ABSTRACT
With the accuracy of metrology frame applications entering the nanometer-range, the necessity arises to tackle all types of disturbances. In the process of estimating the relative importance of the different types of disturbances on the machine accuracy, also called dynamic error budgeting, acoustic excitation has gained more importance, mainly because of the tightening of the performance requirements as well as the relatively high noise levels of air-conditioning systems in clean room. To estimate the dynamic error budget claimed by acoustic excitation already in the design phase of the machine, there is a clear need to predict this type of excitation in the design stage. This paper presents an efficient numerical procedure to predict the errors due to acoustic excitation and introduces an active structural damping concept to control acoustically induced vibrations.

To validate the numerical procedure, tests were done on a relatively simple structure consisting of a cantilever beam and a base plate that represents the typical structural dynamics of a practical high-precision mechatronic system. The sensitivity of this structure to diffuse acoustic sound field excitation, typical for the sound fields in clean rooms, was predicted and measured. The simulations agree reasonably well with the measurements in a reverberant room.

The level of metrology frame vibrations caused by internal or external sources, such as acoustically induced vibrations, is to a large extent determined by the (lack of) structural damping. This lack of damping is typical for metrology frames. Designing for stiffness, so as to achieve nanometer accuracy, inevitably implies a low intrinsic structural damping. In order to solve this dilemma, an alternative approach for the creation of active damping in the structure is proposed. The approach will be referred to as ASD - Active Structural Damping. The effectiveness of the ASD approach is demonstrated by simulation and experimental results on a cantilever beam and a base plate subjected to acoustic excitation.

NUMERICAL SIMULATION PROCEDURE
High precision equipment in clean rooms are subjected to relatively high acoustic noise levels. The acoustic noise spectrum of clean room air-conditioning systems usually dominates at frequencies around the 125 Hz 1/3rd octave band, reaching levels of up to 85 dB [1]. For many high precision mechatronic applications this implies that both the rigid body motion and the first internal resonance of a machine are excited significantly.

The nature of the acoustic noise in clean rooms is such that the acoustic sound fields are impinging random from all directions, called a diffuse sound field. In this section we will postulate an expression for the determination of the acoustic sensitivity of structures due to a diffuse sound field. The expression will be applied in the next section “Simulations and experiments”.

For linear systems there exists a relationship between the sound radiated by a vibrating mechanical system and the reciprocal problem, i.e. a mechanical system excited by sound. Following Cremer, Heckl and Petersson [2] the following relation can be derived

\[
\frac{|v'|^2}{|p'|^2} = \frac{8\pi}{k^2\rho^2c} \frac{P_{rad}}{|F|^2}
\]

where \(|v'|^2\) is the structural velocity response squared at a certain point \(A\) of the structure due to a diffuse acoustic sound field with a (spatially averaged) sound pressure level \(|p'|^2\) (the sound incidence case), whereas \(P_{rad}\) is the acoustic sound
power which is radiated by a force \( F \) acting on the same point \( A \) (the sound radiation case).

Relationship 1 can be used to determine the sensitivity of a structure to a diffuse acoustic field, expressed mathematically as \( |v'|^2/|p'|^2 \), simply by performing a reciprocal numerical experiment in which a mechanical force \( F \) acts upon the structure (at the point where we want to know the structural response due to the acoustic excitation) and determine the acoustic power \( P_{\text{rad}} \) radiated by the structure. In other words, the response of a structure may be determined from its radiation (and vice versa), provided a point force is employed and the incident sound field is random i.e., all directions of incidence are equally probable. For the interested reader reference is made to Roozen [3], [4] who used a similar approach.

Equation 1 will be used in the following section to calculate the sensitivity of structures due to acoustic excitation.

**SIMULATIONS AND EXPERIMENTS**

A relatively simple structure consisting of a cantilever beam and a base plate is investigated. The structural dynamics of this test set-up is typical for high-precision mechatronic systems in terms of its eigenfrequencies; the base plate is suspended with springs, as shown in Figure 1, resulting in suspension eigenfrequencies at around 1 Hz, whilst the first structural mode is at about 40 Hz, which is a mode for which the beam deforms relative to the base plate. Furthermore, the system is equipped with the standard components for active damping, as will be discussed in the next section. During the experiments discussed in this section, however, the active damping system is not activated.

A finite element model of the base-plate and cantilever beam was made, as shown in Figure 2. The first three (non-suspension) structural modes, including the already mentioned 40 Hz bending mode of the beam relative to the base plate, are shown in Figure 3. The numerical model predicts structural resonances at 43 Hz, 186 Hz and at 536 Hz. These three internal structural resonances as well as the six suspension modes with eigenfrequencies smaller than 1 Hz are used as a modal base for the description of the structural behavior of the mechanical system.
under investigation. A specific point of interest is the vibration of the tip of the cantilever beam. During the experiments a geophone mounted at the tip of the cantilever beam is used to measure the acoustically induced vibrations in a reverberant chamber test. For this reason, we are specifically interested in the structural dynamics at that point.

The test set-up was excited by a diffuse acoustic field in a reverberant room as shown in Figure 1. The reverberant room has a net volume of approximately 300 m$^3$.

Figure 4 compares the measured transferfunction from microphone pressure to geophone velocity with the numerically predicted result for the diffuse sound field excitation. The cumulative power spectra of the squared velocity over the squared acoustic pressure is given in Figure 5. Though the cumulative spectra look similar, there is a difference in the level of the measured and predicted cumulative power spectra, especially around the 60 Hz resonance. The reason for this discrepancy is most probably due to the rather difficult type of measurement in the reverberant room. Above 80 Hz the acoustic mode density of the room is sufficiently high for the acoustic field to be diffuse. Below this frequency the measurements are unreliable, causing this discrepancy. It can be noted that above 80 Hz, the measurements and simulations agree reasonably well.

**ACTIVE STRUCTURAL DAMPING**

The control of acoustically induced vibrations in high precision equipment can be done effectively by damping the structural resonances. "Sky hook damping" is a widely accepted method to damp suspension modes of a mechanical system in an active manner. However, in the tuning process of the sky hook damping the internal structural modes, which are the modes specifically excited by acoustic excitation rather than the suspension modes, generally show up in the open loop as parasitic dynamics. Fortunately, from a controller point of view this implies that those parasitic structural modes are both 'controllable' and 'observable'. The active structural damping (ASD) approach presented in this paper utilizes the presence of 'parasitic dynamics', and turns this into an advantage by creating active damping for those modes. Key is the implementation of a controller that uses the controllability and observability of the internal structural modes, so as to damp those modes. Furthermore it was found that an addition of a phase lag in the controller can significantly increase the achievable damping of the internal structural modes.

An extensive discussion of the ASD approach is beyond the scope of this paper. In this paper the major conclusions will be presented, with reference to [5] for a more detailed discussion.

The proposed method was tested by means of the test rig as shown in Figure 6. The test rig is equipped with the standard components for active suspension damping. The base plate houses 6 Lorentz actuators and 6 geophones for velocity measurements. For the initial measurements the first structural mode around 43 Hz is defined...
as the relevant structural mode, and one, vertical oriented actuator-sensor pair on a corner of the base plate will be used to dampen this structural mode. Applying DVF (direct velocity feedback), using a standard controller implementation without additional phase lag, resulted in 15% critical damping. This is confirmed with a model of the fitted plant. By adding a low pass filter to create phase lag, a relative damping of up to 39% critical was reached with manual loop shaping of the plant model. The effectiveness of the active damping is illustrated with a measurement of the following frequency response function. The system is excited by means of a nearby placed loudspeaker and the resulting velocity of the top of the beam is measured (using an additional geophone, not used by the controller). The reference signal for the measurement is the input signal of the loudspeaker. In Figure 7 the frequency response function from ‘excitation voltage’ to measured velocity is shown, with and without feedback. The influence of the feedback loop with extra phase lag in the controller on the 40 Hz mode is clearly visible. A more detailed discussion about the controller implementation can be found in [5].

CONCLUSIONS
A numerical procedure to predict the disturbances due to acoustic excitation of machinery is presented. The procedure appears to be a numerically efficient way (both in terms of CPU and in terms of disk usage) to predict the structural response due to diffuse sound field excitation. The procedure was validated on a relatively simple structure consisting of a cantilever beam and a base plate that represents the typical structural dynamics of a practical high-precision mechatronic system. The correspondence between experiments in a reverberant room and the numerical predictions were satisfactory.

By means of the ASD Active Vibration Damping concept, employing an additional phase lag in the controller, the acoustically induced vibrations can be controlled effectively.

REFERENCES