementation.
3. Micro active suspension design and fabrication

3.1. Design characteristics

The proposed strategy architecture, IeFF, has been applied to the design of a micro active suspension (MAS) based on a piezoelectric MEMS transducer working along one degree of freedom (Figure 2). The central platform dedicated to the support of the vibration-sensitive device is connected to the external support structures by a network of silicon-based composite beams incorporating a piezoelectric thin layer (Figure 6). The overall dimension is \(1 \times 1 \text{ cm}^2\). The whole system can also be fixed onto a shaker table for testing controlled behaviour in front of externally imposed acceleration and validate the active isolation functionality.

In order to obtain dynamical behaviour limiting residual modal coupling effects on the first fundamental suspension natural frequency, a strong contrast in thickness for external frame, beams and the central platform has been used. The realised system can also localise strain energy into supporting beams in a large frequency band. The central platform can also be considered as a rigid element and be moved from the external frame along three degrees of freedom: one translation considered as the active degree of freedom and two out-of-plane rotations. These modes are described in Figure 7. Some particular choices in the design parameters allowed the system to present a large frequency gap between the first natural suspension mode and the residual rotation eigensolutions.

The integrated transducers are made of a thin lead zirconate titanate (PZT) layer obtained by sol–gel deposition on the superior face of the micro system. The transverse piezoelectric effect is obtained by using electrode deposition on inferior and superior faces of the PZT layer to give a transversal piezoelectric polarity. Their locations and sizes have been determined by using a simple finite element model. As shown in Figure 8, the piezoelectric system has also been designed for maximising the piezoelectric coupling effect [18,19] when unitary suspension modal displacement has been imposed. This optimal point corresponds also to the optimal actuation on the targeted mode. The final silicon-based piezocomposite beam can be described using the parameters shown in Figure 9. One electrode pair is also used for picking up the sensing current \(q_S\) or the voltage \(\Delta v_S\) and the other for the actuating command imposed by imposing current \(q_A\) or the voltage \(\Delta v_A\), as described in Figure 10. The complete set of parameter values describing the realised prototype are given in Table 1.

Figure 6. Drawing of the micro active suspension prototype using SolidWorks CAD software.
Figure 7. Strain energy density on ‘suspension modes’ predicted by FEM software COMSOL. The mechanical parameters chosen induce concentration of the strain energy inside suspension beams. (a) Mode 1 ($f_1 = 5372$ Hz). (b) Mode 2 ($f_2 = 16,072$ Hz). (c) Mode 3 ($f_3 = 16,077$ Hz).
Figure 8. Charge density induced by direct piezoelectric coupling on suspension modes of the silicon substrate coated with a PZT layer. The PZT film is 2.15 μm thick and equipped with a pair of grounded electrodes (zero electric field configuration). (a) Mode 1 \( (f_1 = 5371 \text{ Hz}) \). (b) Mode 2 \( (f_2 = 15758 \text{ Hz}) \). (c) Mode 3 \( (f_3 = 15764 \text{ Hz}) \).
3.2. Device fabrication process

The device manufacturing process was carried out at the Center of MicroNanoTechnology and in the Ceramics Laboratory of the Swiss Federal Institute of Technology at Lausanne (EPFL). The process starts on a SOI substrate whose silicon-device, silicon-handle and buried silicon dioxide layers are, respectively, 50 μm, 380 μm and 2 μm thick. Silicon beams are realised in the silicon-device layer by successive front side deep reactive ion etching of the silicon-device and buried silicon dioxide layers. Release of suspension beams and introduction of thickness contrast (for central platform and external frame) are performed by a back side deep dry etching of the silicon-handle layer. The process has a high Si:SiO2 selectivity and is stopped by the remaining buried silicon dioxide to obtain suspension beams with a uniform silicon thickness.

A complete description of the fabrication processes is provided in [17]. Concerning the PZT sol–gel deposition, the entire process has been developed by the Ceramics Laboratory of EFPL [21,22]. It uses four solutions with specific precursor concentrations to obtain a specific gradient, as shown in Table 2. After rapid thermal annealing, the final composition of the obtained PZT layer tends to homogeneous median targeted morphotropic phase boundary composition [22,23]. The different steps of the procedure are described
Table 1. Dimensions, material properties and mechanical parameters of the piezoelectric MAS.

| Geometric dimensions (see Figure 9) |  |
|-------------------------------------|---|
| Beam length                         | L = 3200 μm |
| Beam width                          | l_S = 1200 μm |
| Beam thickness                      | h_S = 50 μm |
| Platform size                       | p = 1600 μm |
| Platform thickness                  | h_R = 200 μm |
| Electrodes width                    | l_P = 880 μm |
| Gap inter-electrodes                | d = 100 μm |
| PZT layer thickness                 | h_P = 2 μm |

| Material properties                  |  |
|-------------------------------------|---|
| Silicon Young modulus [20]          | E_Si = 131 GPa |
| Silicon mass density                | ρ_Si = 2330 kg.m⁻³ |
| PZT transverse coupling coefficient | e_{f31} = -10 C m⁻² or N m⁻¹ V⁻¹ |
| PZT permittivity                    | ε_{f33} = 1200 ε_0 (ε_0 = 8.85 × 10⁻¹² F/m) |

| Mechanical parameters of the simplified lumped mass model (see Figure 10) |  |
|-----------------------------|---|
| Static bending stiffness     | k ≈ 2.4 mN/μm |
| PZT transducer capacitance   | (C_S, C_A) ≈ 27.6 nF |
| PZT transducer transverse coupling | (e_S, e_A) ≈ 0.4 mN/V |
| Beam mass                    | μ ≈ 0.5 mg |
| Platform mass                 | m_P ≈ 1.5 mg |
| Frame mass                    | m_B ≈ 73.3 mg |
| Total mass                    | m_{tot} = m_B + m_P + 4μ ≈ 76.8 mg |
| Suspension frequency         | ω_0 = \sqrt{\frac{k}{m_P + (1/5)4μ}} ≈ 2π × 5371 rad/s |

Table 2. Composition of the solutions used for the gradient-free sol–gel process.

| Solution | Composition (Zr % : Ti %) |
|----------|---------------------------|
| S1       | 63% : 37% |
| S2       | 58% : 42% |
| S3       | 48% : 52% |
| S4       | 43% : 57% |
| S5       | 53% : 47% (MPB) |

in Figure 11. The final 2.15 μm thick PZT layer is obtained after seven interactions. Characterisation of the obtained thin film can be found in [23].

4. Experimental characterisation of the micro active suspension

The finally obtained micro active suspension (Figure 12) was embedded onto a shaker table to allow application of imposed external acceleration (\ddot{w}_B). The targeted suspension mode at 2514 Hz can be measured and controlled by connecting suitable electronic circuits onto the realised transducer (Figure 13). The piezoelectric layer between the central-cross electrodes (V1 and V2) is used as a displacement sensor. Its electrodes are connected to the electronic measurement interface constituted of a voltage amplifier stage with high input impedance. Amplifier output voltage (y) is thus proportional to the voltage across resistance $R_S$, itself proportional to the internal relative displacement of the suspension.
Figure 11. Steps of the gradient-free sol–gel process [22].

Figure 12. Close up of the microsystem and of the vibrating table used to prescribe frame acceleration.

The command interface allows driving safely the PZT film sandwiched between external electrodes (V3 and V4). The resistance $R_A$ connected in parallel with the capacitive load allows the dissipation of static charges and prevents the actuator from depolarising. The resistance $R_L$ limits high frequency input current to protect the MEMS connectors.

The IeFF control law is synthesised into the electronic circuit by the implementation of an analogical circuit made of three subsets (Figure 13). The voltage non-inverting amplifier ($R_3, R_4$) applied on the output of the feedback allows adjustment of the loop gain associated to the IeFF parameter $a$. This amplifies the integrator ($R_G, C_G$) output signal of which the cut-off frequency is ruled to adjust the targeted damping value corresponding to the IeFF
Figure 13. Circuit diagram of the closed loop system composed of (from top to bottom), the electromechanical suspension, the sensor ($y$) and actuator ($u$) interfaces and the IeFF controller.

Figure 14. Interfaced microsystem experimental frequency response to a white noise voltage imposed on the vibrating table. The acceleration level prescribed by the table ($\ddot{w}_B$) is given by the black dashed line (in g/V). Resulting suspended platform acceleration ($\ddot{w}_P$) and sensor interface output signal ($y$) levels are plotted in black (in g/V) and grey (in V/V), respectively.

g parameter. An additional high pass filter ($R_F, C_F$) has been added to prevent amplifier static saturation.

Figure 14 shows the frequency response functions of the frame acceleration (dashed line), the platform acceleration obtained by post-processing the velocity signal measured by a Doppler laser velocimeter (black plain line) and the signal $y$ picked up through our electronic circuit measurement interface (grey plain line). By comparing the frame and the platform accelerations, we observe the suspension pumping mode at 2514 Hz. After this cut-off frequency the isolator transmissibility decreases and we note a strong attenuation in the transmitted acceleration. The internal measured system $y$ is totally coherent and proportional to the absolute acceleration of the central platform as indicated in Equations (1)–(2) when no control force $u$ is applied on the system. The measured sensibility of the internal sensor is also 13 mV/g (where g is $9.81 \text{ m s}^{-2}$).

Figure 15 shows up the frequency response functions (FRFs) of the measured signal $y$ in grey and of the platform displacement (obtained by post processing absolute velocity
measurement) in black, when a control signal is applied to the electronic circuit control interface through resistances $R_A$ and $R_L$ (fixed at 680 $\Omega$). Both plotted FRFs are identical in the low-frequency band, which demonstrates that our integrated transducers only control and observe symmetric pumping mode in a large frequency band. It does not appear that any residual electrical coupling effects exist that could greatly affect the control performance, as shown in [3,13]. The internal sensor also picks up a signal proportional to the relative displacement of the central platform from the supporting frame. With a resistance ratio $R_2 / R_1 = 100$, the sensor sensibility is 1 V/μm. The observed differences in the high-frequency band are due to high-frequency beam modes inducing a zero in the transducers collocated transfer function. The implemented control has to take in charge the induced phase delay for avoiding spill-over effect. The roll-off characteristic of the implemented control circuit (see Figure 16) limits this potential problem.

The IeFF circuit presented is governed by using fixed capacitors such as $C_F = C_G = 100$ nF and a high filter resistance $R_F = 82$ k$\Omega$. The transfer function of this circuit,

$$
\frac{u}{y} = \left(1 + \frac{R_4}{R_3}\right) \frac{1}{\frac{2}{j\omega + 2\tau_G j\omega} + 1/(2\tau_F)} \quad \text{with} \quad \begin{cases} 
\tau_G = R_G C_G \\
\tau_F = R_F C_F 
\end{cases},
$$

(13)

Figure 16. Experimental transfer function of the electronic IeFF controller ($u/y$) for $R_4 = 0$ $\Omega$. The targeted skyhook damping ratio $\xi_{sky}$ is tuned by the resistance value ($R_G$) according to the relationship $2\xi_{sky}\omega_0 = (R_G C_G)^{-1}$. 
also corresponds to a high-pass filter with an cut-off frequency at 9.7 Hz in series with an 
integrator element whose cut-off frequency is ruled by adjusting the resistance $R_G$ directly 
linked to the targeted suspension damping. Figure 16 presents different measured transfer 
functions with $R_4 = 0 \, \Omega$ for $R_G$ values from 2 k$\Omega$ to 680 $\Omega$.

The damping objectives $\xi_{\text{sky}}$ from 35% to 100% are obtained for resistance values $R_G = 1/(2\xi_{\text{sky}}\omega_0 C_G)$ from 2 k$\Omega$ to 680 $\Omega$. The chosen value for testing the control system is $R_G = 1.2$ k$\Omega$ in order to impose a damping ratio of $\xi_{\text{sky}} = 58.9\%$ close to the optimal one defined by the performance and robustness characterisation of the control strategy. The loop gain, ruled by fixing resistance ratio $R_3 / R_4$, is directly related to the estimator quality coefficient ratio $a$. As shown in Figure 17, this gain can be modified to adjust this term from an initial value $a_0 = 10\%$ when $R_4 = 0 \, \Omega$. As shown by Equation (10), the open loop static gain should be less than 1 V/V corresponding to a ratio $R_3 / R_4 < 8$. By fixing $R_3 = 1.2$ k$\Omega$ and $R_4$ from 0 to 10 k$\Omega$ one can impose static open loop gain without reaching the unit corresponding to the theoretical instability limit of the used IeFF strategy. Figure 17 presents a set of obtained open loop transfer functions for different $R_4$ values and $R_G = 1.2$ k$\Omega$.

The experimental implementation of the micro active suspension is performed by connecting the control electronic circuit to the system’s transducers. The transfer function between the central platform acceleration ($\ddot{w}_P$) and the imposed frame one ($\ddot{w}_B$) is plotted in Figure 18 for different values of the loop gain $R_4$. The measured transmissibility corresponds to those of a skyhook controlled suspension with a strong attenuation of the modal overshot and unmodified static stability and decay rate of about $-40$ dB/decade after the cut-off frequency. The transfer function between the frame applied acceleration and the command signal $u$ is given in Figure 19. Once again, the adaptive suspension behaviour totally corresponds to that predicted theoretically (Figure 3).

5. Conclusions

The major contribution of this paper concerns the development and implementation of 
an original active isolation strategy allowing integration of a micro active skyhook isolator by using a single relative displacement measurement. The name integral-estimated force feedback (IeFF) adopted to describe this strategy underlines its connection with the classical integral force feedback and the use of a specific transmitted force estimator. The performance and robustness properties of the proposed IeFF strategy have also been studied to provide all the design parameters for designing and testing a technological

![Figure 17](image-url)  
Figure 17. Experimental open loop transfer function for $R_G = 1.2$ k$\Omega$. Static gain of this transfer function corresponds to estimation quality (e.g. parameter $a$) that can be adjusted with $R_4$ value.
The established design rules allowed micro-system processing in the Centre of MicroNanoTechnology and the Ceramic Laboratory of Ecole Polytechnic Federal of Lausanne. The obtained micro active suspension (MAS) has been completely characterised and the control performance experimentally evaluated. We underline that the MAS system exhibits measured transmissibility corresponding to those of a skyhook controlled suspension with a strong attenuation of the modal overshot and unmodified static stability and decay rate of about –40 dB/decade after the cut-off frequency.

This work represents a breakthrough towards electromechanical integration. The simplicity of the IeFF strategy is particularly suitable for using piezocomposite structures and greatly reduces the complexity of the active isolator electronics. The number of electronic components required for signal processing and the command synthesis is limited. It could be possible to reduce it further by exploiting the dielectric properties of the actuator to perform direct integration in transducers itself. Under these conditions, the control loop could be implemented with a single amplifier stage and only five resistors. The main task now is to work on hybridisation of sensitive components on the central platform and to give design indicators for reliability-based robust optimisation when embedded mass is considered. This last step before effective industrial development can be addressed based on modelling tools carried out by this work. Other perspectives can concern the IeFF distributed integration in a MEMS device for the synthesis of a multi degrees of freedom skyhook isolator. The IeFF decentralised strategy can be applied independently to different
pairs of transducers to introduce different and independent micro active suspensions for controlling all transmission modes of the device.

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