Review

Smart structure dynamics

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Received 31 May 2005; received in revised form 16 August 2005; accepted 22 August 2005
Available online 12 October 2005

Abstract

This paper gives an overview of research in the area of smart structure dynamics. A general description of smart material systems is given. Particular focus is given to the following fields of application: semi-passive concepts, energy harvesting, semi-active concepts, active vibration control, and active structural acoustic control. The use of smart structures in structural health monitoring applications is also considered.

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Keywords: Active vibration control (AVC); Active noise control (ANC); Shunted piezoelectricity; Energy harvesting; Semi-active damping; Structural health monitoring (SHM)

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doi:10.1016/j.ymssp.2005.08.025
1. Introduction

Passive measures for reducing noise and vibration or for ensuring optimal structural performance have reached certain limits. For this reason, smart structures are becoming increasingly important. As funds have become available to pursue research in this area, terminologies have been introduced to define the field of study. The terms smart structures, intelligent structures, adaptive structures, active structures, adaptronics, and structronics all belong to the same field of study [1]. All these terms refer to the integration of actuators, sensors in structural components, and the usage of some kind of control unit or enhanced signal processing with a material or structural component (Fig. 1). The goal of this integration is the creation of a material system having enhanced structural performance, but without adding too much mass or consuming too much power. Due to its nature, the field of smart structures depends on inter-disciplinary research since numerous disciplines (e.g. material science, applied mechanics, control theory, etc.) are involved in the design of a smart structure system solution.

The materials used in smart structures often have interesting and unusual properties. Electrostrictive materials, magnetostriuctive materials, shape memory alloys, magneto- or electrorheological fluids, polymer gels, and piezoelectric materials, for example, can all be used to design and develop structures that can be called smart. However, the materials themselves are not smart. “Smartness” refers to the exploitation of material properties to better serve a design function than would be possible through conventional structural design.

2. Modelling of smart materials and structures

The field of literature related to the modelling of smart structures is vast. Tzou et al. [3] recently presented an overview of smart materials and their modelling. In the same spirit, this paper presents a general summary of the more intensively researched smart materials.

2.1. Piezoelectric material

Piezoelectric elements serve as effective sensors and actuators in many smart structure applications. Even in the case of simple smart structures, such as transducer-implemented beams and plates, accurate and efficient
prediction of the response is a challenging task. Early smart structure models assumed that the inertial mass or stiffness of the structure is not affected by the presence of transducers [4,5]. This holds true in the case of piezopolymer polyvinylidene fluoride (PVDF) transducers, which are thin and which typically have a much smaller elastic modulus than the host structure. The assumption does not hold true in the case of piezoceramic PZT (lead zirconate titanate) transducers, which are relatively thick and which typically have an elastic modulus which is comparable to that of the host structure. Additionally, since the transducers are usually bonded to one side of the beam/plate structure, the neutral axis of the beam/plate is shifted. The classical assumption of pure bending about the geometric axis of symmetry then no longer applies [6]. Finally, since only pure bending is allowed by these early models, the curvatures which would occur due to applied twist moments, cannot be modelled.

Spurred by improvements in computing speed over the past few decades, the finite element approach has gained increased acceptance as a modelling method. The finite element (FE) method is capable of handling complex geometries and complex boundary conditions, as well as the transducer–structure coupling which occurs in smart structures. In a representative study, FE modelling was used to accurately predict the response of a cantilever beam instrumented with a piezoelectric sensor and actuator [7]. In this study, a three-dimensional (3-D) FE formulation is employed in the vicinity of the transducers. Flat-shell elements, connected to the 3-D elements by transition elements, are used to model the remaining part of the plate structure. The electrical input admittance and the sensor response predicted using the FE approach compared well with experimental measurements.

Piefort [6] presents an electromechanically coupled FE formulation. A review of FE modelling of piezoelectric structures is given, and a general electromechanically coupled FE formulation is derived by using the Hamilton’s principle. Additionally, a Kirchhoff piezoelectric multilayer element formulation is derived and then extended to a Mindlin formulation.

Lammering and Mesecke-Rischmann [8] focused on composite shell structures having thin piezoelectric patches bonded to the surfaces. The starting point for this analysis is the well-known two-field variational formulation. Specifically, a linear piezoelectric effect is assumed, and the displacements and the electric potential are taken as the independent variables. They compare a quadratic variation of the electric potential in the thickness direction with the classical linear variation. Additionally, they present a three-field variational formulation to allow for an alternative selection of the independent variables and non-linear material behaviour. Adopting this variational principle, they derived a hybrid FE in which the dielectric displacement is formulated as an additional degree of freedom.

Hagedorn et al. [9,10] worked on travelling wave ultrasonic motors based on piezoelectric elements. They noted that the standard linear material description for piezoceramics is insufficient when it comes to applications in which the piezoceramic is used for energy transformation (from electrical to mechanical). In a number of projects they developed non-linear material laws for piezoceramics and identified the corresponding material parameters [11].

2.2. Shape Memory Alloy

There are a wide range of application areas for shape memory alloys (SMA): medical devices, MEMS devices, actuators, and composites. In its low-temperature state, an SMA has a martensitic crystalline structure and can be deformed with relatively low force. When heated above a certain transition temperature, a solid–solid phase transition to an austenitic state occurs. Additionally, shape recovery from the deformed martensitic shape will occur. In the austenitic state, the material has a symmetric crystalline structure and a relatively high elastic modulus. In general, when the SMA recovers its austenitic form, it will produce a displacement, a force, or a combination of the two, depending on the boundary conditions. Above the transition temperature, superelastic behaviour is observed, that is, the SMA may be reversibly stretched or compressed five to ten times the amount of conventional materials. Finally, if the material’s temperature drops below the transition temperature, the alloy will assume its martensitic crystalline structure again [12].

SMAs may be heated via application of a thermal load, or more typically, via application of an electrical current. Subsequent cooling is achieved via free/forced convection of the surrounding medium and/or conduction via the host structure. In theory, the phase transformation process propagates with the speed of
sound, but only if the necessary heat is supplied or dissipated. Due to the limited heat transfer rate which is physically achievable, the dynamic response of typical SMA devices is limited to frequencies less than 1 Hz. SMAs are therefore not well-suited for the task of actively controlling structural vibrations using conventional feedback control.

The dynamic characteristics of a structure, including damping, natural frequencies, and mode shapes, can be influenced through the use of SMA actuators. The low-frequency (<4 Hz) damping properties of SMA (nickel–titanium, NiTi) wires loaded dynamically within the transformation range from austenite to martensite are investigated in Ref. [13]. The transformation between stress-induced martensite and austenite during loading and unloading is responsible for the observed hysteresis characteristic. It was found that the area of the measured hysteresis loop decreased at higher strain rates. The ability of the material to dissipate high-frequency structural vibrations is therefore limited. Certain tuning techniques can however be used to influence higher-frequency vibrations [14]. The stiffness of a SMA component bonded to or embedded in the structure will change as it transforms from martensite to austenite. As a result, the structure’s natural frequencies will be increased, and the vibration amplitudes will be decreased. The SMA component can also be attached to the structure in a prestrained state, so that large internal forces and changes in stiffness are developed during activation.

At present, the majority of SMA applications involve one-dimensional (1-D) wire, ribbon, or rod actuator shapes. Correspondingly, a lot of effort has been devoted to the development of 1-D models for SMA actuators. Three 1-D, phenomenological, macroscopic models are compared in [15]. According to these models, the total change in stress within a SMA consists of three components: an elastic component proportional to Young’s modulus and a change in strain; a transformational component proportional to a phase transformation coefficient and a change in the martensite volume fraction; and, a thermal component proportional to the thermoelastic coefficient and a change in temperature.

A particularly practical 1-D SMA model has been developed in [16]. This model expresses the constitutive relation for an 1-D SMA actuator in terms of an effective coefficient of thermal expansion,

$$\sigma = E(T)\left[\varepsilon - \int_{T_0}^{T} a(\tau) \, d\tau\right].$$

(1)

Here, $\sigma$ is the stress, $\varepsilon$ is the mechanical strain, $E$ is Young’s modulus, $T$ is the temperature, and $a$ is the effective coefficient of thermal expansion which includes thermal and transformational strain effects. $T_0$ denotes the reference temperature. In a recent study [17], the dynamic response of a clamped–clamped composite beam having embedded, prestrained SMA (NiTi) ribbon layers was measured and compared to an FE model of the beam incorporating the SMA constitutive law in Eq. (1). Measured thermomechanical parameters for the SMA and the composite material and measured damping ratios were implemented in the FE model. Excellent agreement was achieved between the measured and predicted beam responses in the examined frequency range 10–400 Hz and temperature range 20–120°C.

DesRoches et al. [18] have taken a multidisciplinary, multiscale research approach to evaluate the efficiency of shape memory alloys as seismic response modifiers in civil engineering structures. The research involves studies correlating thermomechanical processes with mechanical properties of large section shape memory alloys as well as evaluating the cyclical properties of SMAs (Fig. 2). They evaluated the cyclical characteristics of Nitinol SMA wires and bars to determine the recentring and damping properties as a function of bar diameter, cyclical strain, and loading frequency. The results show that with proper heat treatment, nearly ideal superelastic properties can be obtained in both wire and bar form of the superelastic NiTi shape memory alloys. However, the wire form of the shape memory alloys shows higher strength and damping properties compared with the bars. The recentring capabilities (based on residual strains) are not affected by section size. Overall, the damping potential of shape memory alloys in superelastic form is low for both wire and bars, typically less than 7% equivalent viscous damping. Cyclical strains greater than 6% lead to degradation in the damping and recentring properties of the shape memory alloys. Strain rate effects are evaluated by subjecting the shape memory alloys to loading rates representative of typical seismic loading. The results show that increased loading rates led to decreases in the equivalent damping, but have negligible effects on the recentring properties of the shape memory alloys.
2.3. Chemomechanical materials

Chemomechanical actuators are promising alternatives to conventional actuators in certain niche applications, such as in biomedical devices. As the name implies, these actuators convert chemical energy directly into mechanical energy. For example, when special polymers are immersed in a solvent, the polymer net will expand so as to maximise contact with the solvent. A volumetric expansion as large as 300% can be achieved by such materials. However, voltage-activated chemomechanical actuators are of more interest due to ease of controllability. Typical voltage-activated chemomechanical actuators include conducting polymers and ionic polymer actuators. Because the operation of such actuators is based on a diffusion process, they are not suited for high-frequency applications.

The volume-changing mechanism for a conducting polymer known as polypyrrole is discussed in [12,19] and is briefly described here. To begin, a p-doped (electron deficient) polymer film is electrochemically grown onto a positively charged electrode. The film is then material doped with a salt having a bulky anion (negatively charged ion), that is, an anion which due to its large size, would not be able to diffuse out of the polymer. When the salt is dissociated, the polymer is electrically neutral since the net negative charge of the trapped anions balances the net positive charge in the polymer matrix. When a voltage is applied across the polymer, electrons fill the holes in the polymer matrix, and the salt cations enter the polymer matrix in order to preserve charge neutrality. The anions and cations bond ionically to form the salt, and as a result, the polymer expands macroscopically. Depending on the magnitude of the applied voltage, such a conducting polymer can expand by as much as 2%. In order to magnify displacements, the conducting polymer can be bonded to a thin metallic layer such as bio-inactive gold. Shear stresses will be induced between the non-expanding metallic layer and the polymer upon expansion, thereby causing the bilayer to bend. As a result, considerable tip displacements can be achieved for relatively modest volume changes.

The actuation mechanism is not completely understood for the class of chemomechanical materials known as the ionic polymers. As described in [20], when an ionic polymer is hydrated, the cations associated with the $\text{SO}_3^-$ groups in the polymer become mobile. Application of a voltage across the polymer causes the cations to migrate from the positive to the negative electrode. According to one physical model, water molecules are dragged along with the cations, causing the water concentration to decrease at the positive electrode and to increase at the negative electrode. This causes contraction and expansion in the respective portions of the polymer, and curvature is induced in the actuator. According to another physical model, as the cations migrate, local charge imbalances induce stress in the individual polymer backbones. The resulting local strains induce macroscopic curvature of the actuator. Although there is disagreement about the physics of actuation in ionic polymers, practical engineering models for such materials are nonetheless possible. In recent studies, a linear electromechanical model for the ionic polymer Nafion (mfg. Dupont) was developed and experimentally...
verified [21,22]. The model is based on an equivalent circuit representation which is related to the mechanical, electrical, and electromechanical properties of the material. Expressions for the quasi-static and dynamic mechanical impedance are derived using beam theory. A series of experiments were performed to validate the linear circuit model. Experimental results were also used to verify the reciprocal mode of operation suggested by the dual actuation-sensing model. Finally, scaling experiments were used to verify model predictions with respect to changes in transducer length and width.

3. Semi-passive damping

Piezoelectrics have the ability to efficiently transform mechanical energy to electrical energy and vice versa. It is this dual transformation ability which makes them useful as structural dampers. Hagood and von Flotow [23] presented a passive damping mechanism for structural systems in which piezoelectric materials are bonded to the structure of interest. Their work is based on the papers of Forward [24] and Edwards and Miyakawa [25] who first presented this type of passive piezoelectric damping for applications on resonant structures. According to the “shunted piezoelectric” approach, the electrodes of the piezoelectric are shunted with an electrical impedance. Electricity generated in the piezoelectric as the host structure deforms, is dissipated as heat in the resistive part of the shunted electrical network. The shunted piezoelectric principle is depicted in Fig. 3. As shown, the piezoceramics are connected to the resistors $R_s$.

Hagood and von Flotow [23] established the analytical foundation for general systems with shunted piezoelectrics. Their work characterises the electromechanical interactions between a structure and the attached piezoceramic network, and offers some experimental verification. Davis and Lesieutre [26] extends previous studies by using the modal strain energy approach to predict the structural damping produced by a network of resistively shunted piezoceramic elements. Using this approach, the amount of added damping per mode caused by an individual ceramic element can be computed. It was also demonstrated that increased damping could be achieved in several modes simultaneously via proper placement of the piezoceramics. Fig. 5 demonstrates the effectiveness of shunted piezoelectricity for three different resistance values. For the case in which the PZT element is shunted using an optimal value, the acceleration is decreased by 5.9 dB [2].

Fein and Gaul [28,29] further developed the shunted piezoelectric approach by replacing the shunt resistors with digital potentiometers and adding sensors to the structure. Based on information about excitation frequency, a feedback controller was used to switch the resistance of the digital potentiometers. This technique is more flexible than previous techniques since optimal damping of different modes is possible. Fig. 4 depicts the experimental setup for this shunted piezoelectric approach.

A structural vibration control concept using piezoelectric materials shunted with real-time adaptable electrical networks has also been investigated by Wang et al. [30]. Instead of using variable resistance only, they implemented variable resistance and inductance in an external RL circuit as control inputs. They created an energy-based parametric control scheme to reduce the total system energy while minimising the energy flowing into the main structure. Furthermore, they proved stability of the closed-loop system and examined the performance of the control method on an instrumented beam. The experiment demonstrated effective suppression of the total system energy and vibration amplitudes (Fig. 5).

![Fig. 3. Passive structural damping using resistively shunted PZTs [27,28].](image-url)
The shunted piezoelectric technique has been applied to improve the performance of sports equipment, including tennis rackets, skis, and snowboards. In the case of tennis rackets, piezoelectric fibres built into the frame have been used to enhance the structural damping. This results in a preciser serve and a decreased risk for injuries such as tendonitis.

4. Energy harvesting

Energy harvesting research has been driven by the need for remote electrical power supplies for applications ranging from structural health monitoring to walking-powered electronics. Portable systems which make use of power harvesting techniques do not have to depend on traditional methods for providing power, such as the battery, which has a limited operating life. The general idea underlying energy harvesting research is the extraction of electrical energy from the operating environment. Potential sources of energy include solar, thermal, mechanical, chemical, or some combination thereof. For example, piezoelectric materials can be used to transform ambient motion into electrical energy that may be stored and used to power other devices. Recent studies, experiments, and patents indicate the feasibility of using piezoelectric devices as power sources. Umeda et al. [31] use a free falling ball to impact a plate which has a PZT wafer attached to its underside. An equivalent circuit model was developed to predict the electric energy which is generated by the mechanical impact. They also investigated the energy storage characteristics for the electrical system composed of the
PZT, a bridge rectifier, and a capacitor. Starner [32] examines the energy available from leg motion of a human being and surveys other human motion sources of mechanical energy including blood pressure. Kymissis et al. [33] examine the use of piezoelectric materials to transform the mechanical energy provided by walking into electrical energy for powering a light bulb in a shoe. Kimura’s patent [34] involves harnessing the vibration energy of a small plate. The work was motivated by the need for an energy source to power a small transmitter fixed to migratory birds. The energy-harvesting approach is also compared to existing battery technologies. Goldfarb and Jones [35] presented a linearized model of a PZT stack and analysed its efficiency as a power generation device. It was shown that the maximum efficiency occurs in a low-frequency region which is much lower than the structural resonance of the stack. It was also found that the efficiency depends on the amplitude of the input force, a fact which is attributable to hysteresis in the piezoelectric material. Clark and Ramsay [36] investigate the performance of piezoelectric generators whose force inputs are parallel and transverse to the poling direction. Their work showed that the $d_{31}$ mode has a mechanical advantage in converting applied pressure to working stress for power generation. They concluded that a $1 \text{cm}^2$ piezoceramic wafer can power a MEMS device in the microwatt range. Elvin et al. [37] theoretically and experimentally investigate the use of self-powered strain energy sensors using PVDF. The sensor was combined with a wireless communication device for human bone strain monitoring. Kasyap et al. [38] formulated a lumped element circuit model to represent the dynamic behaviour of PZT in multiple energy domains. Their model has been experimentally verified using a 1-D beam structure. The beam structure demonstrated peak power efficiencies of approximately 20%. Gonzalez et al. [39] consider several methods for increasing the electrical output power from piezoelectric generators to theoretical levels. Sodano et al. [40], consider methods for storing the electrical energy generated by piezoelectric devices. Their research is motivated by the fact that the power generated by PZT is far smaller than that required for the normal operation of most electronics in real-field applications. Additionally, the time required by a PZT generator to charge a power storage device is too long for certain applications. In their study, the energy produced by a PZT generator was stored in two different ways: using a capacitor, as in previous studies, or using a nickel metal hydride battery. The battery charging method was found not only to increase the level of output power, but it also allowed electrical energy to be stored for a longer period of time.

Lesieutre et al. [41] describe an approach in which electrical energy is harvested from a mechanically loaded piezoelectric structure, and simulataneously, structural damping is introduced. The harvesting system consists of a full bridge rectifier with a filter capacitor, a switching d.c.–d.c. step-down converter, and a battery. This system has two modes of operation: At low excitation levels, the rectifier charges the battery directly, and at higher excitation levels, the d.c.–d.c. converter is placed between the rectifier and the battery. At higher levels of excitation, the d.c.–d.c. converter delivers more than four times the power to storage as compared to directly charging the battery from the rectifier. Under harmonic forcing conditions, the effective modal loss factor depends on the piezoelectric system coupling coefficient and the ratio of the operating rectifier output voltage to its maximum open-circuit value. The loss factor is comparable to that achieved using resistive shunting (see Section 3), but it does not have the corresponding strong frequency dependence.

5. Semi-active concepts

The concept of semi-active damping was formally proposed by Karnopp et al. [42]. The concept involves the use of control theory to augment the damping properties of a passive element in real time. Sometimes referred to as active–passive damping, the technique offers considerable advantages in performance over passive damping elements, and with only a slight increase in system cost/complexity. On the other hand, semi-active damping cannot deliver the level of performance of a fully active system. However, semi-active damping requires much less energy than active control (since one is only changing a passive damping level), and it can usually be implemented with less weight and cost. As energy can only be dissipated, spillover phenomena are avoided.

There are several means of realising semi-active damping, see for example those suggested by Karnopp [43]. The most common implementation is the viscous dashpot with a variable (controlled) orifice. This technique has been explored extensively in the field of semi-active automotive suspension [44–46]. Moreover, the use of semi-active damping in flexible structure control was first studied by Karnopp and Allen [47] and later also by
Davis et al. [48]. Another way of achieving semi-active damping is through the use of electrorheological fluids whose viscosity can be controlled through application of an electric field. This technology has been applied in semi-active suspension and flexible structure control [49,50]. Semi-active damping techniques have also been implemented for impact damping [51].

The concept of semi-active friction damping has been studied for use in flexible structure control and semi-active automotive suspension systems. The concept uses control theory to vary the normal force, and thus the friction force, in response to sensory feedback. Semi-active friction damping was first considered by Anderson and Ferri [52]. The concept was also mentioned by Karnopp [43] as one of several ways of developing semi-active damping forces. Karnopp’s idea was based on an antilock braking system (ABS). In the case of ABS braking, it is important to avoid sticking surfaces at a frictional interface, and in the case of semi-active damping, it is important to dissipate energy as quickly as possible. While the two objectives are related, especially since a sticking interface cannot dissipate energy, the control systems that maximise these two objectives are different.

5.1. Control of semi-active viscoelastic dampers

Several approaches have been taken to derive control laws which maximise the energy dissipated by a damper. One approach involves deriving a Lyapunov function representing system energy. By inspection of its time derivative, one may arrive at a bang–bang control law that maximises the damping contribution of the controlled term. This approach was applied to electrorheological fluid dampers by McClamroch et al. [49]. Although unstated, the fluid friction model employed in that study was equivalent to Coulomb plus viscous friction.

Sliding mode control of an electrorheological fluid damper has also been investigated by several authors [50,53]. In this study, the fluid is modelled with a viscous component related linearly to the applied electric field and a Coulomb component related quadratically to the electric field. A first-order sliding surface is defined for each damper, and a bang–bang control law is developed which maximises the rate at which each damper approaches its sliding surface. Since the only point on the sliding surface with zero velocity is the origin, the damper appears unlikely to stick. The developed controller outperforms both a system whose modes are critically damped and a system in which all electric fields are set to their maximum.

Vaculin et al. [54] propose a model of a magnetorheological damper suitable for the simulation of vehicle dynamics. The model consists of three submodels, namely the mechanical model itself, the dynamic model of the driving circuit, and the dynamic model of the magnetorheological fluid. Model parameters were identified using measurements performed on a hydraulic test rig. Additionally, the semi-active damper model was implemented in a 3-D dynamic model for a test vehicle. Agreement between the measured response of the system and the predicted response was good. The vibration isolation of the driver’s seat by means of a controllable magnetorheological damper is investigated by Sika and Valasek [55]. First they modelled the plant and the damper, and then they formulated a performance index for vibration isolation. Their control synthesis is done using the multi objective parameter optimization (MOPO) approach, whereby the unknown control law parameters are determined using optimisation of the performance index. Their results demonstrate the enhanced performance of semi-active vibration isolation over optimal passive vibration isolation for different excitations. Preumont [56] investigated semi-active sky–hook–dampers based on magnetorheological fluids for the cases of narrowband and broadband disturbances. He concluded that the semi-active controlled system behaves significantly better for high frequency, narrowband disturbances than for broadband disturbances. Additionally, it was found that the semi-active controlled system did not behave significantly better than for the case in which a magnetorheological device was driven with constant current. Based on these results, De Man et al. [57] developed a control law for the semi-active suspension of a fork lift. For narrowband disturbances, they use a semi-active controller, and for broadband disturbances, they apply constant current to the magnetorheological damper (the element then becomes completely passive).

5.2. Control of semi-active joint connections

Structural joints are a primary source of energy dissipation as compared to material damping if no additional measures are used, such as damping coating. Gaul and Bohlen [58] have shown by investigating
isolated lap-joints as well as assembled rods and frames, that a characteristic equivalent modal damping ratio, which stems from the material damping influence only (homogeneous aluminium rod) is raised by a factor of 10 in the presence of one joint connecting two rods or beams, and raised by a factor of 20 in the presence of two joints. It is most logical, therefore, to actively influence the damping in a structure at the joints. This is achieved by controlling the joint clamping forces and hence the relative interfacial slip.

Onoda et al. [59] suggest varying the stiffness of a specific structural element in order to influence the structure’s behaviour. Using this method, they achieved higher damping in orbital truss structures. Onoda and Minesugi [60] further develop this variable stiffness technique by placing Coulomb elements having adaptable friction force in parallel with the spring elements. In that study, the friction force could only be adapted at discrete time steps, with each step being greater than the structure’s lowest eigenfrequency. The authors conclude that continuous adjustment of the friction force would produce better results.

Holnicki-Szulc et al. [61] focus on active adaptation of energy-absorbing structures. The structures are equipped with sensors which detect impact in advance and controllable semi-active dissipaters. They suggested four different fields of application, namely (i) structures exposed to the risk of extreme blast, (ii) light, thin-walled tanks with high impact protection (iii) vehicles with high crashworthiness [62], and (iv) protective barriers. These systems are based on aluminium and/or steel honeycomb packages and are characterised by a high ratio of specific energy absorption.

In 2000, Gaul [63] patented a semi-active friction joint (Fig. 6). Application of a voltage to the piezoelectric stack results in controllable normal force at the friction interface [64]. For modelling the non-linear behaviour in the friction joint, one possibility is the so-called LuGre model [65] which was reviewed among others in [66]. This model is capable of predicting relevant friction phenomena [67], such as presliding displacement, stick–slip motion, and the Strubeck effect. The model describes the friction interface as a contact between bristles (Fig. 7). The internal state variable $\varphi$ represents the average deflection of the bristles, and it is governed by a first-order differential equation.

This model was designed to reproduce all observed friction phenomena over a wide range of operating conditions. It is given by

$$F_f = (\sigma_0 \varphi + \sigma_1 \dot{\varphi} + \sigma_2 v) F_N = \mu(\varphi, \dot{\varphi}, v) F_N,$$

$$\dot{\varphi} = v - \frac{\varphi}{F_c + F_A} \exp\left(-\frac{v}{v_s}\right) \varphi, \quad \varphi(0) = \varphi_0,$$

where $F_f$ is the friction force, $v$ is the relative sliding velocity at the friction interface, and $F_N$ is the normal force. The internal friction state $\varphi$ represents the average deflection of the bristles which simulate the rough surface asperity contact (Fig. 7). The parameter $F_c$ is the Coulomb friction level, and the sum $F_c + F_A$ corresponds to the stiction force. The so-called Strubeck velocity is $v_s$ [67]. The stiffness of the bristles is described by $\sigma_0$, and the two parameters $\sigma_1, \sigma_2$ describe the dynamic dependence of friction on velocity. The function $\mu$, which is defined in Eq. (2), can be interpreted as a state-dependent friction coefficient.

![Fig. 6. Semi-active joint][63,68]
Nitsche [68] and Gaul and Nitsche [69,70], presented a non-linear feedback design method based on Lyapunov techniques for linear mechanical systems with non-linear semi-active joint connections. The feedback maximises the energy dissipation in an instantaneous and local sense. Since the control law requires the knowledge of an internal variable from the LuGre friction model, they considered the design of appropriate observers. Specifically, they considered an operating point observer and a Kalman filter. Moreover, they presented an effective disturbance estimation method for improving estimation accuracy when considerable disturbances are present. The proposed feedback and observer design methods are suitable for use with a computer, and the method is applied to a flexible two-beam system containing a semi-active joint.

Another approach for controlling a semi-active friction joint employs LQR theory [71]. The cost function is an infinite time integral of a weighted sum of system energy and control effort. Ferri and co-workers [71] have compared controllers in which the input is constrained during and after optimisation. In the latter case, the controller may call for negative normal forces. An ad hoc modification of this controller which requires $F_N \geq 0$, where $F_N$ is the normal force in the friction interface, is called clipped LQR control. Both the optimal and clipped controllers are shown by simulation to perform favourably in comparison to a control law in which $F_N = k|v|$, where $v$ is the relative sliding velocity at the damper, and $k > 0$ is a constant. The velocity proportional controller does however prevent the damper from sticking and thereby ensures energy dissipation (a sticking damper does not dissipate energy).

Albrecht [72], Gaul et al. [73,74], and Wirnitzer [75] suggested a method for optimal placement of semi-active joints for vibration suppression of large lightweight structures (Fig. 8). At optimal locations, they replaced conventional rigid connections of a large truss structure by semi-active friction joints. They implemented two different concepts for the control of the normal forces in the friction interfaces. In the first approach, each semi-active joint has its own local feedback controller (SISO control), whereas the second concept uses a global, clipped-optimal controller (cLQG control). Simulation results for a 10-bay truss structure demonstrate the potential of the proposed semi-active concept. A model of the truss structure is shown in Fig. 9. The response of the system for three control strategies are compared in Fig. 10. In the figure, LQG denotes fully active control. The fully active system performs only slightly better than the semi-active cLQG controller. However, the semi-active approach requires only a fraction of the control power required by the active control approach.

### 6. Active vibration control

One of the earliest studies of active vibration control was completed by Swigert and Forward [76]. They conducted a theoretical and experimental study that involved electronic dampers. In that study, a system of electromechanical transducers made from lead zirconate titanate (PZT) were implemented to control the mechanical vibration of an end-supported mast. The output signals from the sensors were amplified and appropriately shifted in time to provide control inputs for actuators positioned symmetrically on the surface of the mast. Bailey and Hubbard [77] developed the first smart structure using PVDF. The PVDF was used as an active element for active vibration control of a cantilever beam. By implementing both constant gain and constant amplitude controllers, they experimentally demonstrated that the PVDF actuator could significantly
increase the measured loss factor (i.e. the system damping) when the structure was subjected to an initial displacement. In a later study, Fuller et al. [4] described a systematic approach for active control of vibration. They summarised the principles underlying active vibration control and its practical applications by combining material from vibrations, mechanics, signal processing, and control theory. Today, two main approaches exist in vibration control: feedforward and feedback control.

The feedforward approach makes use of adaptive filtering methods such as X-filtered LMS algorithms [4]. The main advantages of these control systems are that no model of the structure is needed and that they can be
employed at high frequencies. The major drawback of the feedforward method is that a reference signal is required, which is somehow correlated with the disturbance.

Feedback methods can further be divided into two parts: active damping and model-based control techniques. In the active damping approach, sensors and actuators are located at the same position. Feedback control is guaranteed to be stable if ideal sensors and actuators are used. The active damping method has the advantage that it does not require a model of the structure. However, it has the major drawback that it works only near structural resonances [78–81].

There exists a large variety of model-based feedback methods, including LQR, $H_{\infty}$, and modal feedback methods. Modal feedback control has been successfully implemented for the reduction of plate vibrations [82]. The modal parameters for plates can be determined using analytical solutions to the governing plate equation or by using FE calculations. For structures with more complex geometry, such as a car body, an analytical solution does not exist, and even a FE calculation is complicated and time intensive. As an alternative, Stöbener and Gaul [83] used an experimental modal analysis to extract modal parameters (eigenfrequencies, mode shapes) from measurements made on a car body. The sensors and actuators are laid out based on information about the experimental mode shapes. After the optimal actuator and sensor positions and dimensions were determined, the modal input and output matrices for a modal state-space controller were computed. The proposed modal concept was implemented on the centre panel within the passenger compartment of a roadster car body, as shown in Fig. 11. Fig 12 depicts the experimental setup for the controlled car body.

Fig. 13 shows the sensor and actuator layout on the front side of the centre and floor panel. Fig. 14 shows the four measured FRFs for the controlled/uncontrolled centre panel of the car body. As a result of the implemented modal controller, the peaks at the resonances (196, 281, 457 and 500 Hz) of the controlled modes are significantly reduced. Additionally, the vibration amplitudes of other modes, which are not explicitly included in the control concept, are also decreased. This can be explained by the shape similarity of the controlled and uncontrolled mode shapes. Nevertheless, not all vibration amplitudes are equally reduced for a particular actuator layout. Only those modes for which a sensor/actuator layout has been tailored, will be effectively controlled. As a consequence, errors in the experimentally evaluated mode shapes have a significant influence on the effectiveness of the implemented control concept.

7. Active noise control/active structural-acoustic control

The concept of active noise control is not new. Lueg [85] received a patent for a system implementing active control of sound in a duct. The sound field is first detected using a microphone. The microphone signal is then used to produce a cancelling wave which is emitted from a loudspeaker in the downstream duct. Superposition of the two waves results in destructive interference at a reference location. In 1953, Olson and May [86]
developed a different active noise control system. In this system, sound is detected with a microphone, and the signal is fed through a controller to a loudspeaker located near the microphone. Good local sound reduction at the microphone over a range of frequencies from 20 to 300 Hz was demonstrated.

The classic studies by Lueg and Olson illustrate two distinct control approaches used in active noise control. Lueg’s approach is a feedforward approach, since a priori knowledge about the disturbance is obtained using an upstream microphone. Olson and May’s approach is a feedback approach since the detection microphone is located close to the active source.

The feedforward approach for active noise control was first formally introduced by Connover [87], who studied the active control of sound radiated from large electrical transformers. The noise radiated from large electrical transformers is dominated by sinusoidal tones which are even multiples of the line frequency and which can be correlated with the electrical line signal. Connover proposed that a reference signal, formed from the line signal, could be used as a control input rather than a detection microphone. The reference signal could then be passed through an electronic controller to the control loudspeakers. Connover also introduced the concept of an error sensor with which the radiated sound field from the transformer was monitored. The signal from the error sensor was used to adjust the controller so as to minimise the radiated sound.
Although the potential of active noise control had been demonstrated in these early studies, the practical implementation of multichannel systems only became feasible with advances in digital signal processors in the 1980s. However, the noise control systems still relied on loudspeakers for actuation. The noise control concepts were successfully implemented in a wide variety of applications, including control of cabin noise in an aircraft [88–90] and control of road noise in cars [91]. It was later shown that actuators directly coupled to the structure in coupled structure–acoustic problems yield a higher noise reduction [92,93]. This approach is termed active structural acoustic control (ASAC). Over the last several years, various control techniques have been established in the field of active noise control and ASAC: adaptive filter techniques [94–97], robust control techniques [98,99], and modal control techniques [100,101].

Hagedorn and von Wagner [102] developed smart pads containing piezoelements to be used as actuators and sensors in passenger car disk brakes. The goal of this research is to superimpose ultrasonic waves to the brake in order to fine-tune the stick coefficient and hence the vibration behaviour of the brake parts. This leads to a reduction of brake squeal.

8. Active vibration isolation

Isolating a piece of delicate equipment from the vibration of a base structure is of practical importance in a number of engineering fields. Passive anti-vibration mounts are widely used to support the equipment and to protect it from severe base vibration. Although conventional passive mounts offer good isolation at high frequencies, they suffer from vibration amplification at the mounted resonance frequency. Generally, the best
isolation performance is achieved by using an active system in combination with a passive mount, whereby the fundamental mounted resonance can be actively controlled without compromising the high-frequency performance.

A good overview of active vibration isolation techniques can be found in [4]. Various control strategies are discussed, including feedforward and feedback concepts for systems under periodic as well as random vibrations.

Stöbener and Gaul [103] modelled a piezoelectric stack actuator with FE s and investigated the response of a one-degree-of-freedom (dof) vibration isolation system having such a stack actuator built in. To validate the FE formulation and to evaluate the performance of the vibration isolation system with the stack actuator, they designed and tested an experimental setup. They examined both feedforward control and feedback control techniques to enhance the isolation effect. Huang et al. [104] presented a theoretical and experimental investigation of an active vibration isolation system. In that study, decentralised velocity feedback control was employed, whereby each electrodynamic actuator is operated independently by feeding back the absolute equipment velocity at the same location. They obtained good control and robust stability both experimentally and theoretically for the multichannel control systems.

Preumont et al. [105,106] compared the force feedback and acceleration feedback implementation of a sky hook damper used to isolate a flexible structure from a disturbance source. They showed that the use of a force sensor always produces alternating poles and zeros in the open-loop transfer function between the force actuator and the force sensor, thus guaranteeing stability of the closed loop. On the contrary, the acceleration feedback produces alternating poles and zeros only when the flexible structure is stiff compared to the isolation system.

Riebe and Ulbrich [107] presented the model of a parallel robot with 6 dof for the use in real-time computation of the inverse dynamic. They modelled the frictional behaviour, and the parameters describing the friction model are identified and optimised. Furthermore, they presented a comparison between the measured and the simulated actuation forces.

Müller et al. [108] and Beadle et al. [109] investigate a four mount, 6 dof for active vibration isolation and suppression. The system itself is based on a decentralised analog velocity feedback controller. Fig. 15 depicts the active vibration isolation system which was investigated. The governing equations of motion are obtained by using the balance of linear and angular momentum. The parameters necessary to describe the system’s behaviour were experimentally determined by comparing measured transfer functions to those calculated using an updated model. Finally, two different active control strategies, specifically SISO (single input, single output) and MIMO (multiple input, multiple output) control schemes, are considered. The predicted transmissibility curves for the uncontrolled and MIMO-controlled vibration isolation system are depicted in Fig. 16. Transmissibility of the receiver isolation is defined here as the displacement amplitude ratio of the receiving upper surface of the vibration isolation system divided by the base structure source amplitude.

9. Structural health monitoring (SHM)

Nearly all in-service structures require some form of maintenance for monitoring their integrity and health condition. Appropriate maintenance prolongs the lifespan of a structure and can be used to prevent
catastrophic failure. Current schedule-driven inspection and maintenance techniques can be time consuming,
labour-intensive, and expensive. SHM on the other hand involves autonomous, in-service inspection
of a structure. The first instances of SHM date back to the late 1970s and early 1980s. The aerospace
community used SHM techniques in conjunction with the development of the space shuttle, and the civil
engineering community applied SHM techniques to bridges. SHM consists of both passive and active sensing
monitoring. Passive sensing monitoring is used to identify the location and force–time–history of external
sources, such as impacts or acoustic emissions. Active sensing monitoring is used to localise and determine the
magnitude of an existing damage. An extensive literature review of damage identification and health
monitoring of structural and mechanical systems from changes in their vibration characteristics is given by
Doebling et al. [111].

9.1. Passive sensing diagnostics

For a passive sensing system, only sensors are installed on a structure. Sensor measurements are constantly
taken in real time while the structure is in service, and this data is compared with a set of reference (healthy)
data. The sensor-based system estimates the condition of a structure based on the data comparison. The
system requires either a data base, which has a history of prestored data, or a structural simulator which could
generate the required reference data.

The input energy (external loads, temperature, pressure, etc.) to a structure is typically random and
unknown. Passive sensing diagnostics are primarily used to determine unknown inputs from changes in sensor
measurements [112]. Choi and Chang [113] suggested an impact load identification technique using
piezoelectric sensors. They used a structural model and a response comparator for solving the inverse
problem. The structural model characterises the relation between the input load and the sensor output. The
response comparator compares the measured sensor signals with the predicted model. An extension of this
work is given in Tracy and Chang [114]. Their work is not only applied to beams, but also to composite plates.
They developed a computer code which automatically identifies the impact load and location.

Only a few studies deal with dispersive waves in structures in conjunction with time–frequency analysis.
Kishimoto et al. [115] developed a tool for determining the impact location of a beam using the wavelet
transform (WT). Here, they restricted their analysis only to beams. Inoue et al. [116] experimentally validated
the method suggested by Kishimoto et al. [115]. In that study, a simply supported beam was impacted with a
sphere, and strain gauges were used to measure the strain caused by the flexural waves.

The experimental setup of a representative passive sensing diagnostic system is depicted in Fig. 17. The
system consists of a freely suspended steel plate which is impacted by a pendulum. Four piezoelectric film
sensors (PVDF) have been attached at the plate corners. The force–time–history at the impact location is
directly measured using a laser vibrometer and is compared to the force–time–history reconstructed from the
PVDF sensor signals. Specifically, the vibrometer is used to measure the velocity, and therefore the velocity-
proportional impact force, at the impact site. The advantage of the piezoelectric film is that this sensor has the
property of multidirectional sensitivity as opposed to strain gages, and they do not require a bridge circuit since the output voltage is proportional to strain. A laser Doppler vibrometer (LDV), together with the controller, is used to compare the determined force–time–history with the “actual” one. The oscilloscope used to record the signals is triggered by one of the sensor signals, and the data is sampled at 1 \( \mu \text{s} \). It is obvious that a measured strain history signal contains some undesired contributions from reflections. A simple way of smoothing digital data is to use moving average techniques. Fig. 18a shows an example of a signal from sensor 1 which has been smoothed to remove unwanted contributions from reflections. The corresponding signals from the other sensors look similar. The signal-processing procedure can proceed once the experimental data is saved on a PC. It is important to window the signals before applying the Fourier transform. The signals are then transformed into the frequency domain using the fast Fourier transform (FFT) algorithm. Then the WT is applied using the Gabor wavelet\[117,118\]. Fig. 18b illustrates the 3-D magnitude plot of the wavelet transform of the signal from sensor 1. Fig. 18c is a contour plot of this 3-D magnitude plot. The maximum value of the wavelet transform was used to indicate the arrival time (Fig. 18d) of the flexural wave at a given PVDF sensor. The impact location and the group velocity of the flexural waves were then determined from these arrival times. Based on the measured strain–history and impact location, the force–time–history at the impact location was reconstructed\[118,119\]. Fig. 19 shows the force–time–history measured with the vibrometer and the force–time–history reconstructed from the PVDF sensor signals. The same localisation procedure has also applied for acoustic emission signals\[120\].

9.2. Active sensing diagnostics

Another important topic in SHM is damage detection. Boller and Biemans\[121\] gave an overview of structural health monitoring on aircraft. They found that the size of a delamination must be at least 10% of the component’s surface if it is to be reliably detected using vibration-based damage detection. Local or wave-propagation-based SHM is therefore advantageous since much smaller defects can be detected. Chang\[112,122\] concentrates his research on wave-propagation-based SHM. He developed Lamb-wave-based techniques for impact localisation/quantification and damage detection. Wilcox et al.\[123\] examined the potential of specific Lamb modes for detection of discontinuities. They considered large, thick plate structures (e.g. oil tanks) and thin plate structures (e.g. aircraft skins). They showed that the most suitable Lamb mode is strongly dependent on what the plate is in contact with. Furthermore, they showed that the properties of the system to be inspected determine which modes can be used, and that this then dictates the type of transducer required. Lemistre and Balageas\[124\] presented a robust method for damage detection based on diffracted Lamb wave analysis by a multiresolution wavelet transform. Berger et al.\[125\] use fibre optic sensors in order to measure Lamb waves.
Fig. 18. Measured signal (a), 3-D-plot of the WT (b), contour plot of the WT (c) and arrival time (d) of sensor 1 [118].

Fig. 19. Vibrometer-measured and PVDF sensor-reconstructed force–time histories [118,119].
Benz et al. [126] and Hurlebaus et al. [120] developed an automated, non-contact method for detecting discontinuities in plates. Laser ultrasonic techniques are used to generate and detect Lamb waves in a perfect plate and in a plate that contains a discontinuity. The geometry of the examined plate is depicted in Fig. 20. The measured signals are first transformed into the time–frequency domain using a short-time Fourier transform (STFT) and subsequently into the group–velocity–frequency domain (see Fig. 21). The discontinuity is then located through the use of a correlation in the group–velocity–frequency domain. Fig. 22 shows the correlations for a perfect plate and a notched plate. A ratio of the correlations is formed to enhance any features that are present in the notched plate correlation, but not in the perfect plate correlation. As shown in Fig. 22, the correlation ratio has a single dominant peak at $d_0 = 40$ mm ($d_0$ corresponds to the round trip distance between source and receiver), which is in excellent agreement with the actual location of the discontinuity [118].

A further SHM technique involves the use of a smart layer for damage detection. The smart layer approach is particular useful for monitoring the so-called “hot spots” of a highly loaded structure. Lin and Chang [122] use a smart layer to excite and detect Lamb waves for the purpose of crack detection. The study by Hurlebaus et al. [118,127] makes use of bulk waves to detect thickness changes and potential delaminations. The smart layer presented by Lin and Chang [122] makes use of a PZT-sensing element, whereas the smart layer presented by Hurlebaus et al. [118,127] uses PVDF-sensing elements. Finally, in the study by Lin and Chang [122], PZT transducers were placed at a few discrete points on the smart layer; and in the study by Hurlebaus et al. [118,127], the PVDF polymer covers the entire surface of the smart layer.

Fig. 23 depicts the experimental setup, where a pulser-receiver is used to generate a short, high-voltage pulse for driving the PVDF material. The pulser-receiver is driven in the pulse-echo mode so that the PVDF film is used both as actuator and sensor. In order to connect the PVDF films with the pulser-receiver, a multiplexer as part of a VXI-measurement system is used. The pulse generator which is enclosed in the VXI-mainframe triggers the pulser-receiver such that it emits no high voltage when the multiplexer switches from one PVDF film to the other. The oscilloscope, which is part of the VXI system, receives the signals from the
pulser-receiver, namely, the sensor signal and the synchron signal. The VXI-system is connected to a PC via a general purpose interface bus (GPIB). The PC is used for controlling the VXI-system, for storing the data, and for further signal processing.

The smart layer developed by Hurlebaus et al. contains an embedded network of distributed piezoelectric polymers. The smart layer was used to identify an “artificial” damage in an aluminium plate (150 × 150 × 15 mm³). The artificial defect is created by milling out some bottom sections at the backside of the aluminium plate. The depth of the milled section is about 2 mm. The boundary of the milled section is shown in Fig. 24 by the solid lines. On top of the aluminium plate, a smart layer is attached using couplant...
material. The smart layer is shown in Fig. 24. The electrical connections were obtained using printed circuit techniques. The dark square regions mark a $10 \times 10$ array of PVDF transducers. The side length of each PVDF element is 10 mm, and the distance between sensors is 2 mm. The transducers are used both as senders and receivers. Fig. 24 shows a C-scan of the aluminium plate. The identified defect is much larger than the actual defect. However, this is a consequence of the size and quantity of PVDF transducers which are used. If one would use a large array of smaller PVDF transducers, the resolution of the identified defect would be finer and smoother.

9.3. Self-healing–self-repairing

Peairs et al. [128] presented a method for the self-healing of bolted joints. The loosening of a bolted joint connection is a common structural failure mode. They reported a real-time condition monitoring and active control methodology for bolted joints in civil structures and components. They used an impedance-based health-monitoring technique which utilises the electromechanical coupling property of piezoelectric materials to identify and detect bolt connection damage. When damage occurs, temporary adjustments of the bolt tension can be achieved actively and remotely using shape memory alloy actuators. Specifically, when a bolt connection becomes loose, the bolted members can move relative to each other. The subsequent frictional heating causes a Nitinol washer to expand axially, thereby leading to a tighter, self-healed bolt connection.

Pang and Bond [129–131] developed a novel composite system which employs a biomimetic approach to perform a self-repairing function. Such a system can perform two functions: the visual enhancement of impact damage by the bleeding action of a highly conspicuous medium such as fluorescent dye; and, the restoration of mechanical properties by a healing agent, which is stored within hollow fibres, and which infiltrates and patches the damaged area upon activation. Impact indentation followed by four-point bend flexural testing
was conducted to evaluate the strength restoration after self-repair. The results of mechanical testing have shown that a significant fraction of strength is restored by the self-repairing effect.

10. Conclusions

This paper addresses several fields of application of smart structure dynamics technologies. After discussing a variety of multifunctional materials, some examples for the application of these technologies were given. First, semi-passive concepts used to enhance the damping behaviour of structures were summarised. Then energy-harvesting technologies are discussed, and their combination with the shunted piezoelectric concept is highlighted. Semi-active concepts are also considered, particularly in the context of control of large lightweight space structures. Next, active concepts are discussed. In addition to active vibration control and active noise control, this paper deals also with the field of active vibration isolation and suppression. Finally, the implementation of smart structure technologies for in-service monitoring is addressed. Three concepts of SHM are discussed in more detail: passive sensing diagnostics, active sensing diagnostics, and self-healing structures. Various smart structure technologies have been developed; the growing use of these technologies in many potential application areas can be expected: automotive engineering (e.g. cars, trucks), aerospace engineering (e.g. space shuttle, airplanes), and civil engineering structures (e.g. bridges, tunnels). Avoiding failure of smart structures during structural life is an important prerequisite and a future challenge.

Acknowledgements

This review article is based in part upon responses to a questionnaire directed at leading scientists in the field of smart structures. The authors gratefully wish to acknowledge the helpful input received from these conscientious contributors.

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