Nationaal Lucht- en Ruimtevaartlaboratorium
National Aerospace Laboratory NLR

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DEVELOPMENT OF AN ANALYSIS AND DESIGN ENVIRONMENT FOR HYBRID DAMPING

A. de Boer, R.P.G. Veul, P. Arendsen and M. Bakker*
National Aerospace Laboratory NLR
P.O. Box 153, 8300 AD Emmeloord, The Netherlands

Vibration reduction by adding damping to the structure is mostly carried out when the structure has been build. The fast developments in computer technology makes it possible that more complex analyses in which damping is included can be applied in the design process. At NLR research is carried out on the integration of advanced analysis tools in design environments. In this paper the tools developed for the analysis of passive and active damping in structures are presented.
The back bone of the design environment is an optimization algorithm which helps the designer to come up with optimal designs of structures. In the case of active damping the optimal design of controllers is a new aspect. This means that in addition to the optimization of the locations of sensors and actuators the control parameters have to be optimized. In this paper a method is proposed to optimize locations and control parameters at once with the standard finite element representation of the equations of motion as a base.

INTRODUCTION

Vibration reduction by adding damping to the structure is mostly carried out when the structure has been build. It is no common practice to analyze the effect of damping treatments already in the design process due to the complexity of the algorithms and the high computation times. However, the fast developments in computer technology makes it possible that more complex analyses can be applied in the design process.
At NLR research is carried out on the integration of advanced analysis tools in design environments. This started at the end of the eighties with a multi level optimization tool for preliminary design of aircraft structures (Ref.1). Currently this tool is extended to multi-disciplinary analyses and optimization (MDO) like aero-elasticity and vibro-acoustics.
Recently a study has been started to incorporate the analyses and optimization of passive and active damping treatments in the MDO environment. The basis for this study is the knowledge and developed analysis tools obtained from NLR research on tuned dampers, constrained layer damping and piezoelectric materials. This will be presented in the first part of this paper.

* University of Twente, Faculty of Mechanical Engineering, P.O. Box 217, 7500 AE Enschede, The Netherlands.

By implementing this knowledge in a MDO environment NLR wants to make it available for the engineer in the industry.
A new aspect which comes up with active damping is control. In the second part of this paper attention is paid to the incorporation of control in the equation of motion and the optimization analysis.

HYBRID DAMPING

At NLR research on damping is especially carried out with the intention to apply it for reduction of structural vibrations, reduction of noise transmission through fuselage walls and the application of Commercial Off The Shelf (COTS) equipment in severe environments.
Structures can be damped with passive treatments like visco-elastic constrained layers (Ref. 2) and tuned dampers (Ref. 3), with active means like piezoelectric materials (Ref. 4), shape memory alloys and magneto-restrictive materials or a combination of passive and active treatments which is called hybrid damping.
Simple and more complex methods (Ref. 5) have been developed to analyze passive damping treatments on a structure. Some of these methods have been implemented in the modular finite element program B2000 at NLR and are used to analyze the influence of different damping treatments on the transmission of noise through double walls (Ref. 6, 7, 8).

Passive damping

Special attention has been paid to the modeling of constrained layer damping (Ref. 7).

In literature different damping models are proposed such as structural damping, viscous damping, complex damping (Ref. 9), Augmented Thermodynamic Field damping (Ref. 5), Augmented Hooke's Law damping (Ref. 10). The choice of the model depends on the material but also the applicability in large numerical analyses. For instance a frequency dependent damping model increases the computation time needed for a frequency response analysis dramatically which makes it practically impossible to use it. From this point of view structural material damping is a favourite model because the damping is represented as a constant material property.

For structural damping the damping is defined by the ratio between the imaginary and real part of the elastic modulus E^* and the shear modulus G^*.

$$\eta_E = E^I/E^R \qquad (1)$$

$$\eta_G = G^I/G^R \qquad (2)$$

with: $E^* = E^R + iE^I$

$G^* = G^R + iG^I$

When the structure consists of an isotropic homogeneous material the shear modulus is derived from E^* and Poisson's ratio and therefore $\eta_G=\eta_E$. In that case the damping of the structure will be frequency independent. However, in the case a complex shear modulus is defined separately with a damping ratio $\eta_G \neq \eta_E$ a frequency dependent damping of the structure can be obtained (see also Ref. 2). Also for a layered structure, for example a visco-elastic layer between two constraining layers where $\eta_{Elayer1} \neq \eta_{Elayer2}$ and/or $\eta_{Glayer1} \neq \eta_{Glayer2}$, the damping of the structure is frequency dependent.

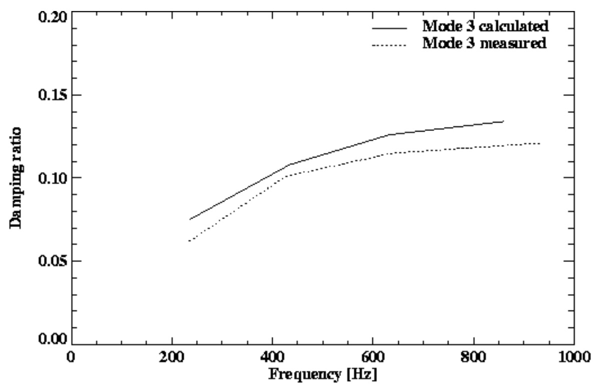

Figure 1 *Calculated and measured damping of the third bending mode as function of the frequency.*

This damping model has been implemented in B2000 and experimentally validated with aluminum strips. On these strips tape is bonded with a visco-elastic layer and a constraining aluminum layer. The strips have different lengths such that the eigenfrequencies belonging to a certain bending mode shape differ per strip. In this way the damping can be measured at different eigenfrequencies for a certain bending mode.

Figure 1 shows the measured and calculated damping ratios as function of the frequency for the third bending mode. In figure 2 the calculated damping ratios as function of time for the first three bending modes are presented. From these figures can be concluded that the measured and calculated damping ratios correspond good and that the damping ratios do not strongly depend on the kind of bending mode.

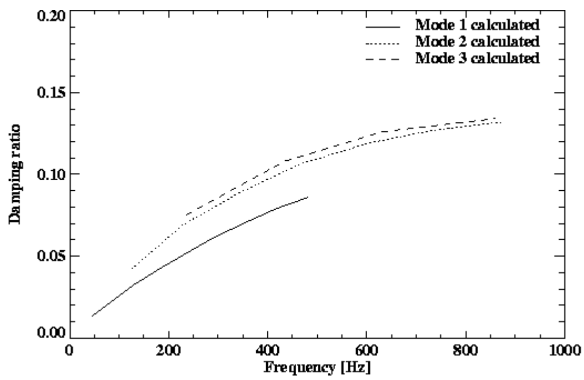

Figure 2 *Calculated damping for the first three bending modes as function of the frequency.*

Active damping

To be able to simulate active damping at NLR a finite element method has been developed with which piezoelectric material behavior can be applied for all existing structural finite elements. This is realized by creating piezoelectric elements as so-called "overlay" elements. This means that the element stiffness matrix of the piezoelectric element is assembled from an existing element stiffness matrix and the dielectric stiffness and coupling matrices of the overlay element. In this way piezoelectric material behavior can be simulated with all existing structural beam, shell and volume elements. The overlay element has been implemented in the modular finite element program B2000 and has been tested and verified with test cases from literature. The effect of piezoelectric material on the shape of a structure is demonstrated with a cantilevered laminated composite plate with on both the upper and lower surface a ceramic piezoelectric layer (see Fig. 3). Thirty six four node shell elements are used to model the plate. The composite plate is made of T200/976 graphite-epoxy composites with a stacking sequence of [-45/45-45/45]. Each layer has a thickness of 0.25 mm. The piezoelectric ceramic is PZT G1195N with a thickness of 0.1 mm.

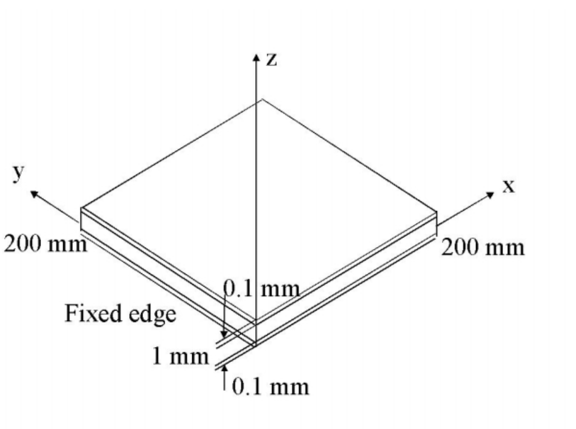

Figure 3 *A laminated composite plate with piezo-electric layers.*

The plate is exposed to a uniform distributed load of 100 N/m^2, a tip force of 0.1137 N and a tip moment of 0.01448 Nm separately. The deflections of the plate are calculated when a voltage of 1 V, 30 V and 50 V is applied over the upper piezoelectric layer and an opposite voltage over the lower piezoelectric layer. Figure 4 shows the calculated centerline deflection of the composite plate under these different active input voltages. It is observed that for 50 V the tip deflection is almost zero. Further is shown that for 50 V combined with the tip moment the deflection of the whole centerline is almost zero. So, with a piezoelectric layer which covers the whole plate the deformation due to a constant voltage corresponds with the deformation caused by a tip moment, as expected from the equations.

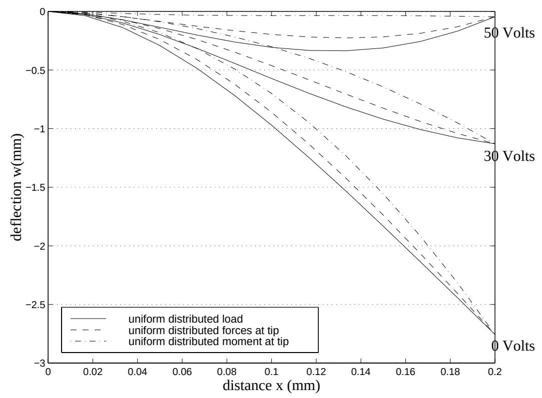

Figure 4 *The center line deflection under various loads and actuator input voltages.*

OPTIMIZATION

In the case damping treatments have to be added to a structure the mass of the structure will increase generally. Especially for light weight structures this is a disadvantage and therefore as less as possible damping material has to be added. This means that damping material (passive and/or active) has to be added at optimal locations.
At NLR a pilot study has been carried out in which the optimal locations of four tuned dampers on a vibrating rectangular plate have been calculated such that the response of the plate in a certain frequency range is minimal (Ref. 3). The initial positions of the four dampers were the centers of each quarter of the plate. The measured frequency response function (FRF) of a point on the plate for this initial configuration in the frequency interval from 350 to 550 Hz is depicted in figure 5a. The FRF in that point for the situation that the positions of the dampers have been optimized such that the FRF in the frequency interval from 400 to 500 Hz is minimal, is depicted in figure 5b. This figure shows that there is a tendency to minimize the FRF between 420 and 480 Hz and to squeeze the maximum of the FRF to the optimization boundaries at 400 and 500 Hz.

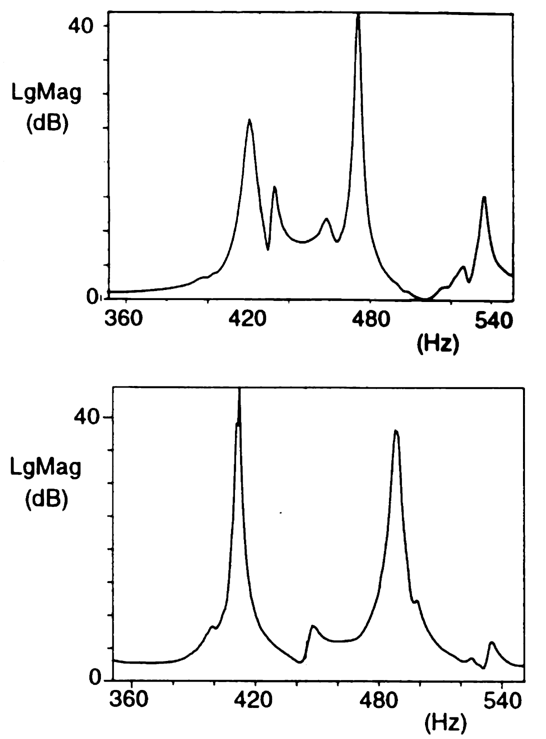

Figure 5 *FRF in a point of the plate with tuned dampers in the initial positions (upper) and the optimized positions (lower).*

For active damping both sensors and actuators have to be positioned optimally. In principle this can be achieved with the same algorithms as used for the tuned dampers. However, in this case the (optimal) locations of the sensors and actuators do not only determine the dynamical behavior of the structure, also the control algorithm does. Therefore, control has to be incorporated in the optimization loop, too. This can be done sequentially, so first the determination of the optimal locations of the sensors and actuators and then for one or more structures with "optimal" sensor and actuator location the optimization of the control algorithm. The best way is to couple both analyses in one optimization loop.

CONTROL

The basic ingredients for the analysis and design environment for hybrid damping are available in the analysis program B2000 namely dynamic and vibro-acoustic analysis tools, optimization algorithms, passive damping algorithms and piezoelectric overlay elements. The objective of this study is to combine these tools and to come up with a tool with which the effect of passive and/or active damping on the dynamic and acoustic behavior of the structure can be predicted and optimized.
To perform a coupled structural dynamic and control optimization in B2000 it should be possible to incorporate the control parameters in the equations of motion. For a structure with actuators and sensors the finite element representation of these equations is:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\} - \{F_C\} \qquad (3)$$

with:
- $[M]$ the mass matrix
- $[C]$ the damping matrix
- $[K]$ the stiffness matrix
- $\{F\}$ the applied force
- $\{F_C\}$ the applied force by the actuators
- $\{x\}$ the vector with nodal displacements and voltages
- $\dot{}$ the derivative with respect to time

In the case of PID control of the velocity, the actuator force can be written as:

$$\{F_C\} = [R_I]\{x\} + [R_P]\{\dot{x}\} + [R_D]\{\ddot{x}\} \qquad (4)$$

with:
- $[R_I]$ the matrix with parameters of the I(ntegration) part of the control
- $[R_P]$ the matrix with parameters of the P(roportional) part of the control
- $[R_D]$ the matrix with parameters of the D(ifferential) part of the control

Substitution of eq. 4 into eq. 3 gives:

$$([M]+[R_D])\{\ddot{x}\}+([C]+[R_P])\{\dot{x}\}+([K]+[R_I])\{x\}=\{F\} \qquad (5)$$

The values for the control parameters in the matrices $[R_I]$, $[R_P]$ and $[R_D]$ can now be obtained with the same optimization methods as used for the determination of the optimal locations of actuators and sensors.
In structural dynamics the dynamic behavior of the structure is optimized by varying the structural properties such as thickness, mass density or damping ratio. This means that the components of the mass, damping and/or stiffness matrices are changed. With the same optimization procedure the values for the control parameters in $[R_I]$, $[R_P]$ and $[R_D]$ can be determined.

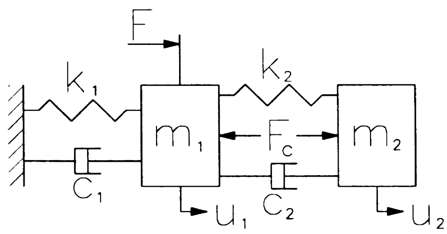

Figure 6 *Two degrees of freedom mass, spring, dashpot system with control force.*

As an example the control parameters are determined for the two degrees of freedom (DOF) system as depicted in figure 6. The properties are summarized in table 1.
If full state feedback is applied with a PD controller, the control force F_c situated between mass 1 and mass 2 depends on the displacements and the velocities and can be written as:

$$F_c = \{r_1\ r_2\}\begin{Bmatrix} u_1 \\ u_2 \end{Bmatrix} + \{r_3\ r_4\}\begin{Bmatrix} \dot{u}_1 \\ \dot{u}_2 \end{Bmatrix}$$

The values of the control parameters can be determined with the standard pole placement routine in MATLAB. This is carried out for the case that the goal is to obtain a system with a certain value for the modal damping with the assumption that the undamped frequencies are equal to the damped ones. The results are depicted in the second row of table 2. The values of the control parameters obtained by optimization of the standard equations of motion with the control parameters as unknowns in the stiffness and damping matrices are depicted in the third row of table 2. This table shows that for both methods approximately the same values for the control parameters are obtained.
At the moment research is going on to apply this

method to determine values for the control parameters for more complex structures.

CONCLUSIONS

At NLR research is carried out on the integration of advanced analysis tools in design environments. The fast development in computer technology makes it possible to apply more complex analyses in such a design environment. Therefore a study has been started to incorporate in the design environment knowledge and analysis tools in the field of passive and active damping treatment which have been obtained the last decade at NLR. In this way (theoretical) knowledge gathered by a research institute like the NLR comes available for the industry. The back bone of the design environment is an optimization algorithm which helps the designer to come up with optimal designs of structures.
In the case of active damping the optimal design of controllers is a new aspect. To incorporate this in the existing environment the control parameters have been included in the stiffness, damping and stiffness matrices of the finite element representation of the equations of motion.
Preliminary results show that this approach is promising.

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Table 1 *Properties of the two degrees of freedom (DOF) system.*

m_1	= 2 kg	m_2	= 3 kg
k_1	= 10 N/m	k_2	= 15 N/m
c_1	= 0.5 Ns/m	c_2	= 0.1 Ns/m
$u_1(0)$	= 0 m	$u_2(0)$	= 0 m
$u'_1(0)$	= 0 m/s	$u'_2(0)$	= 0 m/s

Table 2 *Calculated values for the control parameters obtained with pole placement method in MATLAB and optimization of the equation of motions.*

Parameter	r_1	r_2	r_3	r_4
Values obtained with Pole placement	-0.66	0.00	0.10	5.44
Values obtained with optimization	-0.62	-0.07	0.09	5.42