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Benchmark on adaptive regulation—rejection of unknown/time-varying multiple narrow band disturbances

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ABSTRACT

Adaptive regulation is an important issue with a lot of potential for applications in active suspension, active vibration control, disc drives control and active noise control. One of the basic problems 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 information upon the disturbances. An adaptive feedback approach has to be considered for this problem. Industry needs to know the *state of the art* in the field based on a solid experimental verification on a representative system using currently available technology. The paper presents a benchmark problem for suppression of multiple unknown and/or time-varying vibrations and an associated active vibration control system using an inertial actuator with which the experimental verifications have been 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 real-time information about the disturbances.

The benchmark has three levels of difficulty and the associated control performance specifications are presented. A simulator of the system has been used by the various contributors to the benchmark to test their methodology. The procedure for real-time experiments is briefly described.[1] The performance measurement methods used will be presented as well as an extensive comparison of the results obtained by various approaches.[2]

1. Introduction

One of the basic problems in active vibration control (AVC) and active noise control (ANC) is the (strong) attenuation of multiple narrow band disturbances[3] with unknown and varying frequencies.

Solutions for this problem using adaptive feedforward compensator techniques have been proposed by signal processing community (see for example [13,29]). These solutions ignore the possibilities offered by feedback and require additional transducers for obtaining correlated measurements with the disturbance (they should provide the "reference" for the feedforward compensator). This approach has a number of disadvantages: (1) it requires the use of additional transducers; (2) implies often a difficult choice for the location of this additional transducers in order to get a relevant image of the disturbance; (3) In many situations the interaction between the compensator system and the measurement of the disturbance cannot be avoided (positive feedback causing stability problems—see [20, Chapter 15]); (4) it requires adaptation of many parameters.

However it is possible to achieve attenuation (rejection) of narrow band disturbances without measuring them by using a feedback approach. A common framework is the assumption that the disturbance is the result of a white noise or a Dirac impulse passed through the *model of the disturbance*. The knowledge of this model allows the design of an appropriate controller. When considering the model of a disturbance, one has to address two issues: (1) its structure (complexity, order of the parametric model) and (2) the values of the parameters of the model. In general, one can assess from data the structure for such *model of disturbance* (using spectral analysis or order estimation techniques) and assume that the structure does not change. However the parameters of the model are unknown and may be time varying. This will require the use of an adaptive feedback approach.[4]

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[1] The GIPSA-LAB team has done the experiments for all the contributors.

[2] Simulation and Real-time results are presented by each contributor in their papers [3,11,17,30,10,1,7].

[3] Called "tonal" disturbances in active noise control.

[4] Since it is not possible to design a robust controller which introduces a strong attenuation over a large frequency region as a consequence of the Bode Integral (water bed effect), one can not construct a single controller achieving strong attenuation of disturbances with varying frequencies.

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The classical adaptive control paradigm deals essentially with the construction of a control law when the parameters of the plant dynamic model are unknown and time varying [20]. However, in the present context, the plant dynamic model is almost invariant and it can be identified. The objective then is the rejection of disturbances characterized by unknown and time varying disturbance models. It seems reasonable to call this paradigm as *adaptive regulation*. In classical "adaptive control" the objective is tracking/disturbance attenuation in the presence of unknown and time varying plant model parameters. Therefore adaptive control focuses on adaptation with respect to plant model parameter variations. The model of the disturbance is assumed to be known and invariant. Only a level of attenuation in a frequency band is imposed (with the exception of DC disturbances where the controller may include an integrator). In *adaptive regulation* the objective is to asymptotically suppress (attenuate) the effect of unknown and time-varying disturbances. Therefore *adaptive regulation* focuses on adaptation of the controller parameters with respect to variations in the disturbance model parameters. The plant model is assumed to be known. It is also assumed that the possible small variations or uncertainties of the plant model can be handled by a robust control design. The problem of adaptive regulation as defined above has been previously addressed in a number of papers [26–28,6,4,12,15,16,24,18,2,9,14,5,8] among others. Ref. [19] presents a survey of the various techniques (up to 2010) used in adaptive regulation as well as a review of a number of applications.

Industry needs to know the *state of the art* in the field based on a solid experimental verification on a benchmark. The objective of the proposed benchmark is to evaluate on an experimental basis the available techniques for adaptive regulation in the presence of unknown/time varying multiple narrow band disturbances. Active vibration control constitutes an excellent example of a field where this situation occurs. But similar situations occur in disc drive control and active noise control. Solutions for this problem in active vibration control can be extrapolated to the control of disc drives and active noise control (see for example the applications described in [19]). The benchmark will effectively test various approaches in the specific context of an active vibration control system which will be used as a test bed.

The scientific objective of the benchmark is to evaluate current available procedures for adaptive regulation which may be applied in the presence of unknown/time varying multiple narrow band disturbances. The benchmark specifically will focus in testing: (1) performance, (2) robustness and (3) complexity.

The test bed is an active vibration control system using an inertial actuator and equipped with a shaker and a measurement of the residual force. It is located at GIPSA-Lab, Grenoble (France).[5] The test bed is representative of many situations encountered in practice and in particular of light weighted mechanical structures featuring strong resonance and antiresonance behavior[6] and impacted by vibration sources of different frequencies.

The paper is organized as follows. Section 2 gives a description of the active vibration control system used, as well as some information about the simulator. Section 3 gives the basic equations describing the system and the disturbance along with some information upon the identified models. Section 4 presents the control specifications as well as the protocols used on the benchmark. Section 5 describes some differences found between the simulator and the real plant and how these were taken into account. A methodological comparison of the various approaches is made in Section 6. The description of the measurements used for the analysis is given in Section 7. Section 8 gives the evaluation criteria defined with respect to the benchmark specifications as well as a comparison of obtained results. The complexity evaluation is done in Section 9 and the performance robustness with respect to experimental protocol changes is analyzed in Section 10. The main conclusions for this benchmark are given in Section 11. Appendix A presents a comparison of the adaptation algorithms used by the various contributors.

2. An active vibration control system using an inertial actuator

2.1. System structure

The basic structure of an active vibration control system using an inertial actuator is shown in Fig. 1. The inertial actuator will create vibrational forces which can counteract the effect of vibrational disturbances (inertial actuators use a similar principle as loudspeakers). A general view of the benchmark system including the testing equipment is shown Fig. 2. It consists of a passive damper, an inertial actuator, a mechanical structure, a transducer for the residual force, a controller, a power amplifier and a shaker. The mechanical construction is such that the vibrations produced by the shaker, fixed to the ground, are transmitted to the upper side, on top of the passive damper. The inertial actuator is fixed to the chassis where the vibrations should be attenuated. The controller, through the power amplifier, generates current in the moving coil which produces motion in order to reduce the residual force. The equivalent control scheme is shown in Fig. 3. The system input, $u(t)$ is the position of the mobile part (magnet) of the inertial actuator (see Figs. 1, 3 and 4), the output $y(t)$ is the residual force measured by a force sensor. The transfer function ($q^{-d_1}C/D$), between the disturbance force, $u_p(t)$, 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 ($q^{-d}B/A$) between the input of the inertial actuator, $u(t)$, and the residual force is called *secondary path*. Since the input of the system is a position and the output a force, the secondary path transfer function has a double differentiator behavior.

The block diagram of the active vibration control system emphasizing the hardware aspects is shown in Fig. 4.

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. This requires that the compensator system (the secondary path) has enough gain in the frequency range where the narrow band disturbances are located [22]. 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 $F_s = 800$ Hz.

Data used for system identification as well as the models identified from these data by the organizers are available on the benchmark website (http://www.gipsa-lab.grenoble-inp.fr/~ioandore.landau/benchmark_adaptive_regulation/index.html).

2.2. Simulator

A *black box* discrete time simulator of the active suspension built on MATLAB© Simulink (2007 version) has been provided (can be downloaded from the benchmark website). It uses the models identified by the organizers.

The control scheme (*Controller*) should be built around the given simulator. The simulator has been used by the participants to the benchmark to set the appropriate control scheme and test the performance.

[5] A first version of the test bed and benchmark specifications has been made available in 2010. Unfortunately because of some hardware and mechanical problems the test bed was redesigned and rebuilt. In this initial benchmark project J. Martinez-Molina, M. Alma and A.Karimi have been involved.

[6] I.e. very low damped complex poles and zeros.