An active vibration control system as a benchmark for adaptive regulation (rejection of unknown-time varying narrow band disturbances)

Ioan Doré Landau * Marouane Alma * John Jairo Martinez *
Gabriel Buche * Alireza Karimi **

Abstract: The adaptive regulation is an important issue with a lot of potential for applications in active vibration control and active noise control. The basic problem from the "control system" point of view is the rejection of multiple unknown and time varying narrow band disturbances without using an additional transducer for getting informations upon the disturbances. An adaptive feedback approach has to be considered for this problem. Industry needs a "state of the art" in the field based on a solid experimental verification on a system using a current used technology. The paper present a benchmark problem for suppression of multiple unknown and/or time-varying vibrations and an associated active vibration control system using an inertia actuator on which the experimental verifications will be done. The objective is to minimize the residual force by applying an appropriate control effort through the inertial actuator. The system does not use any additional transducer for getting in real time information upon the disturbances.

The final version of the paper will contain a comparative evaluation on the real system of the various approaches proposed.

1. INTRODUCTION

One of the basic problems in active vibration control is the attenuation of multiple vibrations of unknown and time varying frequencies using only the measurement of the residual force (or acceleration).

From a hardware point of view one of the solutions is to use an active suspension system. See for example (Landau et al. [2005]). However the technology have evolved towards the use of inertia (electro-dynamic) actuators (Marcos [2000]). Using this technology, the system can be composed on co-located and non co-located passive dampers, inertia actuators and force/acceleration measurements leading to a better efficiency. In this paper we will consider an active vibration control system using an inertia actuator in a co-located configuration.

Vibrations correspond to disturbances with energy concentrated in a narrow band around unknown and/or time-varying frequencies. From a "signal and system" perspective one can view these disturbances as a white noise or a Dirac impulse passed through a "model of the disturbance". To be more specific, the disturbances considered can be defined as "finite band disturbances". This includes single or multiple narrow band disturbances or sinusoidal disturbances. While in general one can assume a certain structure for such "model of disturbance", its parameters are unknown and may be time varying. From a control point of view the objective is the attenuation (rejection) of unknown disturbances without measuring them. Since the model of the disturbances is unknown and time varying, this will require to use an adaptive approach. We have to solve an "adaptive regulation" problem since the objective is the attenuation of unknown disturbances without measuring them. While the disturbances are narrow band disturbances of unknown and time varying frequency (within a certain frequency region), the dynamic characteristics of the active vibration system itself for a given physical realization are practically constant. The corresponding "control model" to be used for control design (and tuning) can be identified from input/output data. Furthermore for robustness reasons the disturbances should be located in the frequency domain within the regions where the plant (the "active part") has enough gain.

Solutions for this problem, provided that an "image" of the disturbance can be obtained by using an additional transducer, have been proposed by the signal processing community and a number of applications are reported (Elliott and Nelson [1994], Elliott and Sutton [1996], Beranek and Ver [1992], Fuller et al. [1995]). However, these solutions (inspired by Widrow’s technique for adaptive noise cancellation (Widrow and Stearns [1985])) ignore the possibilities offered by feedback control systems and require an additional transducer. The principle of this signal processing solution for adaptive rejection of unknown disturbances is that a transducer can provide a measurement, highly correlated with the unknown disturbance. This information is applied to the control input of the plant through an adaptive filter (in general a Finite Impulse Response - FIR)

1 With respect to classical “adaptive control” paradigm there is a fundamental difference. In the case of "adaptive regulation" the model of the plant is assumed known and almost time invariant and the parameters of the model of disturbances are unknown and possible time varying while in classical "adaptive control" the model of the plant is unknown and possible time varying and the model of disturbances is supposed to be known.
whose parameters are adapted such that the effect of the disturbance upon the output is minimized. The disadvantages of this approach are:

- It requires the use of an additional transducer.
- Difficult choice for the location of this transducer (it is probably the main disadvantage).
- It requires the adaptation of many parameters.

To achieve the rejection of the disturbance (at least asymptotically) without measuring it, an adaptive feedback solution has to be considered. This approach does not require the use of an additional measurement.

The industry needs a "state of the art" in the field based on a solid experimental verification on a benchmark using an inertial actuator.

The scientific objective of the benchmark is to evaluate current available procedures for adaptive regulation which may be applied in the fields of active vibration control (and possible in active noise control). The benchmark specifically will focus in testing: 1) performances, 2) robustness and 3) complexity.

The test bed is an active suspension using an inertial actuator and equipped with a shaker and a measure of the residual force. It is located at GIPSA-Lab, Grenoble (France) which has already experience on organizing benchmarks on test beds (see European Journal of Control, no.2, 1995 and no.1, 2003).

2. AN ACTIVE VIBRATION CONTROL SYSTEM USING AN INERTIAL ACTUATOR

The structure of the system used used for the benchmark is presented in figure 1. A general view of the whole system including the testing equipment is shown figure 2. It consists on a passive damper, an inertial actuator, a load, a transducer for the residual force, a controller, a power amplifier and a shaker. The mechanical construction of the load is such that the vibration produced by the shaker, fixed to the ground, are transmitted to the upper side, on top of the passive damper. The inertial actuator will create vibrational forces which can counteract the effect of vibrational disturbances (inertial actuators uses a similar principle as loudspeakers). It is fixed to the chassis where the vibrations should be attenuated. The controller, through the power amplifier, will generate current in the mobile coil which will produce a movement in order to reduce the residual force. The equivalent control scheme is shown in figure 3. The system input, \( u(t) \) is the position of the mobile part (magnet) (see figures 1, 3), the output \( y(t) \) is the residual force measured by a force sensor. The transfer function \( \frac{q^{-d}C}{D} \), between the disturbance force, \( u_p \), and the residual force \( y(t) \) is called primary path. In our case (for testing purposes), the primary force is generated by a shaker driven by a signal delivered by the computer. The plant transfer function \( \frac{q^{-d}B}{A} \) between the input of the inertial actuator, \( u(t) \), and the residual force is called secondary path. The input of the system being a position and the output a force, the secondary path transfer function has a double differentiator behavior. The system itself in the absence of the disturbances will feature a number of low damped vibration modes as well as low damped complex zeros (anti-resonance).

The control objective is to reject the effect of unknown narrow band disturbances on the output of the system (residual force), i.e. to attenuate the vibrations transmitted from the machine to the chassis. The physical parameters of the system are not available. The system has to be considered as a "black box" and the corresponding models for control design should be identified. The sampling frequency is 800 Hz.

3. PLANT/DISTURBANCE REPRESENTATION AND CONTROLLER STRUCTURE

The structure of the linear time invariant discrete time model of the plant -the secondary path- used for controller design is:

\[
G(z^{-1}) = \frac{z^{-d}B(z^{-1})}{A(z^{-1})} = \frac{z^{-d-1}B'(z^{-1})}{A(z^{-1})},
\]  

(1)
with:
\[ d = \text{the plant pure time delay in number of sampling periods} \]
\[ A = 1 + a_1 z^{-1} + \cdots + a_n z^{-n_a}; \]
\[ B = b_1 z^{-1} + \cdots + b_n z^{-n_b} = q^{-1} B' ; \]
\[ B' = b_1 + \cdots + b_n z^{-n_b+1}, \]
where \( A(z^{-1}), B(z^{-1}), B'(z^{-1}) \) are polynomials in the complex variable \( z^{-1} \) and \( n_A, n_B \) and \( n_B - 1 \) represent their orders.\(^2\) The model of the plant may be obtained by system identification. Details on system identification of the models considered in this paper can be found in Landau and Zito [2005], Constantin [2001], Landau et al. [2001b,a], A.Karimi [2002], Constantin and Landau [2003].

Since the benchmark is focused on regulation, the controller to be designed is a RS-type polynomial controller (or an equivalently state space controller + observer) (Landau et al. [1997], Landau and Zito [2005]) - see also figure 3.

The output of the plant \( y(t) \) and the input \( u(t) \) may be written as:
\[ y(t) = \frac{q^{-d} B(q^{-1})}{A(q^{-1})} u(t) + p(t); \]
\[ S(q^{-1}) \cdot u(t) = -R(q^{-1}) \cdot y(t), \]
where \( q^{-1} \) is the delay (shift) operator \( s(t) = q^{-1} x(t+1) \) and \( p(t) \) is the resulting additive disturbance on the output of the system. \( R(z^{-1}) \) and \( S(z^{-1}) \) are polynomials in \( z^{-1} \) having the orders \( n_B \) and \( n_S \), respectively, with the following expressions:
\[ R(z^{-1}) = r_0 + r_1 z^{-1} + \cdots + r_{n_B} z^{-n_B} = R(z^{-1}) \cdot H_R(z^{-1}) ; \]
\[ S(z^{-1}) = 1 + s_1 z^{-1} + \cdots + s_{n_S} z^{-n_S} = S(z^{-1}) \cdot H_S(z^{-1}) , \]
where \( H_R \) and \( H_S \) are pre-specified parts of the controller (used for example to incorporate the internal model of a disturbance or to open the loop at certain frequencies).

We define the following sensitivity functions:

- **Output sensitivity function (the transfer function between the disturbance \( p(t) \) and the output of the system \( y(t) \))**:
\[ S_{yp}(z^{-1}) = \frac{A(z^{-1}) S(z^{-1})}{P(z^{-1})} , \]

- **Input sensitivity function (the transfer function between the disturbance \( p(t) \) and the input of the system \( u(t) \))**:
\[ S_{up}(z^{-1}) = -\frac{A(z^{-1}) R(z^{-1})}{P(z^{-1})} , \]

where
\[ P(z^{-1}) = A(z^{-1}) S(z^{-1}) + z^{-d} B(z^{-1}) R(z^{-1}) \]
\[ = A(z^{-1}) S(z^{-1}) \cdot H_R(z^{-1}) + z^{-d} B(z^{-1}) R(z^{-1}) \cdot H_S(z^{-1}) \]
(8)
defines the poles of the closed loop (roots of \( P(z^{-1}) \)).

In pole placement design, the polynomial \( P(z^{-1}) \) specifies the

2 The complex variable \( z^{-1} \) will be used for characterizing the system’s behavior in the frequency domain and the delay operator \( q^{-1} \) will be used for describing the system’s behavior in the time domain.

Fig. 4. The block diagram of the data acquisition system.

desired closed loop poles and the controller polynomials \( R(z^{-1}) \) and \( S(z^{-1}) \) are minimal degree solutions of (8) where the degrees of \( P, R \) and \( S \) are given by: \( n_P = n_A + n_B + d - 1, \)
\( n_S = n_B + d - 1 \) and \( n_R = n_A - 1 \).

Using equations (2) and (3), one can write the output of the system as:
\[ y(t) = \frac{A(q^{-1}) S(q^{-1})}{P(q^{-1})} \cdot p(t) = S_{yp}(q^{-1}) \cdot p(t). \]

For more details on RS-type controllers and sensitivity functions see Landau and Zito [2005].

Suppose that \( p(t) \) is a deterministic disturbance, so it can be written as
\[ p(t) = \frac{N_p(q^{-1})}{D_p(q^{-1})} \cdot \delta(t), \]
where \( \delta(t) \) is a Dirac impulse and \( N_p(z^{-1}), D_p(z^{-1}) \) are co-prime polynomials in \( z^{-1} \), of degrees \( n_{N_p} \) and \( n_{D_p} \), respectively.

In the case of stationary disturbances the roots of \( D_p(z^{-1}) \) are on the unit circle (which will be the case for the disturbances considered in the benchmark). The energy of the disturbance is essentially represented by \( D_p \). The contribution of the terms of \( N_p \) is weak compared to the effect of \( D_p \), so one can neglect the effect of \( N_p \).

4. DATA ACQUISITION AND SYSTEM IDENTIFICATION

The block diagram of the data acquisition system is shown in figure 4. The data provided have been obtained in open loop operation. (to be downloaded from: www.lag.ensieg.inpg.fr/jairo/benchmark)

The excitation signals were PRBS (pseudo random binary sequence) of various lengths.

Figures 5 and 6 give the frequency characteristics of the primary and secondary path respectively for non parametric models obtained by spectral analysis.

5. SIMULATOR

A “black box” discrete time simulator of the active suspension built on matlab simulink (Matlab 2007) has been provided (can be downloaded from the benchmark website) The main simulation scheme is shown in figure 7.
The primary path and the secondary path are indicated. A generator for the PRBS is included in the scheme as well as the generator for the disturbances (up to three sinusoids). A sample of the system noise in open loop is added to the simulation (a record of 10000 samples). The saturation of the actuator is included in the simulator. The control scheme (Controller) should be built around the given simulator using the actual "black box" model or an identified one. The simulator has been used by the participants to the benchmark to set the appropriate control scheme and test the performance. It will also be used by the organizers to test the various solutions proposed before their implementation on the real system.

6. REAL TIME IMPLEMENTATION

The real time implementation use the Matlab xPC Target environment (2007). The PC for program development is a Dell Optiplex 760. The PC target (Dell Optiplex GX270 with Pentium4 at 2.86 GHz) is equipped with I/O data acquisition, A/D and D/A converters

The procedure will compile the algorithms directly from the Simulink scheme provided by the participants. The execution time should be less than 1.25 ms

The experiments on the benchmark test bed (for all the contributions) will be done by the organizers of the benchmark.

7. CONTROL SPECIFICATIONS

The non parametric models of the primary path and of the secondary path obtained by the spectral analysis method shows that there are several resonance modes modes. The most significant are those near 50 Hz (secondary path) and 100 Hz (primary and secondary paths).

The narrow band disturbances are located in the range 45 to 105 Hz. It is important to take in account the fact that the secondary path (the actuator path) has no gain at very low frequencies and very low gain in high frequencies near 0.5 Fs. Therefore the control system has be designed such that the gain of the controller be very low (or zero) in these regions(preferably 0 at 0.5Fs). Not taking in account these constraints can lead to undesirable stress of the actuator.

There are three level of difficulty corresponding to one, two or three unknown time varying narrow band disturbances

**Level 1**: Rejection of a single time varying sinusoidal disturbance within 45 and 105 Hz

**Level 2**: Rejection of two time varying sinusoidal disturbances within 45 and 105 Hz

**Level 3**: Rejection of three time varying sinusoidal disturbances within 45 and 105 Hz

The control objectives for all levels can be expressed as follows:

**Protocol 1**: Step changes in frequencies. Time domain performances.

Test 1: Step application of the disturbances Test 2: Step changes in the frequencies of the disturbances An upper bound for the duration of the adaptation transient is imposed (2 sec) since the time separation between the steps in frequencies will be between 2 and 3 sec. However, the transients will be further compared in terms of maximum value and duration.

Test sequences have been provided to the participants **Protocol 2**. Evaluation in steady state operation at the frequencies considered above (after adaptation settles).

Test 1: The steady state performance in time domain will be evaluated by measuring the mean square value of the residual force which will be compared with the value of the residual force in open loop (providing a measure of the global attenuation.

Test 2: Power spectral density performances. For constant frequency disturbances, once the adaptation transient is over, the performance with respect to the open loop will be evaluated as
The transients for the case when the disturbances vanish will be made. From 1s to 16s two constant frequencies [70-100Hz] are applied on the shaker. From 6 to 11s two chirps between [60-90Hz] and [70-100Hz] are applied. After that, an inverse operation (from [70-100Hz] to [60-90Hz]) is considered. At 21s, constant disturbances of [60-90Hz] are applied and the tests are stopped after 26s. The time-domain results obtained in open and in closed loop are presented in Figure 10.

8. SOME EXPERIMENTAL RESULTS

It what follows some examples of the type of experiments which will be made on the real system will be presented. Two simultaneous time varying frequency sinusoids will be considered as disturbances. Time domain results obtained in "adaptive" operation regime are shown in Figure 8. The disturbances are applied at 1s (the loop has already been closed) and step changes of their frequencies occur every 3s. The convergence of the output requires less than 0.7s in the worst case.

Figure 9 shows the spectral densities of the residual force obtained in open loop and in closed loop using the adaptation scheme (after the adaptation algorithm has converged). The results are given for the simultaneous applications of two sinusoidal disturbances (65Hz and 95Hz). One can remark a strong attenuation of the disturbances (larger than 45dB).

We note the presence of some harmonics in open loop, as a consequence of a slightly non-linear behavior of the primary path. These harmonics are reduced in adaptive closed loop operation. There is also a permanent measurement noise at 50Hz (the power network), which is however not amplified in closed loop.

The variance of the residual force in open loop is: \( \text{var}(y_{ol}) = 1.087510^{-1} \). In closed loop (after the adaptation algorithm has converged), the variance is: \( \text{var}(y_{cl}) = 5.6428.10^{-5} \). This corresponds to a global attenuation of 66dB.

Consider now that the frequencies of the two sinusoidal disturbances vary continuously and let us use a chirp disturbance signal (linear swept-frequency signal) between [60-90Hz] and [70-100Hz]. The test has been done as follow: Start up in closed loop at \( t = 0 \) with the central controller \((S_0,R_0)\). The adaptation algorithm works permanently. At 1s, two sinusoidal disturbances of [60-90Hz] (constant frequencies) are applied on the shaker. From 6 to 11s two chirps between [60-90Hz] and [70-100Hz] are applied. From 11s to 16s two constant frequencies [70-100Hz] are applied. After that, an inverse operation (from [70-100Hz] to [60-90Hz]) is considered. At 21s, constant disturbances of [60-90Hz] are applied and the tests are stopped after 26s. The time-domain results obtained in open and in closed loop are presented in Figure 10.

<table>
<thead>
<tr>
<th>Protocol 3</th>
<th>Chirp changes in frequency. Time domain evaluation.</th>
</tr>
</thead>
<tbody>
<tr>
<td>A linear time varying change between two situations will be made. The maximum value of the residual force during the chirp will be measured as well as the mean square value of the residual force.</td>
<td></td>
</tr>
</tbody>
</table>

Test sequences have been provided to the participants. The loop is closed before the disturbances are applied for all the above tests.

**Supplementary tests:**

The operation of the system should remain stable for all the levels if one, two or three sinusoidal disturbances are applied simultaneously.

The operation of the loop should remain stable if the disturbance is applied simultaneously with the closing of the loop.

The transients for the case when the disturbances vanish will be tested and measured for all the levels.

Routines for executing the protocols and the measurements have been provided (see www.lag.ensieg.inpg.fr/jairo/benchmark).

For the final tests on the real system the frequencies of the disturbances will be randomly chosen within given bounds (but the same for all the participants).

<table>
<thead>
<tr>
<th>Control specifications</th>
<th>Level 1</th>
<th>Level 2</th>
<th>Level 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum disturbance attenuation</td>
<td>( \geq 45dB )</td>
<td>( \geq 45dB )</td>
<td>( \geq 40dB )</td>
</tr>
<tr>
<td>Maximum amplification</td>
<td>( \leq 6dB )</td>
<td>( \leq 6dB )</td>
<td>( \leq 9dB )</td>
</tr>
</tbody>
</table>

Table 1. Control specifications in the frequency domain.
For the preliminary version, real time results are not yet available from all the participants and therefore a comparative evaluation of performances is not available.

Results on the simulator are presented by each participating team.

REFERENCES


