Chapter 2
The Test Benches

2.1 An Active Hydraulic Suspension System Using Feedback Compensation

The structure of the active hydraulic suspension (active isolation configuration) is presented in Fig. 2.1. Two photos of the system are presented in Fig. 2.2 (Courtesy of Hutchinson Research Center, Vibrachoc and GIPSA-LAB, Grenoble.) It consists of the active suspension, a load, a shaker and the components of the control scheme. The mechanical construction of the load is such that the vibrations produced by the shaker, fixed to the ground, are transmitted to the upper side of the active suspension. The active suspension is based on a hydraulic system allowing to reduce the overpressure at the frequencies of the vibration modes of the suspension. Its components are (see Fig. 2.1):

- an elastomer cone (1) which marks the main chamber filled up with silicon oil;
- a secondary chamber (2) marked by a flexible membrane;
- a piston (3) attached to a motor (when the piston is fixed, the suspension is passive);
- an orifice (4) allowing the oil to pass between the two chambers; and
- a force sensor located between the support and the active suspension.

The size of the main chamber of the elastomer cone is modified by the effect of the piston driven by a linear motor. The controller will act upon the piston (through a power amplifier) in order to reduce the residual force. The equivalent control scheme is shown in Fig. 2.3. The system input, $u(t)$ is the position of the piston (see Fig. 2.1), the output $y(t)$ is the residual force measured by a force sensor. The transfer function 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 controlled by a signal given by the computer. The transfer function between the input of the system, $u(t)$, 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 behaviour. The sampling frequency used is $f_s = 800\,\text{Hz}$.

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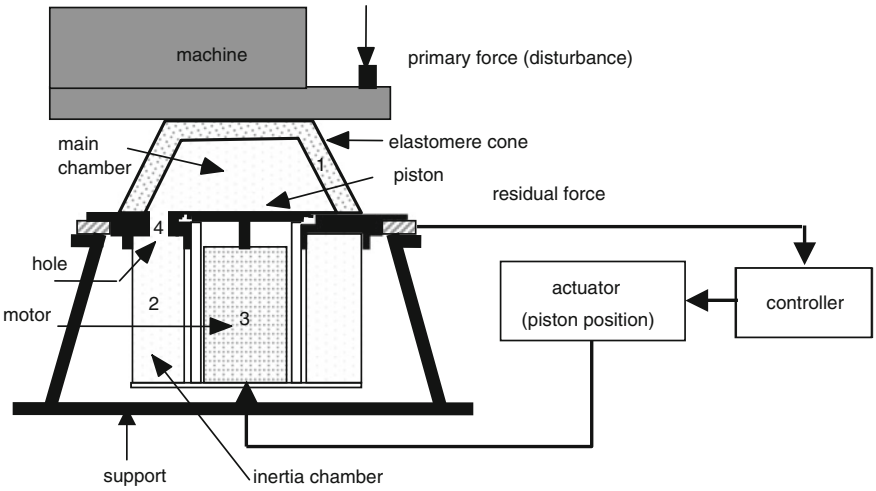


Fig. 2.1 Active suspension system (scheme)

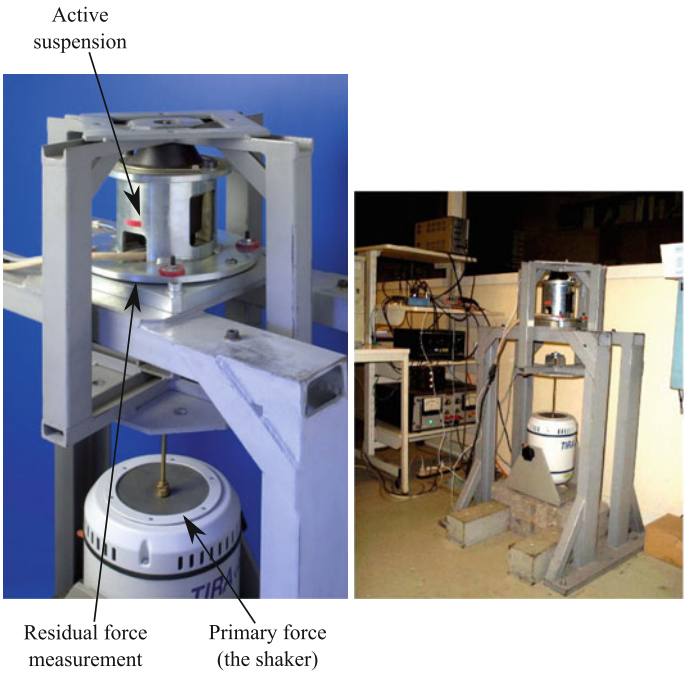


Fig. 2.2 Active suspension system (*left*). View of the experimental setting (*right*) (Courtesy of Hutchinson Research Center, Vibrachoc and Gipsa-lab, Grenoble, France.)

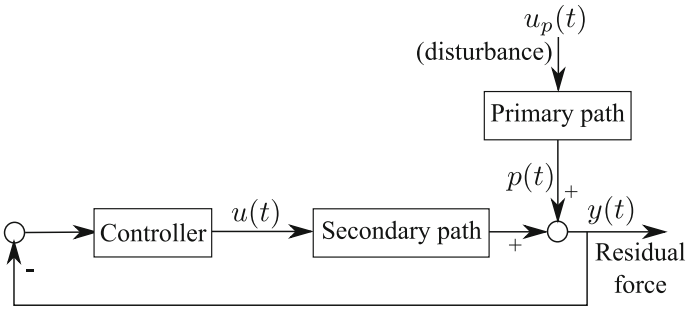


Fig. 2.3 Block diagram of active vibration control systems

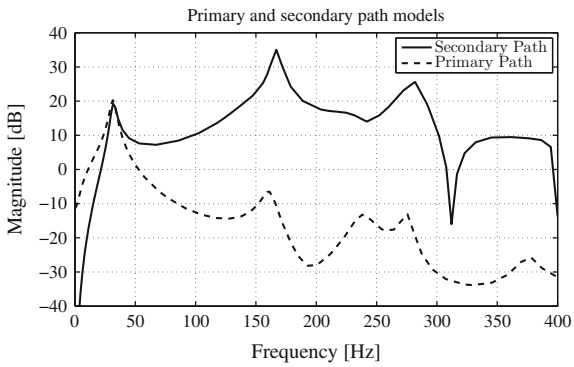
The control objective is to strongly attenuate (cancel) the effect of unknown narrow-band disturbances on the output of the system (the residual force).

The system has been considered as a "black box". A system identification procedure has been used in order to obtain the dynamic model of the system (called also *control model*) to be used for control design. The identification procedure will be described in Sect. 6.1.

The frequency characteristic of the identified primary path model (open-loop identification), between the signal u_p sent to the shaker in order to generate the disturbance and the residual force $y(t)$, is presented in Fig. 2.4. The first vibration mode of the primary path model is near 32 Hz. The primary path model has been only used for simulation purposes.

The frequency characteristic of the identified secondary path model (open-loop operation) is presented also in Fig. 2.4. There exist several very low damped vibration modes on the secondary path, the first one being at 31.8 Hz with a damping factor 0.07. The identified model of the secondary path has been used for the design of the controller.

Fig. 2.4 Frequency characteristics of the primary and secondary paths



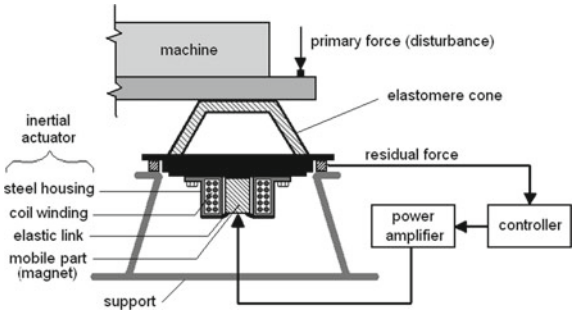


Fig. 2.5 Active vibration control using an inertial actuator (scheme)

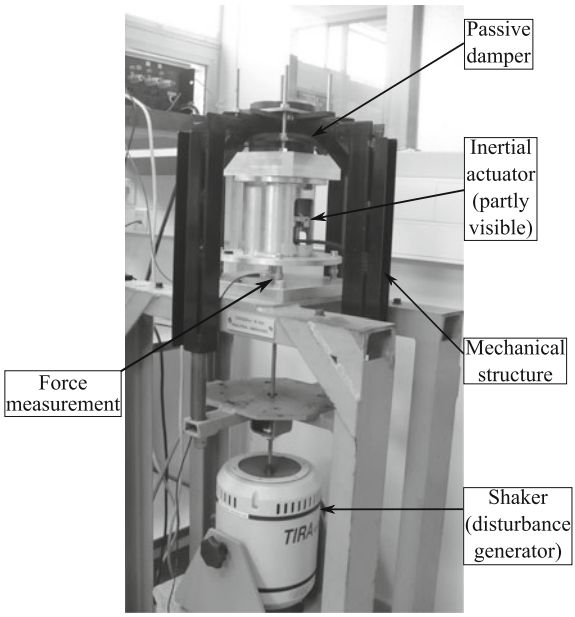


Fig. 2.6 The active vibration control system using an inertial actuator (photo)

2.2 An Active Vibration Control System Using Feedback Compensation Through an Inertial Actuator

The structure of the system is presented in Fig. 2.5. A general view (photo) of the system including the testing equipment is shown Fig. 2.6. It consists of a passive damper, an inertial actuator,[1] a chassis, a transducer for the residual force, a controller, a power amplifier, a shaker and a load which also transmits the vibration from the shaker to the chassis. The mechanical construction of the load is such that the vibrations produced by the shaker, fixed to the ground, are transmitted to the upper

[1] Inertial actuators use a similar principle as loudspeakers (see [1]).

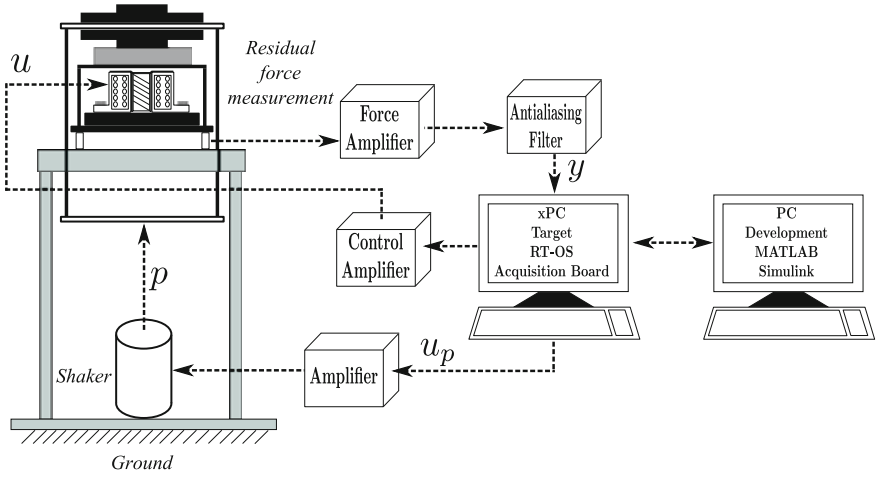


Fig. 2.7 The active vibration control system using an inertial actuator—hardware configuration

side, on top of the passive damper. The inertial actuator will create vibrational forces which can counteract the effect of vibrational disturbances.

The equivalent control scheme is shown in Fig. 2.3. The system input, $u(t)$ is the position of the mobile part (magnet) of the inertial actuator (see Fig. 2.5), the output $y(t)$ is the residual force measured by a force sensor. The transfer function 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 between the input of the inertial actuator, $u(t)$, and the residual force is called *secondary path*.

The complete hardware configuration of the system is shown in Fig. 2.7. The control objective is to cancel (or at least strongly attenuate) 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 been considered as a *black box* and the corresponding models for control have been identified from data. The details of the identification procedure will be given in Sect. 6.2. The sampling frequency is $f_s = 800\,\text{Hz}$.

Figure 2.8 gives the frequency characteristics of the identified models for the primary and secondary paths.[2] The system itself in the absence of the disturbances features a number of low damped resonance modes and low damped complex zeros (anti-resonance).

More details can be found at: http://www.gipsa-lab.grenoble-inp.fr/~ioandore.landau/benchmark_adaptive_regulation/.

[2]The primary path model is used only for simulation purposes.

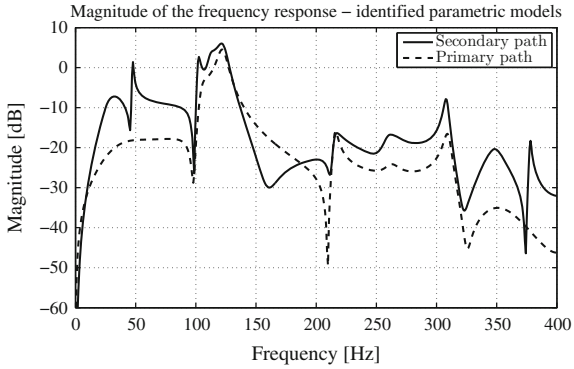


Fig. 2.8 Frequency response (magnitude) for the primary and the secondary path models

2.3 An Active Distributed Flexible Mechanical Structure with Feedforward–Feedback Compensation

Figure 2.9 shows a distributed flexible mechanical structure equipped for implementing an AVC using feedforward and feedback compensation. Figure 2.10 shows the details of the complete system including the AVC control scheme. The corresponding control block diagram is shown in Fig. 2.11.

The mechanical structure consists of five metal plates connected by springs. The uppermost and lowermost ones are rigidly jointed together by four screws. The middle three plates will be labeled for easier referencing M1, M2 and M3 (see Fig. 2.10). M1 and M3 are equipped with inertial actuators. The one on M1 serves as disturbance generator (inertial actuator I in Fig. 2.10), the one at the bottom serves for disturbance compensation (inertial actuator II in Fig. 2.10). The correlated measurement with the disturbance (image of the disturbance) is obtained from an accelerometer which is positioned on plate M1. Another sensor of the same type is positioned on plate M3

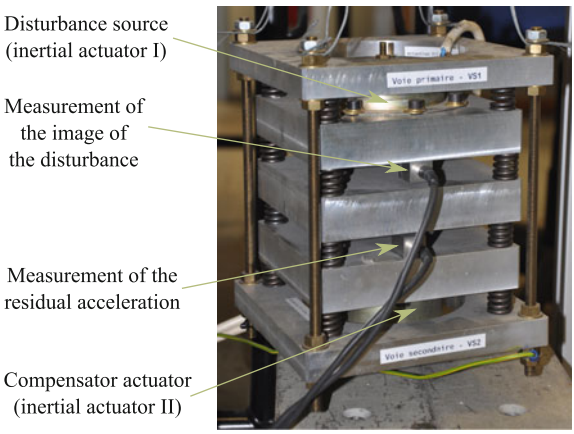


Fig. 2.9 An active distributed flexible mechanical structure equipped with sensors and actuators for feedforward–feedback disturbance compensation (photo)

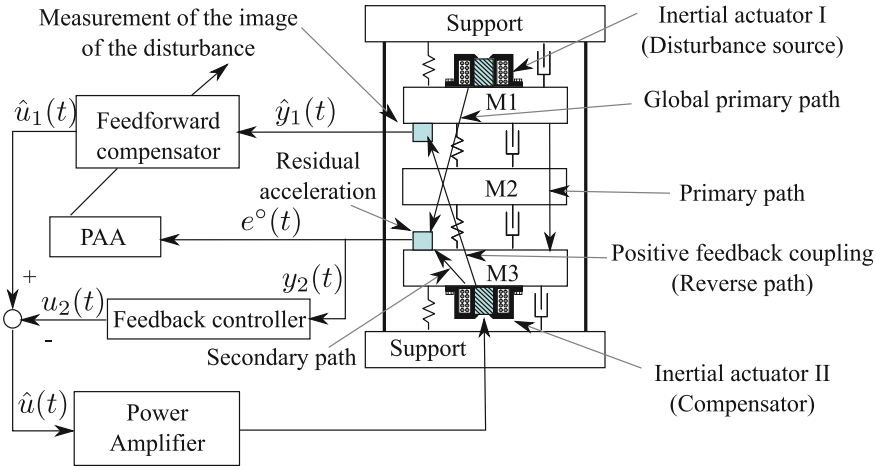


Fig. 2.10 An AVC system using feedforward and feedback compensation for the distributed flexible mechanical structure (scheme)

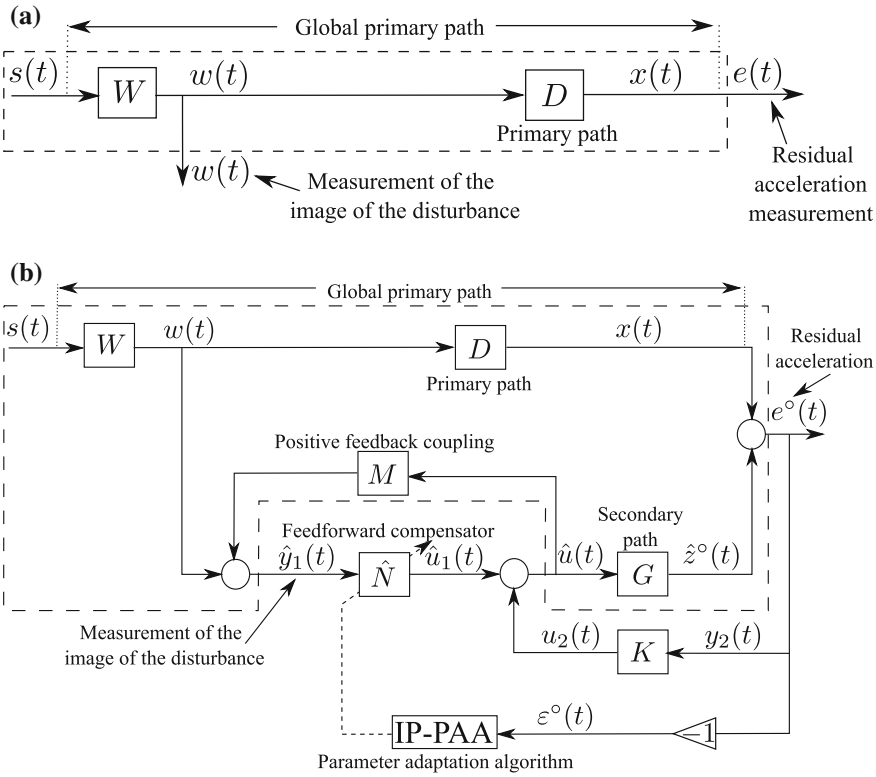


Fig. 2.11 Feedforward–feedback AVC—the control scheme: **a** in open loop and **b** with adaptive feedforward + fixed feedback compensator

and serves for measuring the residual acceleration (see Fig. 2.10). The objective is to minimize the residual acceleration measured on plate M3. This experimental setting allows to experiment both adaptive feedforward compensation (with or without additional feedback) as well as adaptive feedback disturbance compensation alone (without using the additional measurement upstream).

The disturbance is the position of the mobile part of the inertial actuator (see Figs. 2.9 and 2.10) located on top of the structure. The input to the compensator system is the position of the mobile part of the inertial actuator located on the bottom of the structure. When the compensator system is active, the actuator acts upon the residual acceleration, but also upon the measurement of the image of the disturbance through the *reverse path* (a positive feedback coupling). The measured quantity $\hat{y}_1(t)$ will be the sum of the correlated disturbance measurement $w(t)$ obtained in the absence of the feedforward compensation (see Fig. 2.11a) and of the effect of the actuator used for compensation (positive internal mechanical feedback). This is illustrated in Fig. 2.12 by the spectral densities of $\hat{y}_1(t)$ in open-loop ($w(t)$) and when feedforward compensation is active (the effect of the mechanical feedback is significant).

As from the previous experimental settings, the system is considered as a *black box* and the models for control design have been obtained by system identification from input/output data. The details of the identification procedure are given in Sect. 6.3. The sampling frequency is $f_s = 800\,\text{Hz}$. The frequency characteristics of the identified models of the primary, secondary and reverse paths are shown in Fig. 2.13.

This mechanical structure is representative for a number of situations encountered in practice and will be used to illustrate the performance of the various algorithms which will be presented in this book.

At this stage it is important to make the following remarks when the feedforward filter is absent (open-loop operation):

- very reliable models for the secondary path and the "positive" feedback path can be identified;

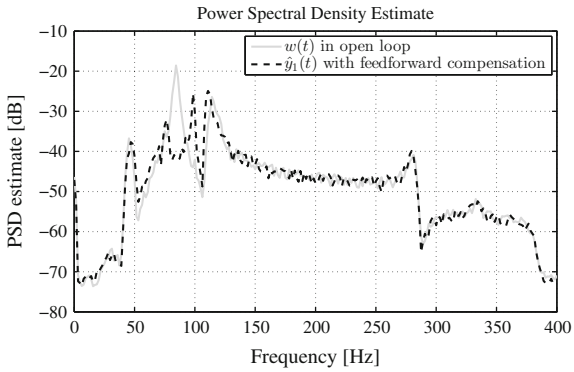


Fig. 2.12 Spectral densities of the image of the disturbance $\hat{y}_1(t)$ in open loop and when feedforward compensation scheme is active (experimental)

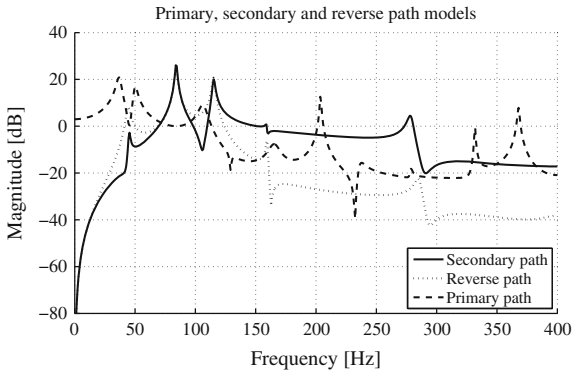


Fig. 2.13 Frequency response (magnitude) for the primary, secondary and reverse paths

- an estimation of the primary path transfer function can be obtained using the measured $w(t)$ as input and $e(t)$ as output (the compensator actuator being at rest);
- the quality of the primary path identified model will depend on the frequency characteristics of the signal $w(t)$ coming from the environment;
- design of a fixed model-based stabilizing feedforward compensator requires the knowledge of the reverse path model only;
- knowledge of the disturbance characteristics and of the primary, secondary and reverse paths models is mandatory for the design of an optimal fixed model-based feedforward compensator [2–4];
- **adaptation algorithms do not use information neither upon the primary path whose characteristics may be unknown nor upon the disturbance characteristics which may be unknown and time-varying.**

2.4 Concluding Remarks

- The test benches considered allow to evaluate different solutions for active vibration control and active damping.
- Their structure emphasizes the difficulties which may be encountered in practice.
- To obtain the complete dynamical models of these systems necessary for control design, identification of discrete time models from input/output data has been used (see Chap. 6).

2.5 Notes and References

For further details on these test benches see [5–8]. All the data for simulating the test bench presented in Sect. 2.2 is available at: http://www.gipsa-lab.grenoble-inp.fr/~ioandore.landau/benchmark_adaptive_regulation.

The book website provides input/output data and models for all the three test benches.

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