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October 22-23, 2009 • Montreal, Quebec, Canada

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Proceedings of Cansmart 2009 – 12th Cansmart Meeting - International Workshop on Smart Materials and Structures


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PREFACE

Researchers in the field of smart materials and structures have made tremendous strides during the past two decades. Throughout this period of progress, the Canadian Smart Materials and Structures Group (Cansmart Group) has been a pioneer for reporting on results in this field of research and development in Canada. Established in 1997, the Cansmart Group provides a platform and the appropriate conditions for future growth and integration of the Smart technologies. The study of these technologies and their applications is a multidisciplinary field. It includes such topics as structural health monitoring, non destructive evaluation, vibration damping and control, actuators and transducers, sensing and energy harvesting, and medical applications. This interweaving of many engineering and scientific disciplines (i.e. civil, mechanical, aeronautical, aerospace, marine engineering, as well as experimental sciences, applied chemistry and physics) called for the establishment of the Cansmart Group whose main objective includes the promotion of collaboration and cooperation among the vibrant community of researchers in the fields of smart materials, intelligent applications and smart structures.

In recent years, the miniaturization of information tools has evolved to the point where their inclusion in smart materials, structures, systems and related technologies have culminated. These advances have provided a comprehensive and theoretical framework for implementing multifunctionality and smartness into new and old materials. They have also allowed the transformation of that framework into methodologies for practical design and production of sensors, actuators, intelligent devices, smart materials, active, adaptive and smart structures, nanotechnology and biomimetics.

Cansmart 2009 is the 12th in a series of annual workshops. It is designed to provide the Canadian and international community with a forum to explore the potential of possible collaborations on projects of mutual interest in this field. This volume of proceedings contains a collection of papers describing the latest developments in smart material and structure technologies. The material and information contained in the papers are published exactly as provided by the authors, and any statement, opinion and view expressed are their sole responsibility. Mention of trade names or commercial products does not constitute endorsement or recommendation for use.

On behalf of the Cansmart Group, I extend our hearty thanks to the keynote speaker and to all the authors for their cooperation and support in helping us to bring out the proceedings on time. Finally the executive committee and I wish to thank the members of the Advisory and National and International Scientific Committees for their help and advice.

For assistance in helping to organize this meeting, I am very thankful to the Cansmart Board for their planning assistance, to Mrs. A. Rogers, Dr. P. Weetman and Mr. G. Milburn from the RMC Centre for Smart Materials and Structures for administrative support and to Mr. G. Locklin for the graphics. The authorization to hold this workshop at the Industrial Material Institute of the National Research Council (IMI-NRC) is greatly appreciated, and for making it possible we owe particular thanks to Dr. Cheng-Kuei Jen. Finally, the Cansmart Group acknowledges the Royal Military College of Canada for its critical and unfailing support in the organisation of the workshop.

George Akhras, P.Eng, PhD
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Director, RMC Centre for Smart Materials and Structures
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PROSPECTING SECURITY AND PROSPERITY IMPLICATIONS OF CONVERGENT TECHNOLOGIES & MATERIALS 2010-2025

Jack Smith
Federal S & T Initiatives, Defence R & D Canada

ABSTRACT

As nanotechnology begins to move from the domains of R&D into applications that advance, address and resolve key societal challenges such as lower carbon energy, countering climate disruptions, enabling healthier environments and people, a key aspect of the development potential will be the identification of priority applications and target sectors for innovation with public funds and private-public ingenuity.

The presentation on PACT - Prospective Applications for Converging Technologies shows how S&T foresight has been used to scope out and shape an emerging policy discussion about which areas and targets will be most important for Canada within the larger context of societal value and needs. This discussion may have some useful parallels to those that other research groups investigating innovative and smart materials will need to adopt to ensure that nanotechnology not only meets the interests and opportunities of global commerce, but also for society.

This discussion is also important to have at the level of applications rather than at more abstract levels since applications will have differing public policy engagement thresholds in terms of potential issues such as perceived toxicology impacts and risks, and public concerns about personal privacy and health.

The presentation therefore combines innovative perspectives linking the technical and social dimensions of nanotechnology.
Contributed Papers
ACTIVE MATERIAL BASED ACTIVE SEALING TECHNOLOGY
PART 2: DIELECTRIC ELASTOMER AND SMA APPROACHES

C. Henry¹, W. Carter¹, G. Herrera¹, G. McKnight¹, A. Browne², N. Johnson², I. Bazzi²

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ABSTRACT

Current seals used for vehicle closures/swing panels are essentially flexible, frequently hollow structures whose designs are constrained by numerous requirements, many of them competing, including door closing effort (both air bind and seal compression), sound isolation, prevention of water leaks, and accommodation of variations in vehicle build. This paper documents the second portion of a collaborative research study/exploration of the feasibility of and approaches for using active materials with shape and stiffness changing attributes to produce active seal technologies, seals with improved performance. An important design advantage of an active material approach compared to previous active seal technologies is the distribution of active material regions throughout the seal length, which would enable continued active function even with localized failure. Included as a major focus of this study was the assessment of polymeric active materials because of their potential ease of integration into the current seal manufacturing process. In Part 1 of this study [1], potential materials were evaluated in terms of their cost, activation mechanisms, and mechanical and actuation properties, with shape memory alloys (SMA) and electroactive polymers (EAP) being judged to be the most promising. In Part 2, which is documented in this paper, a more in-depth development was made of embodiments of active seals using SMA and EAP. Specific improvements that must occur with regards to use of SMA's are in the areas of cycle speed and higher temperature activation. Among the key development risks that were identified for EAP's are operating voltage reduction, the need for expanded operating temperature range, and supplier base development. The study concluded with the concepts proposed being embodied only to a rudimentary stage because of the previously stated issues.

Keywords: seals, shape memory alloys, electroactive polymers.
INTRODUCTION

Current seals used for vehicle closures/swing panels are essentially flexible, frequently hollow structures whose designs are constrained by numerous requirements, many of them competing, including door closing effort (both air bind and seal compression), sound isolation, prevention of water leaks, and accommodation of variations in vehicle build. The collaborative research project described here had its genesis in brainstorming on ways in which the field activated shape and stiffness changing attributes of several classes of active materials could be utilized to create active seals whose geometry and stiffness could be changed on demand. The hope was (and remains), that with this capability provided through the use of active materials, a seal technology could be achieved with situationally tailorable attributes that would allow it to excel in all performance requirements.

As background, in terms of function, seals between automotive doors and openings are used for sound and/or fluid management. Increasing the pressure and/or area of the seal against the opposing surface can generally increase seal effectiveness. However, this has a negative impact on door opening and closing effort. Therefore, it is desirable to have seals that can be controlled and remotely changed to alter the seal effectiveness so that door opening and closing efforts can be minimized yet seal effectiveness maximized.

As indicated, this paper documents the second phase of a collaborative research study/exploration of the feasibility of and approaches for using active materials with shape and stiffness changing attributes to produce active seal technologies, seals with improved performance. An important design advantage of an active material approach compared to active seal technologies is the distribution of active material regions throughout the seal length, which would enable continued active function even with localized failure.

In this study we evaluated tradeoffs between various active material options. However, because of the potential ease of integration of polymeric active materials into current seal manufacturing and installation processes, a considerable portion of the effort in the present study was focused on their assessment.

As illustration, a rather generic example of an active material based active sealing scheme/approach is as follows. Prior to opening or closing the door, a trigger (such as an accelerometer or handle sensor) initiates a signal to turn on the control stimulus in or near the active material portions of the seal. The active material then causes a contraction of the top seal surface. The design geometry varies from material to material depending on the return spring geometry, mode of actuation and control stimulus. When the door is closed, a trigger sends a signal to turn off the control stimulus. The active material relaxes and a spring in parallel with the active material causes the top seal surface to return to its initial position. When the seal makes contact with the opposing surface, the seal and return spring apply a force greater than that of the seal alone. This force increase improves sealing effectiveness.

The next section outlines some of the requirements associated with active seals developed during the course of this collaborative effort between HRL and GM. The section after that outlines several important tradeoffs with the particular actuation properties of various specific active materials and sheds light on important material limitations. The section after that goes into more depth concerning dielectric elastomer materials including.
their actuation properties and their suitability for sealing, and recommends further courses of action. The final set of sections details calculations for seal geometries, including those utilizing two promising candidate active material technologies: shape memory alloys (SMA) and dielectric elastomers.

ACTIVE SEAL REQUIREMENTS

The fundamental long term goal of this collaborative effort, which goal has yet to be reached, is an active seal that is less expensive than a triple seal but with improved functionality compared to continuous double seals. The newly incorporated materials must be inexpensive, easy to manufacture into seal geometries with minimal complexity, and easy to install into vehicles on the assembly line. Important requirements that an active seal must exhibit include the following items.

Active Seal Requirements:

- Maintain cost below triple seal
  - Difference between double and triple seal including installation costs
- Simple is better
  - Keep aesthetic complexity minimal
- Maintain seal effectiveness
  - Active material provides sufficient pressures to work against seal stiffness
  - Compression and contact width meet application requirements
- Minimum actuation timescale upon door closing/opening ~100ms.
- Manufacturability
  - Should not be complicated to manufacture or install
- Fatigue, lifetime issues
  - Needs to pass current seal/door tests
  - Needs to survive over vehicle lifetime (50,000 to 200,000 cycles)
- Environmental robustness
  - Must function in wide range of temperatures & typical / common automotive fluids
- Failure mechanisms
  - Distributed embedded actuation so that local failure does not cause global failure
  - Default to status quo operation if active component fails
  - Seal function should not be compromised if punctured, abraded, or cut stretched

ACTIVE MATERIALS ACTUATOR PROPERTIES

Active materials can provide a seal with variable force-displacement characteristics over a fixed geometry seal. This can be accomplished in several ways. Through actuation, a sealing surface may be displaced or enlarged. Alternatively, by changing the seal stiffness via actuation or inherent modulus change of the active material, the sealing force between two opposing surfaces may change. Fig. 1 shows several classes of active materials that can provide the appropriate stress-strain characteristics for a sealing function shown in the red dashed box. As long as the energy density (diagonal line of constant stress * strain in Fig. 1) of the material is greater than that required by the application, the material can theoretically
provide the necessary stress and strain.

Fig. 1: Effective output stress versus strain for various active materials technologies.

Table 1 shows a sample of different active materials, reasonable bounds on their actuation stresses and strains, how they might be embedded in a seal geometry, and size, cost, time response, power requirements and additional pros and cons associated with the use of each material and its particular activation means. Mechanical advantage schemes inherently go against the important advantages of distributed actuation, elegant failure mode and ease of manufacturability in using active materials for seals. Thus, commercially available active material based point actuators that can develop a high force such as PZT and Terfenol-D would require a mechanical advantage and force distribution schemes. Additionally, these materials are brittle which precludes their acceptance into a very compliant component. Just as alternative materials have been whittled down on the basis of the suitability of their actuation stress and strain, other factors such as time response, cost and input stimulus further reduce viable active material options for seals. In reviewing Table 1, it was decided (Fig. 2) that active polymers would be easier to distribute in a seal structure, integrate into existing manufacturing and assembly processes and better match the mechanical stiffness of current seal designs. An assessment of active polymer technologies and their readiness was provided in Part 1 [1] of this study.
**Fig. 2:** Schematic of active material maturity versus seal application fit. More mature active materials such as piezoceramics (PZT) and magnetostrictives (Terfenol-D) are poorly suited for actuating seal material. In contrast, active polymers may be a better fit, but require further development to be applied to seals in an elegant fashion.

**Table 1:** Active materials choices for seals.

<table>
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<tr>
<th>Material</th>
<th>Properties</th>
<th>Geometry</th>
<th>Size, etc…</th>
<th>Pros &amp; Cons</th>
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<tr>
<td>Shape memory alloy (NiTi)</td>
<td>$\sigma \sim 200$ MPa $\varepsilon \sim 4%$ contractile</td>
<td>Off $\rightarrow$ On</td>
<td>Vol. = 0.3 cm$^3$ / m</td>
<td>Pros: Robust, multiple commercial suppliers, suitable form factor</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Dim. = 12 mil wire</td>
<td>Cons: Temperature driven, requires bias force to reset</td>
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<td></td>
<td></td>
<td></td>
<td>12V, 1A $\sim$12W</td>
<td></td>
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<tr>
<td>Dielectric Elastomer</td>
<td>$\sigma \sim 3$ MPa $\varepsilon \sim 60%$ expanding</td>
<td>Off $\rightarrow$ On</td>
<td>Vol. = 1.3 cm$^3$ / m</td>
<td>Pros: Inexpensive materials, fast response</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td>Dim. = 0.2 mm film</td>
<td>Cons: difficult to integrate, only one small commercial supplier,</td>
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<td></td>
<td>1kV, 10 mA $\sim$10W</td>
<td>requires bias strain, high voltage</td>
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<td>Piezoelectric bimorphs</td>
<td>$\sigma \sim 20$ MPa $\varepsilon \sim 10%$ Bending</td>
<td>Off $\rightarrow$ On</td>
<td>Vol. = 23 cm$^3$ / m</td>
<td>Pros: Mature, comm avail, fast response</td>
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<td></td>
<td></td>
<td></td>
<td>Dim. = 0.04 mm strip</td>
<td>Cons: Cost. Brittle, small deflection, mechanical advantage needed, high</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Power $\sim$100-</td>
<td>voltages</td>
</tr>
</tbody>
</table>
DIELECTRIC ELASTOMER ACTUATOR TECHNOLOGIES

Active polymers are thought to be more amenable to the current seal manufacturing processes than more mature active materials such as piezoelectrics, magnetostrictives, or shape memory alloys. Furthermore, the stroke required for seal deformation is considerably higher and the pressure is considerably lower than what the above materials can provide, i.e. a poor performance match. Mechanical advantage schemes for these more mature active materials add much parasitic mass and may severely reduce robustness when putting them into the high compliance path. For these reasons, a significantly more in-depth assessment was made in Part 1 [1] of this study of polymeric materials.

Of the many active polymer technologies reviewed therein, dielectric elastomers or “artificial muscles” were identified as being a promising technology because of their actuator (1-5 MPa stress, 100% strain, 10-100 Hz) properties and were chosen for more extended evaluation in this the second part of this study. As technical background, in practice, dielectric elastomer actuators consist of a sheet/slab of dielectric elastomer sandwiched between two compliant electrodes. When electric fields on the order of 100MV/m are applied between the compliant surface electrodes the electrostatic forces resulting from the free charge create a Maxwell stress, which squeezes the elastomer and stretches it outward as shown in Fig. 3a. The dielectric elastomer needs to be biased or prestrained in some fashion to prevent strain from being accommodated and the film from buckling.

![Figure 3a](image1.png)  
![Figure 3b](image2.png)

**Fig. 3**: (a) Schematic of operation for a dielectric elastomer actuator. A dielectric elastomer is sandwiched between compliant electrodes that stretch. When kilovolt level potentials are applied, the force of attraction squeezes the thickness while elongating the elastomer laterally. (b) Effect of the % volume fraction of conductive carbon black on the figure of merit $\psi/Y$

DIELECTRIC ELASTOMER AND SMA ACTIVE SEAL EMBODIMENTS

Given that the foundational work in this study had identified SMA and dielectric elastomer actuator technologies as being the leading active material candidates for enabling active seals, an important portion of Part 2 of this study was the development, at least to a
rudimentary stage of embodiments based on these two classes of active materials. This development effort, beginning with the requisite calculations for selected seal geometries, is described in the remaining sections of this paper.

CALCULATIONS FOR SELECT SEAL GEOMETRIES

Preliminary calculations are made for the force and displacement requirements of a circular seal cross section that is contracted by stimulation of an active material. For conventional active materials that contract upon activation, the material could be vertically oriented across the diameter to decrease seal height. For dielectric elastomers that expand like a rubber band with an applied voltage, the material should be horizontally oriented across the diameter to decrease seal height. Directly inserting the active material into a spring-like structure minimizes the number of springs against which the material mechanically works. If the active material is made into an actuator device with a built-in return spring and then it is inserted into the seal cross section, the active material must then work against two springs: the actuator’s internal spring and the seal cross section. This effectively permits the initial seal cross section to be stiffer, increasing overall seal effectiveness. Contracting the top seal surface inward via an embedded active material decreases door closing effort.

The inherent spring properties of a passive seal are described here. They provide a good estimate of the magnitude of the forces and displacements that a circular cross section would require. Fig. 4 shows a schematic of the sealing force versus the sealing surface displacement for a generic door seal. In Fig. 4, force is measured in Newtons per 100mm seal length and associated pressure (in psi) assuming a 3mm contact width. Displacement in Fig. 4 is a relative measure of seal compression from the nominal design point of 2psi, 4N/100mm. Thus, if the mating door surface ends up –3mm inboard from the nominal seal compression, sealing force is increased. In this case, 4psi (8N/100mm) seal pressure results in a high closing effort. Conversely, if the mating door surface ends up +3mm outboard from the nominal seal compression, the seal force is diminished.

**Fig. 4**: Schematic of door seal force vs. displacement. The desired design point is 2 psi if the mating door surface closes to a position that produces nominal seal compression.

For an active material attached to a seal cross section, the displacement capabilities must
be matched, while the force the active material can provide must exceed that of the seal cross section. Thus, the actuation stiffness should exceed that of the circular cross section. This concept is exemplified using an embedded shape memory alloy wire in the next section. The condition under which a horizontally oriented dielectric elastomer may operate is calculated also.

**EMBEDDED SMA CONCEPT**

Described here is an embedded “shoelace” design for SMA wire (Fig. 5). This design meets the stress and strain requirements (putting SMA wires into the right area of Fig. 1 to consider its use) by canting the SMA wire to increase its length within the circular cross section. This increases the actuation stroke of the SMA wire, while reducing the excess force capability of the SMA wire. Without considering force output, the “shoelace” design was predicted (calculated) to meet the displacement criterion of >3mm displacement. During these initial calculations the actuation load was thought to greatly exceed the required force, which was indeed found to be the case after measurement. Displacement driven calculations estimated a “shoelace” distance of 8cm, to cause a 30cm long section of seal to be deflected 0.5cm vertically. During actuation demonstrations, the stiffness of the SMA wire remained sufficiently high to permit 13.1 N/100mm seal pressure with a 3 mm contact width.

![Fig. 5: Embedded “shoelace” design with SMA wire and reinforcement bar on the top and bottom surfaces. It meets the actuation requirements as described above.](image)

While the bottom rod can be replaced with the existing steel insert, the top rod provides force distribution along the top surface and is an electrode for resistively heating the wire segments. This section consumed about 5W and had a cycle time of several seconds. With stitching techniques, this concept could be made easy to manufacture. The real drawback of this design is the upper rod, which needs to be stiff in the actuation direction. Conventional steel rods used in this research cannot bend or conform to a door opening. An alternative design is to have bicycle chain-like rod, which is rigid in one direction but flexible in the other to permit contouring of the seal. Also, the rod could be fixed below the upper surface by additional seal material. This would permit the seal to flex at the surface and still strain under actuation. In this alternative, greater “shoelace” distance would be needed to maintain 6mm deflection as shown in Fig. 6. Thus, there are variations on this design which could make this an inexpensive, distributed, robust, and low power concept.
Fig. 6: Alternative SMA shoelace design with a force distributing rod in the center (versus the top) of the seal. This enables greater compliance at the top of the seal surface, while still permitting 6mm deflection at 4psi.

DIELECTRIC ELASTOMER CONCEPT

Described here is an embedded concept for incorporating dielectric elastomers in a circular seal cross section. Fig. 7a shows schematically how such a seal would operate. On the right in Fig. 7a, the elongated oval is the equilibrium seal shape after prestraining the dielectric elastomer without applied voltage. On the left, voltage is applied to the dielectric elastomer causing it to stretch, flattening the seal.

Fig. 7b illustrates and describes the change in vertical, $\Delta D_V$, and horizontal, $\Delta D_H$, diameter of a thin-walled circular ring with diametrically opposite and equal loads, $P$. The radius of the ring is denoted by $R$, modulus, $E$, and area moment of inertia, $I$. This is Case 1 of Table 9.2 in Roark’s Formulas for Stress and Strain, 7th Ed. as shown in Fig. 7b.

$$\Delta D_V = -0.1488 \frac{PR^3}{EI}$$

$$\Delta D_H = 0.1366 \frac{PR^3}{EI}$$

Thus, for a ring 2cm in diameter, 2mm thick and a load of 5psi (8.3N/100mm), the deflections are -8.2mm vertically and 6.1mm horizontally. For a dielectric elastomer stretched across the interior diameter, its resulting strain is 51%. Equation 2 is used to determine the force in the dielectric elastomer where $LF_H$, $B_{HV}$ and $B_{HH}$ are geometry and applied load dependent reaction force coefficients, $P/2$ are the vertical reaction forces, $\delta_H$ is the horizontal displacement and $H$ is the horizontal reaction. This is Case 5a of table 9.3 from Roark 7th Ed. shown schematically in Fig. 7c.

$$\delta_H = \frac{R^3}{EI} \left[ B_{HH} H + B_{HV} \frac{P}{2} - LF_H \right]$$

(2)

If the horizontal diameter change from Equation 1 can be equated to a horizontal displacement in Equation 2, then the horizontal reaction can be solved in terms of the applied load as shown in Equation 3.

$$H = \frac{1}{B_{HH}} \left[ LF_H + \left( 0.1366 - \frac{B_{HV}}{2} \right) P \right]$$

(3)

For the above set of values, a horizontal force of 33N/100mm is required to deliver the required strains. For SRI-like silicone dielectric elastomers that have a thickness of 50μm (each exert 125kPa at 3.6kV), 9 layers are required to meet the horizontal reaction force calculated above. Thus, 9 layers of dielectric elastomer oriented horizontally in a circular
A circular seal cross section could exert 125kPa at 50% strain, create a 6.1mm deflection of the top seal surface and reduce the seal pressure by 5psi.

![Fig. 7](image)

Fig. 7: (a) Dielectric elastomer prestrained to give circular cross section a vertically elongated oval shape at equilibrium, without applied voltage. When voltage is applied, the prestrain is relaxed resulting in a flattened shape. (b) Case 1 of Table 9.2 Roark 7th Ed. that illustrates the change in horizontal and vertical deflections of a circular ring due to an applied force, P. Schematically shown is a dielectric elastomer with a horizontal orientation. (c) Case 5a of Table 9.3 Roark 7th Ed. that illustrates reaction force and horizontal deflection to be achieved by the dielectric elastomer in response to a load.

Fabrication issues and alternative designs still need to be worked out. For example, how is it possible to fabricate a continuous multilayer, attach it internally in a prestrained condition, attach and integrate electrode leads to the elastomer and the car electrical system, isolate actuation regions to make the active seal robust and failure tolerant, and use inexpensive small high voltage stepup transformers.

CONCLUSIONS

- Dielectric elastomer and SMA wire designs warrant further development because they meet the functional requirements to improve seal effectiveness.
- Dielectric elastomers are a good choice to engineer into seals because of their electrically-controlled actuation, fast response time, and relative cost compared with other active material candidates. However several key issues must be addressed to enable implementation of active polymer seals: distributed actuation, incorporation of graceful failure modes, operating voltage reduction, the need for expanded operating temperature range, and supplier base development and the immaturity of active polymer manufacturing and processing technologies.
- Specific improvements that must occur with regards to use of SMA's are in the areas of cycle speed and device integration in terms of physical connection to and distributed actuation of the polymer seal body.

REFERENCES

ADAPTIVE PENDULUM MASS DAMPER FOR THE CONTROL OF STRUCTURAL VIBRATIONS

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ABSTRACT

Incorporating adjustment mechanisms to compensate for de-tuning in traditional tuned mass dampers is essential for the advancement of this technology. In this paper, experimental and simulation results on the control of structural vibrations using an adaptive pendulum mass damper are presented. The adaptive pendulum mass damper is a spherical pendulum augmented with a tuning frame to adjust its length and two adjustable air dampers connected to the mass for achieving the damping adjustment. The mechanical adjustments are implemented using three independent stepper motors, one micro-controller and three drives. The mass damper is used to control the vibration responses of a bench-scale two-story model structure with sufficiently long fundamental period representative of flexible structures such as towers. The basic architecture of the system proposed consists of two components; identification and control, one followed by the other in that order. The identification is carried out using traditional Fourier methods and a second-order blind identification method in the time-domain. The control phase consists of position control based on the identified frequency from the identification phase. The paper focuses on the hardware and software aspects of the real-time implementation of this adaptive mass damper. The identification methods used in this study rely only on acceleration measurements collected from high accuracy-low frequency accelerometers mounted on the structure, and do not utilize the excitation information. This study is primarily intended to demonstrate the feasibility of employing adaptive pendulum mass damper designs to enhance the robustness of passive tuned mass dampers to structural, environmental and design changes.

Keywords: Adaptive Pendulum Mass Dampers, TMDs, Structural Control.
INTRODUCTION

Tuned mass dampers (TMDs) have the potential to reduce structural vibrations by adding additional damping to flexible structures. The literature on this topic is extensive [1-8] and a comprehensive review of TMDs is not attempted here. A large number of TMDs are of pendulum type, called pendulum-TMDs (PTMDs), which essentially consist of a suspended mass tuned to a fraction of the fundamental period of the structure, and a viscous damper which acts as an energy sink transferring the mechanical motion of the TMD to heat that is eventually dissipated to the environment. A major drawback of TMDs is their sensitivity to changes in their operating environment such as structural deterioration, which results in de-tuning. Failure to account for de-tuning may cause the performance of TMDs to degrade over time. In this paper, an adaptive pendulum TMD, called adaptive pendulum mass damper (APMD) henceforth, is presented to compensate for the de-tuning and other environmental effects. The main goal of this paper is to illustrate a relatively simple method of harnessing the advantages of semi-active control using pendulum TMDs, while focusing on the algorithmic and design aspects involved in achieving the optimal tuning in the proposed APMD configuration.

Semi-actively controlled TMDs have been studied by various researchers and several effective designs have been presented. For example, variable stiffness TMDs consisting of adjustable stiffness elements have been studied for seismic and wind applications [3, 4]. Pendulum mass dampers with nonlinear dampers have also been studied by researchers for structural vibration reduction and a comprehensive review of the literature on this subject has been published recently [9]. In this paper, a pendulum tuned mass damper (PTMD) is enhanced with adaptation mechanisms to adjust its length and damping through feedback. A pendulum TMD is chosen as the underlying passive system because a large number of TMDs in operation today are of this type. In this paper, the effective length is related to the identified natural frequency of the structure through simulation studies incorporating the three-dimensional behaviour of the pendulum mass. Once the effective length is known as a function of the fundamental period of the structure, the frequency tuning is implemented in two stages: first, the identification phase is carried out while the structure and TMD are in operation and in a pre-assigned configuration, followed by a control phase where a tuning frame is hoisted to the desired position to achieve the fundamental natural frequency. The level of damping is adjusted by closing or opening a valve at one end of the air-damper using a stepper motor.

The paper is organized as follows. The equations of motion for the PTMD attached to a flexible structure are presented first. Then, an experimental set-up of a new APTMD is presented followed by brief experimental results using this set-up. Finally, important conclusions of this paper are presented.

EQUATIONS OF MOTION FOR A PTMD

Fig. 1 shows the position of the auxiliary mass, $m_a$. The origin of the system is set up to coincide with the suspension point of the auxiliary mass. Critical parameters of the geometry include $\theta$, the angle of the swing of the auxiliary mass away from the vertical, $\phi$, the angle
of the auxiliary mass rotating about the vertical line, and \( L_a \) the length of the pendulum. The vectors \( u, v, \) and \( w \) are the displacements of the suspension point in the \( x-, y-, \) and \( z- \) directions, respectively. All of the aforementioned parameters vary with time with the exception of the pendulum length, which has been kept constant for this derivation. Consider also a linear spring and damper in both horizontal directions connected to the pendulum length at a distance \( h_x \) and \( h_y \) away from the suspension point.

![Diagram of the PTMD system](image)

**Fig. 1:** Formulation of the equations of motion for the PTMD system.

The position, relative to the origin, of the auxiliary mass is

\[
\mathbf{r}_a = \left[u + L_a \sin \theta \cos \varphi\right] \mathbf{i} + \left[v + L_a \sin \theta \sin \varphi\right] \mathbf{j} + \left[w - L_a \cos \theta\right] \mathbf{k}
\]  
(1)

where \( i, j, \) and \( k \) are unit vectors along the \( x-, y-, \) and \( z- \) directions, respectively. The velocity vector of the auxiliary mass is

\[
\mathbf{v}_a = \left[u + L_a \cos \theta \cos \varphi - L_a \sin \theta \sin \varphi \dot{\varphi}\right] \mathbf{i} + \left[v + L_a \cos \theta \sin \varphi + L_a \sin \theta \cos \varphi \dot{\varphi}\right] \mathbf{j} + \left[w + L_a \sin \theta \dot{\theta}\right] \mathbf{k}
\]  
(2)

The linear spring constant is \( k_x \) in the \( x- \) direction and \( k_y \) in the \( y- \) direction. Similarly, the damping coefficient is \( c_x \) in the \( x- \) direction and \( c_y \) in the \( y- \) direction. Ignoring vertical movement of the spring/damper attachment point, the position of the attachment point, relative to a vertical line passing through the moving suspension point, in the \( x- \) and \( y- \) direction is

\[
r_{p,x} = [h_x \sin \theta \cos \varphi] \mathbf{i}
\]  
(3a)

\[
r_{p,y} = [h_y \sin \theta \sin \varphi] \mathbf{j}
\]  
(3b)

The velocity at the attachment point is

\[
\mathbf{v}_{p,x} = \left[h_x \cos \theta \cos \varphi - h_x \sin \theta \sin \varphi \dot{\varphi}\right] \mathbf{i}
\]  
(4a)
The kinetic energy of the auxiliary mass is
\[ T_a = \frac{1}{2} m_a v_a \cdot v_a = \frac{1}{2} m_a \left( \dot{u}^2 + \dot{v}^2 + \dot{w}^2 + L_a^2 \dot{\theta}^2 + L_a^2 \dot{\phi}^2 \sin^2 \theta + 2 u \dot{L}_a \cos \theta \cos \phi \dot{\theta} - 2 u \dot{L}_a \sin \theta \sin \phi \dot{\phi} + 2 \dot{v} L_a \cos \theta \sin \phi \dot{\phi} + 2 \dot{v} L_a \sin \theta \cos \phi \dot{\theta} \sin \theta \right) \] (5)

The potential energy of the auxiliary mass is
\[ U_a = \frac{1}{2} k_x r_x^2 + \frac{1}{2} k_y r_y^2 + m_a g (w - L_a \cos \theta) = \frac{1}{2} k_x h_x^2 \sin^2 \theta \cos^2 \phi + \frac{1}{2} k_y h_y^2 \sin^2 \phi + m_a g (w - L_a \cos \theta) \] (6)

where \( g \) is the acceleration due to gravity. The Raleigh dissipation function for the auxiliary mass is
\[ F_a = \frac{1}{2} c_x v_x^2 + \frac{1}{2} c_y v_y^2 + \frac{1}{2} c_x \left( h_x^2 \cos^2 \theta \cos^2 \phi \dot{\phi}^2 - 2 h_x^2 \cos \theta \cos \phi \dot{\theta} \sin \theta \sin \phi \dot{\phi} + h_x^2 \sin^2 \theta \cos^2 \phi \dot{\phi}^2 \right) + \frac{1}{2} c_y \left( h_y^2 \cos^2 \theta \sin^2 \phi \dot{\phi}^2 + 2 h_y^2 \cos \theta \cos \phi \dot{\theta} \sin \theta \sin \phi \dot{\phi} + h_y^2 \sin^2 \theta \cos^2 \phi \dot{\phi}^2 \right) \] (7)

The kinetic energy of the main mass is
\[ T_m = \frac{1}{2} \{\dot{\Delta}\}^T [M] \{\dot{\Delta}\} = \frac{1}{2} \begin{bmatrix} \ddot{u} \\ \ddot{v} \\ \ddot{w} \end{bmatrix}^T \begin{bmatrix} m_{uu} & m_{uv} & m_{uw} \\ m_{vu} & m_{vv} & m_{vw} \\ m_{wu} & m_{wv} & m_{ww} \end{bmatrix} \begin{bmatrix} \ddot{u} \\ \ddot{v} \\ \ddot{w} \end{bmatrix} \] (8)

where \([M]\) is the mass matrix for the entire main structure. The first three rows and columns correspond to the three degrees of freedom (DOF) of the main structure at the suspension point. The potential (strain) energy of the main mass is
\[ U_m = \frac{1}{2} \{\Delta\}^T [K] \{\Delta\} = \frac{1}{2} \begin{bmatrix} u \\ v \\ w \end{bmatrix}^T \begin{bmatrix} k_{uu} & k_{uv} & k_{uw} \\ k_{vu} & k_{vv} & k_{vw} \\ k_{wu} & k_{wv} & k_{ww} \end{bmatrix} \begin{bmatrix} u \\ v \\ w \end{bmatrix} \] (9)

where \([K]\) is the stiffness matrix of the main structure. The Raleigh dissipation factor is given by
\[ F_m = \frac{1}{2} \{\dot{\Delta}\}^T [C] \{\dot{\Delta}\} = \frac{1}{2} \begin{bmatrix} \ddot{u} \\ \ddot{v} \\ \ddot{w} \end{bmatrix}^T \begin{bmatrix} c_{uu} & c_{uv} & c_{uw} \\ c_{vu} & c_{vv} & c_{vw} \\ c_{wu} & c_{wv} & c_{ww} \end{bmatrix} \begin{bmatrix} \ddot{u} \\ \ddot{v} \\ \ddot{w} \end{bmatrix} \] (10)

where \([C]\) is the damping matrix of the main structure.
The total kinetic energy is $T = T_a + T_m$, the total strain energy is $U = U_a + U_m$, and the dissipation function is $F = F_a + F_m$. The Lagrangian of the system is $L = T - U$. The Lagrange’s equation is

$$\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{q}_r} \right) - \frac{\partial L}{\partial q_r} + \frac{\partial F}{\partial \dot{q}_r} = 0$$

(11)

where $\dot{q}_r$ and $q_r$ are the general velocities and coordinates of the system. For the PTMD system, the general coordinates are $u, v, w, \{\Delta\}, \theta$, and $\phi$ and the general velocities are $\dot{u}, \dot{v}, \dot{w}, \{\dot{\Delta}\}, \dot{\theta}$, and $\dot{\phi}$. Substituting equations (7) through (10) into (11) produces the following system of equations for the MDOF system with a PTMD suspended from the top floor, corresponding to the first three DOF in the mass, damping, and stiffness matrices:

$$\begin{bmatrix} M \end{bmatrix} \ddot{u} + \begin{bmatrix} c \end{bmatrix} \dot{u} + \begin{bmatrix} k \end{bmatrix} u = f$$

(12)

$$\begin{bmatrix} M \end{bmatrix} \ddot{v} + \begin{bmatrix} c \end{bmatrix} \dot{v} + \begin{bmatrix} k \end{bmatrix} v = 0$$

$$\begin{bmatrix} M \end{bmatrix} \ddot{w} + \begin{bmatrix} c \end{bmatrix} \dot{w} + \begin{bmatrix} k \end{bmatrix} w = 0$$

The equations of motion were cast in state-space form and implemented in MATLAB. A simple 5DOF ($u, v, w, \theta, \phi$) was developed for simulating the response of the structure with the PTMD. The frequency response, for the controlled and uncontrolled cases using 25% of that excitation in the $y$-direction as in the $x$-direction is shown in Fig. 2. The magnitude is normalized with respect to the maximum uncontrolled magnitude in the $x$-direction and the frequency is normalized with respect to the natural frequency of the system.
EXPERIMENTAL SET-UP

The experimental set-up of the APMD is shown in Fig. 3. The basic mechanism consists of a 1.5 kg suspended mass, three stepper motors; one for hoisting a tuning frame, and the other two to adjust the valves in the two air dampers, in two orthogonal directions (only one motor and one damper is shown here for clarity). The motors, pendulum and the dampers are mounted on an aluminum frame. The structural model shown in Fig. 3 consists of two steel plates, weighing approximately 120 N each, that serve the function of floor masses, and four aluminum angles that provide the story flexibility. This structure has the first fundamental frequency in the range of 2.1-2.3 Hz with diagonal bracing on the upper floor. The fundamental frequency of the APMD is adjusted by hoisting or lowering a tuning frame whose ends are connected to a sliding aluminum rail through a smooth Teflon bearing assembly. It is worth mentioning here that if one were to hoist the suspended mass instead, torque requirements increase significantly. Hence, this idea was abandoned by the authors during the early design stages. The damping is controlled using two air-dampers with valve adjustments that are capable of generating peak (adjustable) forces between 0-88.0 N.s/m. The valves of the dampers are mated through a customized connector to stepper motors in order to adjust the opening and closing of the valves.

The APMD is controlled using a DSP controller and data acquisition board (dSPACE 1104) with 8 parallel output and 4 parallel and 4 mixed analog inputs. The board also contains extensive digital I/O capabilities that are necessary to control the motors. In order to minimize the overall weight of the adaptive components, the stepper-motor controller was custom-built and consists of an Atmel-Atmega8 microcontroller and three L293 drivers capable of driving three stepper motors at upwards of 2A and 12V each. All three stepper motors are geared with bi-polar windings operating at 2.75W per phase and running torques of 0.21 N-m and 0.85 N-m at 240 PPS. A draw wire potentiometer is used to determine the vertical displacement of the pendulum mass relative to the moving pivot point on the tuning frame. Two laser sensors (Wenglor) are mounted orthogonally in the horizontal plane to determine the location of the pendulum mass. Each measurement sensor outputs an analog voltage that is sent to the analog inputs of the dSPACE controller and data acquisition board.

Fig. 2: Frequency response in x- and y-directions for controlled and uncontrolled
IDENTIFICATION ALGORITHMS FOR TUNING THE ATMD

The effective length of the pendulum is available from the optimization study (from the simulation model) as a function of the imposed excitation and the fundamental natural frequency of the structure. An issue that is critical to the application of adaptive TMDs is the fact that the effective length should be based on as-built conditions and while the structure is in operation. Hence, the identification has to be carried out while the structure is in operation. This is the approach taken in this paper. The motions of the TMD mass is limited by lowering the tuning frame so that it rests flush with the surface of the pendulum mass. Once the motion of the mass is limited in this fashion, the identification phase is carried out. Once the effective length is computed using the identification algorithm, the tuning frame is hoisted to its final location in the control phase.

Two algorithms, one in the frequency domain and the other in the time-domain, are implemented, both using the SIMULINK environment provided by dSPACE. The first method is based on the power spectral density estimate of the data in a moving window. The signal is buffered and windowed at each time step and the buffer length is set at 1024. The process of determining the fundamental frequency is achieved by a peak-picking process and averaging the frequency estimates for multiple windows of the samples with appropriate threshold conditions. The algorithm involves several trigger and state-flow features that cannot be presented here due to space limitations. The second method is carried out in the time-domain and is based on second-order blind identification method, SOBI [11]. A brief summary is provided below.

SOBI is carried out by diagonalizing one or more covariance matrices of measurements. The basic problem statement is cast in the form of a linear static mixtures problem as:

$$
\begin{align*}
\mathbf{x}(k) &= \mathbf{A}\mathbf{s}(k) \\
\mathbf{y}(k) &= \mathbf{Wx}(k)
\end{align*}
$$

where $\mathbf{A} = [a_{ij}]_{n \times n}$ is the instantaneous mixing matrix and $\mathbf{W}_{n \times n}$ is the un-mixing matrix, to be determined. The sources are then given by the vector $\mathbf{y}(k)$, which provides an estimate of

Fig. 3: Experimental set-up of the APMD.
the sources. SOBI seeks to determine the un-mixing matrix $W$ using the information contained in $x(k)$ only. The first step in this method is the simultaneous diagonalization of two covariance matrices $\hat{R}_s(0)$ and $\hat{R}_s(p)$ evaluated at zero time-lag and non-zero time lag $p$, defined as

$$R_s(0) = E \left\{ x(k)x^T(k) \right\} = A R_s(0) A^T$$
$$R_s(p) = E \left\{ x(k)x^T(k) \right\} = A R_s(p) A^T \quad (16)$$

where, $R_s(p) = E \left\{ s(k)s^T(k-p) \right\}$. The simultaneous diagonalization is performed in three basic steps: whitening, orthogonalization and unitary transformation. Whitening is a linear transformation in which $\hat{R}_s(0) = \left( \frac{1}{N} \right) \left( \sum_{k=1}^{N} x(k)x^T(k) \right)$ is first diagonalized using singular value decomposition $\hat{R}_s(0) = V_s \Lambda_s V_s^T$ where $V_s$ are the eigenvectors of the covariance matrix of $x$. Then, the standard whitening is realized by a linear transformation expressed as

$$\bar{x}(k) = Q x(k) = \Lambda_{\bar{x}}^\frac{1}{2} V_s^T x(k) \quad (17)$$

Because of whitening, $\hat{R}_s(p)$ becomes $R_s(p)$ which is given by the equation

$$R_s(p) = \left( \frac{1}{N} \right) \left( \sum_{k=1}^{N} \bar{x}(k) \bar{x}^T(k) \right) = Q R_s(p) Q^T \quad (18)$$

The second step, called orthogonalization, is applied to diagonalize the matrix $\hat{R}_s(p)$ whose eigen-value decomposition is of the form $\hat{R}_s(0) = V_s \Lambda_s V_s^T$. Using equations (16) and (18),

$$\hat{R}_s(p) = QAR_s(p) A^T Q^T \quad (19)$$

If the diagonal matrix $\Lambda_s$ has distinct eigen-values, then the mixing matrix can be estimated uniquely by the equation

$$\hat{H} = Q^{-1} V_\bar{x} = V_s \Lambda_\bar{x}^\frac{1}{2} V_\bar{x} \quad (20)$$

It is easy to see that the product $QA$ is a unitary matrix (since the sources are assumed to be uncorrelated and scaled to have a unit variance), and the problem now becomes one of unitary diagonalization [12] of the correlation matrix $\hat{R}_s(p)$ at one or several non-zero time lags. Equation (19) is a key result, which states that the whitened matrix $\hat{R}_s(p)$ at any non-zero time lag $p$, is diagonalized by the unitary matrix, $QA$. Since $R_s(p)$ is a diagonal matrix (since the sources are assumed to be mutually uncorrelated), the problem now becomes one of diagonalizing the matrix $\hat{R}_s(p)$ resulting in the unitary matrix, $QA$.

Both the frequency domain and the time-domain identification algorithms were implemented using MATLAB scripts and functions embedded into SIMULINK and executed aboard the DSP chip on the dSPACE1104 controller board. Since the identification phase occurs prior to the control phase, time-delay issues are not critical in the current configuration, and a time-step of 0.01 seconds did not cause implementation or stability issues.
BRIEF SUMMARY OF THE EXPERIMENTAL RESULTS

In order to test the experimental set-up proposed earlier, two tests were conducted. First, a forced excitation test was conducted using a harmonic forcing function of 2.2 Hz and the identification carried out using the algorithms described earlier. The valve of the air damper (in one direction) was nearly open. The time-step in the experiment was set to 0.01 sec. For the de-tuned case, the frame was lowered to about 5 mm from the top of the suspended mass, and for the tuned case the frame was hoisted using a motor to the correct height from the center of the suspended mass once the identification phase was completed. The results are shown in Fig. 4. The results of identification from both the identification methods described earlier were identical, and hence only one set of results are presented.

The second test was carried out using an impact excitation from an electro-dynamic shaker. The impact was in the form of a half-sinusoid with a total duration of approximately 0.25 sec. The optimum suspended length was adjusted based on the identified fundamental frequency and the results from the optimal length from simulations performed under similar conditions. The results of this test are shown in Fig. 5. Both these results appear to be promising. These results are of preliminary nature, and efforts are currently underway to conduct a comprehensive set of tests to quantify the behaviour of the APMD systems under various de-tuning conditions, excitations and damping. Having said that, these preliminary tests do demonstrate conclusively that the proposed APMD system, both hardware and software designs, is feasible for practical implementation.

CONCLUSIONS

The issue of de-tuning in TMDs is an important problem that needs effective and practical compensation designs. In this paper, an adaptive pendulum tuned mass damper has been proposed that is capable of tuning its properties to compensate for the de-tuning caused due to varying operating conditions, environment or imposed loading. The hardware and software design used to identify and control the APMD system has been presented. Preliminary experimental results are promising and demonstrate that the proposed APMD system is both
feasible and effective for the control of structural vibrations. More comprehensive experimental studies on this device are currently underway at the University of Waterloo.

![Normalized Top Storey Acceleration](image)

**Fig. 5**: Impact test

**REFERENCES**

DESIGN OF ACTIVE NOISE AND VIBRATION CONTROL FOR CAR OIL PANS USING NUMERICAL SIMULATIONS

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ABSTRACT

The purpose of this paper is to design a smart car oil pan with surface-attached piezoelectric actuators for active noise and vibration reduction using numerical simulations. In the analyses the FEM is applied to model the structural behavior of the oil pan as well as the surface-attached piezoelectric actuators. At first uncoupled structural FE simulations of the oil pan are presented, which are aimed to indentify the most dominant mode shapes within a frequency range of 0-1200 Hz. Based on these results the definition of the actuator positions is performed. In a next step, a fully coupled electromechanical FE model is created by including the piezoelectric actuators. Then, a velocity feedback control algorithm is implemented into the electromechanical FE analysis to provide a closed loop model. In order to evaluate the performance of the designed system, test simulations of the actively controlled oil pan are carried out in the frequency domain and the results are compared with experimental data. Additionally, the exterior noise radiation of the oil pan is computed with the help of the BEM to examine the noise reduction efficiency of the designed system.

Keywords: FEM, BEM, Velocity feedback control, Oil pan, Piezoelectric actuators

INTRODUCTION

Over the past years an increasing attention has been paid to vibration and noise control in automotive engineering. The control of noise and vibration is essential in the design process of an automobile, since it contributes to the comfort, efficiency and safety. There are two different approaches to achieve noise and vibration attenuation. On the one hand, there is the widely used passive approach. Mostly, passive control techniques reduce the vibration and
sound emission of structures by modifying the structural geometry [1] or by applying additional damping materials. These methods are best suited for a frequency range above 1 kHz.

Active noise and vibration control is an alternative way to minimize unwanted structural vibrations and noise that moves more and more into the field of vision for designers. The purpose of this concept is to control the structural and acoustic response by applying actuator forces to the structure. Active noise and vibration control is similar to active structural acoustic control (ASAC) since the actuator signals are also determined based on vibrational inputs, but the goals are different. The ASAC concept is aimed to reduce the acoustic response only.

In active systems piezoelectric ceramics are widely used as sensors and actuators, because they can easily be bonded on or imbedded into conventional structures. In addition, they are lightweight and have relatively high actuating force and relatively low power consumption characteristics. Active control techniques are usually employed in applications where the frequency range of interest is between 50 Hz and 1 kHz.

Considering passenger cars, the power train represents one of the main noise sources. The major contributor to the power train noise emission is the engine oil pan. Therefore, the aim of the paper is to design a smart car oil pan with surface-attached piezoelectric actuators for active vibration and noise reduction. In the following a design study is presented, where the oil pan is regarded separately and free-free boundary conditions are assumed. The decoupled oil pan in combination with an excitation at the oil pan flange captures the relevant structural acoustic behavior, since the sound dominating mode shapes of the oil pan bottom remain unchanged. Other power train components that contribute significantly to the overall sound radiation, such as the valve cover, the gearbox and the cylinder head are not considered here.

The development of a smart oil pan for active noise and vibration control requires efficient and reliable simulation tools. A virtual model is of particular interest in the design process, since it predicts the performance of the smart oil pan and enables the engineer to compare different sensor-actuator configurations and control algorithms. An appropriate model includes not only the passive oil pan and the exterior sound field, but also the sensors and actuators as well as the employed control algorithm. Due to the interactions between these subsystems the simulation becomes a coupled multi-field problem involving the fields of structural dynamics, electromechanics, acoustics and control theory. Hence, the accurate modeling of active noise and vibration control is a challenging task, especially when dealing with complex structures such as the oil pan, where no analytical models exist. In this case numerical methods have to be used, such as the finite element method (FEM) and the boundary element method (BEM).

From the acoustical point of view the oil pan can be treated as a thick-walled structure. This means the influence of the surrounding air on the structural vibrations can be neglected. Additionally it is assumed that the employed control uses only vibrational input signals, and consequently, the acoustical field can be decoupled from the smart oil pan and treated separately. Thus, the calculation of the acoustical field can be done in an independent BE simulation after the vibrational behavior of the smart oil pan has been simulated using a FE analysis.

The first part the paper presents a method for computing the optimal actuator locations based on a FE model. Due to the application of FEM it is possible to model the irregular-shaped geometry of the oil pan. An initial structural uncoupled FE model is aimed to indentify the most dominant mode shapes within a frequency range of 0-1200 Hz. Based on
these results the definition of the piezoelectric actuator positions is performed. A fully coupled electromechanical FE model is created by including the piezoelectric actuators. In a next step, a velocity feedback control algorithm is implemented into the electromechanical FE analysis to provide a closed loop model. The method is called velocity feedback because the output signals from a velocity sensor are multiplied by a constant gain and directly fed back to the surface-attached piezoelectric actuators [2]. The velocity feedback control algorithm combines high performance control with robustness against time variance of the operating parameters such as oil temperature and excitation frequency.

The main purpose of the present paper is to evaluate the performance of the designed system. For that reason FE simulations of the actively controlled oil pan are carried out in the frequency domain. Additionally, the sound field radiated by the vibrating oil pan is computed using the BEM. The BEM offers computational advantages, because it requires a discretization of the boundary only. The BEM automatically fulfills the Sommerfeld radiation condition and does not produce reflections at the boundaries. For this reason the BEM allows the numerical prediction of sound pressure fields in exterior so-called unbounded domains. One drawback of the classical BEM is the fact that the resulting matrices are frequency dependent, fully populated and non-symmetric.

The modeling of the oil pan, the piezoelectric actuators as well as the implementation of the controller is performed using the software package MATLAB. In order to verify the numerical modeling, experiments have been done, which show that the results of the computation are almost identical to those of the experiments. Furthermore, the experimental and the numerical results reveal that the structural vibration of the oil pan and the radiated sound pressure is reduced by means of velocity feedback control.

**FINITE ELEMENT ANALYSIS OF THE DOMINANT MODE SHAPES**

In order to design an active system to reduce the vibrations of the oil pan in a noise reducing manner, it is essential to indentify the most dominant mode shapes. This step is carried out by means of harmonic FE simulations using quadratic 10-node tetrahedral elements. The FE formulation for modeling the stationary behavior of the uncoupled oil pan can be written as

\[
\begin{bmatrix}
-\Omega^2 M_u + i \Omega C_u + K_u
\end{bmatrix}
\begin{bmatrix}
\ddot{u}
\end{bmatrix}
= \begin{bmatrix}
\tilde{f}_u
\end{bmatrix},
\]

where the vector \( \ddot{u} \) represents the complex amplitudes of the nodal structural displacements. The variable \( \Omega \) denotes the excitation frequency and \( i \) is the imaginary unit. The matrices \( M_u \) and \( K_u \) are the structural mass and the structural stiffness matrix, respectively. For convenience, a Rayleigh damping is introduced into the system of equations (1) assuming that the damping matrix \( C_u \) is a linear combination of the matrices \( M_u \) and \( K_u \). The external loads are stored in the mechanical load vector \( \tilde{f}_u \).

A harmonic analysis in combination with a point force excitation at the oil pan flange has been used to compute the dominant mode shapes. It is noted that the point force location was chosen in such a way that all eigenmodes in the frequency range up to 1200 Hz are excited. Figure 1 shows the frequency response function (FRF) between the structural displacement at
the center of the oil pan bottom and the excitation force at the flange. In addition, the mode shapes that are associated with the respective resonance frequencies are illustrated.

**Fig. 1:** Computed FRF.

In Figure 1, it can be seen that the first eigenmode is a pure torsional mode. The second, fourth and fifth modes are pure bending modes of the oil pan sides and the third and sixth modes are pure bending modes of the oil pan bottom. Under real operating conditions the bottom modes are the main contributor to the overall sound emission. Due to this fact, the present paper aims to control the bottom modes only.

**DEFINITION OF THE ACTUATOR POSITIONS AND MODELING**

The choice of suitable actuator positions depends on many factors, such as the employed control and the vibrational behavior of the structure. An often used method for the actuator placement is based on the assumption that an actuator is placed well when it is able to influence significantly the shape of the structural modes. This means that an actuator should be placed at positions on the surface of the structure, where the strains are the highest [3,4]. In case of the oil pan, the third and sixth eigenmode are considered. In order to obtain the modal strains of these modes, the linear eigenvalue problem

\[
\left[ -\omega^2 M_u + K_u \right] \hat{u} = 0 ,
\]

needs to be solved first. The solutions of equation (2) are the angular eigenfrequencies $\omega_j$ and the corresponding eigenmodes of the nodal displacements $\hat{u}_j$ with $j = 3$ and $6$.
The modal displacements $\hat{u}_j$ are associated with the modal strains $\hat{\varepsilon}_j$ by the relationship

$$\hat{\varepsilon}_j = B\hat{u}_j, \quad (3)$$

where $B$ is the matrix that calculates the modal equivalent strains at the Gauss points using the von Mises equation. By means of a multiplicative superposition of the modal strains $\hat{\varepsilon}_j$ one obtains the superposed strain field

$$\hat{\varepsilon}_{\text{max}} = \prod_{j=3, 6} \hat{\varepsilon}_j. \quad (4)$$

In contrast to an additive superposition, the multiplicative superposition makes sure that the actuators are not placed on node lines. A contour plot of the superposed strain field $\hat{\varepsilon}_{\text{max}}$ allows the definition of optimal actuator positions.

Two actuator positions have been chosen according to the contour plot visible on the left-hand side of Figure 2. On right-hand side of Figure 2 the light gray areas mark the selected positions.

An electromechanical FE model of the oil pan is obtained by integrating the piezoelectric actuators in equation (1). The actuators are modeled using 6-node multilayer triangular shell elements. Each Element has one additional degree of freedom $\varphi$ to model the electrical potential of the piezoelectric layer. Throughout the shell element it is assumed that the electric potential is constant and varies linearly through the thickness of the piezoelectric layer.

As derived in [5] the FE formulation for modeling the stationary behavior of the oil pan and the surface-attached piezoelectric actuators can be written as

$$\begin{bmatrix} -\Omega^2 M_u + i\Omega C_u + K_{ep} & K_{ep}^T \\ K_{ep} & -K_{ep} \end{bmatrix} \begin{bmatrix} \bar{u} \\ \bar{\varphi} \end{bmatrix} = \begin{bmatrix} \bar{f}_u \\ \bar{f}_\varphi \end{bmatrix}, \quad (5)$$

where the vector $\bar{\varphi}$ represents the complex amplitudes of the electric potentials and the matrix $K_{ep}$ is the dielectric matrix. The piezoelectric coupling arises in the piezoelectric coupling matrix $K_{ep}$. The charge on the actuator is stored in the electric load vector $\bar{f}_\varphi$. It is important to notice that the vector $\bar{u}$ contains the nodal displacements of the triangular shell elements and of the tetrahedral elements.
MODELING OF CONTROL

The design of a smart oil pan requires not only the simulation of the electromechanical system, but also the implementation of a suitable control algorithm. In the present study, the robust and widely used velocity feedback control is applied [2]. This means that the normal velocity $u_c$ of a given point on the surface of the oil pan is amplified by a constant gain $g_c$ and directly fed back to the surface-attached piezoelectric actuators in terms of the voltage $\phi_c$. With this voltage the piezoelectric actuators generate counteracting forces, which suppress the structural vibrations, and consequently, the resulting sound radiation. The control law of velocity feedback reads

$$\tilde{\phi}_c = g_c i\Omega \tilde{u}_c.$$  \hspace{1cm} (6)

One important requirement for a successful feedback is the availability of appropriate feedback points. Based on the computed mode shapes, shown in Figure 1, one feedback point is chosen for each actuator. The chosen points are located close to the corresponding actuators. The collocated design is important to guarantee control stability.

Using a separate sensor for each actuator leads to a decentralized feedback control strategy with two independent local feedback loops. Thus, the control law (6) can be written in the vector-matrix notation

$$\tilde{\phi} = i\Omega g_c \mathbf{P}_c \tilde{u},$$ \hspace{1cm} (7)

where the matrix $\mathbf{P}_c$ characterizes the position, where the feedback velocities are detected.

The influence of the controller can be considered in the numerical modeling by substituting the control law into the equation (5). Due to the substitution, an additional damping term occurs on the left-hand side of equation (5). A simplification of the equation can be obtained by deleting the rows, which are related to the electric degrees of freedom of the used piezoelectric actuators. The reduced system reads

$$[-\Omega^2 \mathbf{M}_u + i\Omega (\mathbf{C}_u + \mathbf{K}_{up} g_c \mathbf{P}_c) + \mathbf{K}_u] \tilde{\mathbf{u}} = \tilde{\mathbf{f}}_u.$$ \hspace{1cm} (8)

The system of equations (8) describes the controlled behavior of the smart oil pan. The additional damping term in equation (8) points out that the velocity feedback control increases the viscous damping.

In order to evaluate the performance of the designed system, test simulations are carried out and the results are compared with experimental data. For the comparison, uncontrolled and controlled FRFs are considered. More details concerning the experimental testing of the designed system can be found in [6].
In both Figures, it can be observed that the measured data and the numerical predictions agree very well. Additionally, the results in Figure 4 show that a significant damping at the dominating resonance frequencies is achieved, due to the implementation of velocity feedback control. The amplitudes are reduced by more than 24 dB at 636 Hz and by about 11 dB at 1050 Hz. The structural response at other resonance peaks is uncontrollable with the applied actuators. Even a slight amplification can be seen at 810 Hz.

**BOUNDARY ELEMENT ANALYSIS OF THE SURROUNDING SOUND FIELD**

Generally, it is assumed that vibration control leads to a simultaneous reduction of sound radiation. However, it is also possible that the controlled vibrations cause a higher sound pressure in some places. For this reason, the resulting sound field plays an important role by evaluating the performance of the designed control. In the present work, the BEM is used to characterize the acoustic field of the smart oil pan. Due to this technique only the surface of the oil pan has to be discretized. Linear boundary elements with four nodes and two degrees
of freedom at each of the nodes are employed for the discretization. The direct BE matrix equation reads [7]

$$Hp = -i \rho_0 \Omega G v_n,$$ \hspace{1cm} (9)

where $H$ and $G$ are the influence matrices and the vectors $p$ and $v_n$ are the nodal values of the acoustic pressure and the normal velocity. The influence matrices $H$ and $G$ are fully populated and have to be computed for each frequency $\Omega$.

In order to perform a frequency response analysis, the structural displacements of the smart oil pan obtained from the spectral FE analysis have to be interpolated onto the grid points of the BE mesh and applied as boundary conditions. To be able to determine the sound pressure distribution, a considerable amount of field points is defined, which are located on a plane parallel to the oil pan bottom. After the acoustic pressure is calculated at all points, a contour plot allows to visualize its spatial distribution. In Figures 10 and 11 the computed sound pressure distribution of the uncontrolled and controlled oil pan are plotted, which occur when the flange is excited with a harmonic force of amplitude 1 N. The chosen plane is approximately 50 mm apart from the bottom surface. To test the noise reduction efficiency of the designed system and to verify the simulated data, near-field airborne noise measurements were carried out in a free-field room [5].

![Simulation (636 Hz) vs Measurement (616 Hz)](image1)

**Fig. 5**: Sound pressure distribution of the uncontrolled oil pan.

![Simulation (636 Hz) vs Measurement (616 Hz)](image2)

**Fig. 6**: Sound pressure distribution of the controlled oil pan.
In both Figures it can be noticed that the simulation results correlate well with the experimental results. Furthermore from Figure 6 can be seen that due to the controller influence the sound pressure level is reduced by approximately 16 dB, which indicates the noise reduction potential of the designed system.

**CONCLUSIONS**

On the basis of FE and BE simulations a smart car oil pan is designed to reduce the structural vibrations in a sound reducing manner. For noise reduction, optimal locations of two piezoelectric actuators attached to the bottom surface have been studied. The FEM is applied to model the structural behavior of the oil pan as well as the surface-attached piezoelectric actuators. The BEM is used to describe the exterior sound field. A velocity feedback control algorithm is implemented into the numerical model to obtain an active damping effect. With velocity feedback control, attenuations of about 24 dB in vibration level and 16 dB in sound pressure level at the resonance frequencies of the most dominant modes of the smart oil pan have been achieved. In order to show that the designed system works also in reality experimental tests have been performed. A comparison between the experimental and numerical results shows a good agreement.

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INTelligent CONTROLLER FOR SMART BASE ISOLATION OF MASONRY STRUCTURES

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ABSTRACT

In this paper, an intelligent controller to control a smart base-isolated masonry structure is proposed. The smart base isolation consists of rubber bearings and a magnetorheological damper. This semi-active scheme takes advantage of the low power requirement of magnetorheological devices to achieve structural control.

The controller is capable of mapping the behaviour of the controlled structure sequentially, using a self-organizing neural network which has time-varying learning rates. This neurocontroller is augmented by a sliding mode controller to account for the uncertainty on the semi-active damping forces. Wavelets are used as functions in the single hidden layer for their capability of localizing functions in the space and frequency domains.

Results show that the intelligent controller is capable of rapidly mapping the inverse state-force relationship, without any prior knowledge of the dynamic properties of the controlled structure. The displacement control performance is similar to the performance from an active hybrid strategy, and the inter-story displacement is efficiently minimized.

Keywords: Neurocontroller, magnetorheological dampers, adaptive control, intelligent control, civil structures, base isolation

INTRODUCTION

In North America including Canada, a large number of historical buildings are unreinforced masonry or brick masonry. Without the presence of reinforcement in these structures, brittle
behaviours are to be expected in extreme loading events such as earthquakes. Such brittle behaviours are typically associated with significant economic and fatal losses. The heterogeneity and anisotropy of masonry structures also complicate the prediction of their mechanical behaviour in different stress states. Numerical modeling of masonry structures at micro- [1-2], meso- [3], and macro- [4-5] levels, as well as experimental investigation [6-9], have been reported in the literature. Failures of masonry structures can be attributed to various causes including lack of anchorage, anchor failure, in-plane failures, out-of-plane failures, combined in-plane and out-of-plane effects, and diaphragm-related failures [10]. These causes represent the low-ductility characteristic of masonry structures, leading to the vulnerability of masonry structures in seismic events.

Base isolation is a well understood technique often used to improve the performance of masonry structures for earthquake loadings. Base isolation is traditionally performed using a rolling pendulum system or lead-rubber bearings (LRB). The rolling pendulum system is known to be more cost effective than the LRB system, but has the disadvantage of a potential roll-off over large displacements [11]. Alhan and Gavin [12] surveyed research on control and uncontrolled base isolation systems, and noted that the addition of a passive system can significantly improve the performance of base isolation. Active systems can outperform passive systems, and their cost can be justifiable over a long period of control [13]. Semi-active systems, on the other hand, require little power and their performance is comparable to the performance of active systems. For this reason, the controlled system this paper is a base isolation system equipped with a magnetorheological (MR) damper. The main advantage of the hybrid semi-active system is the exclusion of the hysteretic material in the rubber bearing, which eliminates replacement cost and inoperability period associated with plastic deformation of the LRB system resulting from an earthquake. This strategy has been studied in the [11, 14]. This paper focuses on the on the semi-active controller of the smart base isolation strategy.

Application of semi-active and active control schemes to civil engineering structures is impeded by the inherent size of plants to be controlled. Three main control obstacles are specific to the field of civil engineering: 1) large actuating forces required; 2) uncertainties in dynamic characteristics; and 3) limited state measurements [15]. The use of semi-active devices, such as MR dampers, has been proposed for control of civil structures because of their capability to perform almost as well as active control schemes for mitigation of natural hazards, while requiring only a fraction of the power input [16-17]. The problem of uncertain dynamic properties and limited state measurement are more of a control and system identification issue. The use of adaptive controllers is therefore recommended.

Several adaptive controllers have been studied for control of civil structures. Robust controllers capable of performing despite system uncertainties, such as sliding mode control [18] and adapted LQR/H2 control [19], have been proposed but the performance of such controllers is dependent on the initial estimation of the system properties. Neurocontrollers have also been proposed because of their capability to map nonlinear complex functions. Sanner and Slotine [20] were the first to use Gaussian radial functions for active control of nonlinear systems. Cannon and Slotine [21] proposed to use adaptive wavelets for neurocontrol.
Some applications of neurocontrol to civil structures have been recently studied, and are reviewed in [22]. This paper presents an adaptive neurocontroller for the application to a hybrid MR-base isolation system for masonry structures. The controller is a modification of prior work from the authors to provide enhanced robustness, as well as better space and frequency localization of the controlled plant. Its performance for a near-field type earthquake is assessed. This proposed controller has the unique characteristics of sequential learning and adaptive learning rates based on Lyapunov stability theory that are capable of accelerating convergence during high magnitude excitations. The applicability of this type of neurocontroller to large-scale civil structures is discussed in [22].

The paper is organized as follows: section 2 discusses MR dampers; section 3 presents the neurocontroller; section 4 describes the simulation; section 5 presents and discusses the results; section 6 concludes the paper.

MAGNETORHEOLOGICAL DAMPERS

The application of MR dampers for control of civil structures has attracted some attention in the research community since the 1990’s. Their low power requirement, along with their mechanical robustness and fail-safe property make them excellent candidate for vibration mitigation of large-scale systems. MR dampers are capable of generating a reaction force of 200 kN in 60 milliseconds, with a 50 W power input [23].

The mathematical complexity of MR dampers renders complicated the mapping of the required voltage for a desired force [24]. Some non-mathematical models have been developed, such as adaptive identification [25]. Despite that those models could be used with sequential learning, the use of the clipped-optimal algorithm proposed in Dyke et al. [26] has shown excellent performance and simplicity [16,27]. This scheme is used for the neurocontroller.

Because of the inherent state-dependence of the semi-active device, the required force by the proposed neurocontroller will not necessarily be achieved by the MR damper. There will be an error \( \bar{u} \) arising from the difference between the required force \( u \) and the actual force \( u_{act} \):

\[
\bar{u} = u - u_{act}
\]  

This error on the force is treated as a system uncertainty and is handled using a sliding mode controller described in the next section.

CONTROLLER

The proposed controller is a modification of the controller presented in [22]. The robustness of the controller is significantly enhanced by the incorporation of a smooth interpolation between the required weight and the actual weight of a newly added node. Also, the functions of the neurocontroller are changed to “mexican hat” wavelets for their better space and frequency localization property, based on the work of Cannon and Slotine [21]. For completeness, the neurocontroller algorithm is fully described in this section.
Neurocontroller Architecture

The controller is a self-organizing single layer neural net comprising mexican hat wavelets $\phi$ of the form:

$$
\phi(x) = \left(1 - \frac{||x - \mu||^2}{\sigma^2}\right)e^{-\frac{||x - \mu||^2}{\sigma^2}}
$$

where $x$ is the input, $\mu$ and $\sigma$ are the center and bandwidth of the function respectively. Fig. 1 illustrates the mexican hat wavelet.

![Fig. 1: Mexican hat wavelet; $\mu = 0$ and $\sigma = 0.01$.](image)

The goal of the neurocontroller is to map the relationship control-input/state-output of the hybrid system. The network output, which is the required force $u$, can be written:

$$
u_j(x) = \sum_{i=1}^{h} \alpha_{ij} \phi_{ij}(x) = \alpha_j^T \phi(x)
$$

where $\alpha$ is the nodal weight associated with the output $j$. For simplicity, the neurocontroller will be presented for the case of a single actuator. The subscript $j$ will be dropped.

The self-organizing feature of the neurocontroller consists of adding a node when the Euclidian distance of a new output to the closest node is farther than the threshold $\lambda$. The network also has the capacity to prune nodes when their weights are found to be under a predefined ratio of the largest weight for several consecutive time steps. New nodes are added at the center of the new input, with the target weight $\alpha_T$ in function of the network error, while the bandwidth is computed based on the designed network density $\lambda$. Weights, centers and bandwidths are adaptive, and their adaptation rules are derived in the next subsection.

Adaptation Rules

Based on (3), the desired (optima) force $u_d$ from the network is written:

$$
u_d = \alpha_T^T \phi(x)
$$
Using (4) and the conventional state-space representation of the equation of motion of a civil structure, the dynamics of the controlled state error can be written as:

$$\dot{e} = \dot{X} - \dot{X}_d = Ae + B(u_{act} - u_d + \epsilon) = Ae + B(\hat{\alpha}^T\phi - \alpha^T\phi - \bar{u} + \epsilon)$$  \hspace{1cm} (5)$$

where $X$ is the system state vector, $A, B, \text{ and } u$ are notations that conform to the traditional state-space representation, the subscript $d$ denotes the desired states, the hat denotes estimated values, and $\epsilon$ is the estimation error. A sliding mode controller is used along with the following sliding surface as:

$$s = Pe$$  \hspace{1cm} (6)$$

where the targeted surface is $s = 0$, $P$ is a user-defined vector that can be designed following [28]. Introducing the following control law:

$$u_{act} = \hat{\alpha}^T\phi - k(\text{sign}(s))$$  \hspace{1cm} (7)$$

where $k$ is a constant to be defined later, (5) becomes:

$$\dot{e} = Ae + B\left(\hat{\alpha}^T\phi - \alpha^T\phi - \bar{u} + \epsilon - k(\text{sign}(s))\right)$$  \hspace{1cm} (8)$$

In order to find adaptation laws that would guarantee the stability of the wavelet network, Lyapunov theory is applied. Consider the following Lyapunov candidate:

$$V = \frac{1}{2}\left[s^T \hat{\alpha} + \hat{\alpha}^T \Gamma^{-1}_\alpha \hat{\alpha} + \phi^T \Gamma^{-1}_\phi \phi\right]$$  \hspace{1cm} (9)$$

where $\Gamma^{-1}_\alpha$ and $\Gamma^{-1}_\phi$ are positive definite diagonal matrices representing learning parameters, the tilde denotes the error between the estimated and real values ($\tilde{\alpha} = \alpha - \hat{\alpha}; \tilde{\phi} = \phi - \phi$). Note that here $s$ is a scalar. Neglecting the higher order term, the time derivative of $V$ is:

$$\dot{V} = s^TPAe + s^TPB\left(\hat{\alpha}^T\phi + \alpha^T\phi\right) + \hat{\alpha}^T \Gamma^{-1}_\alpha \dot{\alpha} + \phi^T \Gamma^{-1}_\phi \dot{\phi} + \alpha^T \Gamma^{-1}_\alpha \tilde{\alpha} + \phi^T \Gamma^{-1}_\phi \tilde{\phi} + s^T PB\epsilon - s^T PB\bar{u} - |s|^2PBk$$

$$\dot{V} = e^T P^T PAe + \phi^T \left(\hat{\alpha}^T B^T P^T s + \Gamma^{-1}_\phi \dot{\phi}\right) + \alpha^T \left(\phi^T B^T P^T s + \Gamma^{-1}_\phi \dot{\phi}\right) - s^T PB(\bar{u} - \epsilon) - |s|^2PBk + \xi^T \Gamma^{-1}_\xi \xi - \phi^T \Gamma^{-1}_\phi \tilde{\phi}$$

with:

$$\xi = \begin{bmatrix} \tilde{\alpha} \\ \tilde{\phi} \end{bmatrix}, \Gamma = \begin{bmatrix} \Gamma_\alpha & 0 \\ 0 & \Gamma_\phi \end{bmatrix}$$

The tilde denotes the error between the optimal and current parameters. By choosing the following adaptation laws:
\[
\begin{align*}
\dot{\alpha} &= -(\Gamma_\alpha \dot{\phi}) B^T P^T s \\
\dot{\phi} &= -(\Gamma_\phi \alpha) B^T P^T s \\
\dot{\Gamma}^{-1} &= -s^T s I
\end{align*}
\]

where \( I \) is an identity matrix to populate \( \dot{\Gamma}^{-1} \), equation (10) becomes:

\[
\dot{V} = e^T P^T P A e - s^T P B (\ddot{u} - \epsilon) - |s^T P B k| \xi^T (s^T s I) \xi - \phi^T \Gamma_{\phi}^{-1} \phi
\]  

In the discrete form, the adaptation laws for \( \mu \) and \( \sigma \) are obtained by taking the partial derivatives of \( \phi \). Choosing \( k = \eta u_b \), where \( \eta \) is positive, and \( u_b \) is a known bound (also positive) on \( \ddot{u} \), (12) can be rewritten as:

\[
\dot{V} = e^T P^T P A e - s^T P B (\ddot{u} - \epsilon) - |s^T P B \eta u_b - \xi^T (s^T s I) \xi - \phi^T \Gamma_{\phi}^{-1} \phi
\]

where the term \( k \) depends on the uncertainty on the error defined in (1). This error can be quite large, and increasing its value too much would lead to a controller based almost exclusively on the sign of the sliding surface. Instead, a bound \( u_b \) is assumed, and adaptation on the network stopped when \( |\ddot{u}| > u_b \). Each term of equation (13) can be shown to be at least negative semi-definite, except for the last term. Furthermore, as discussed previously, a smooth interpolation is incorporated when nodes are added to ensure the smoothness of the Lyapunov function (9):

\[
\dot{\alpha} = \begin{cases} 
-(\Gamma_\alpha \dot{\phi}) B^T P^T s & \text{if the node reached } \alpha_T \\
\eta(t) \alpha_T & \text{if the node was added but has not reached } \alpha_T \\
0 & \text{if the node is added}
\end{cases}
\]  

with \( \eta(t) \) being an infinitely differentiable function such as the sigmoid function.

**SIMULATION**

A simulation has been done on a three degrees-of-freedom system used in previous work [23,27]. The stiffness at the base has been reduced by 40% to model a base isolation system consisting of rubber bearing. A 1100 N MR damper is attached between the ground slab and the first floor. Structural properties are as follows:

\[
M = \begin{bmatrix} 98.3 & 0 & 0 \\ 0 & 98.3 & 0 \\ 0 & 0 & 98.3 \end{bmatrix} \text{ kg}; \quad C = \begin{bmatrix} 175 & -50 & 0 \\ -50 & 100 & -50 \\ 0 & -50 & 50 \end{bmatrix} \text{ N} \cdot \text{s} / \text{m}; \quad K = 10^5 \begin{bmatrix} 7.2 & -6.84 & 0 \\ -6.84 & 13.7 & -6.84 \\ 0 & -6.84 & 6.84 \end{bmatrix} \text{ N} / \text{m}
\]

To evaluate the performance of the controller, the simulation subjected the structure to the ElCentro 1940 earthquake, North-South component. This earthquake is a near-field type, and has the possibility to provoke a roll-off of the structure. The time scale of the earthquakes has been scaled by a factor of five to take into consideration the model sizing. The neurocontroller uses displacement and velocity inputs, along with the previous applied forced. Because large displacements are the main cause of structural failures during an earthquake, the dynamic
outputs to be controlled are the floor displacements. Results are shown and compared in the next section.

RESULTS

The network parameters quickly converge when the neurocontroller uses an actuator, but the convergence rate is slower when the MR damper is used, as expected. Fig. 2 shows the time series plots obtained from the simulation. Fig. 2a) compares the performance of the controller using the MR device to the performance of the same controller using an actuator. Results show that the performance of an MR damper is similar to the performance of an actuator. Fig. 2b) shows the time series results for the semi-active controller along with the passive-on and passive-off cases. Passive-off refers to the MR damper with no current input, while the passive-on refers to the MR damper with full current input. In addition to the advantage of saving power, the semi-active control strategy appears to be more efficient at mitigating the 3rd floor displacement.

Fig. 2: 3rd floor displacements for El Centro earthquake;
(a) uncontrolled case compared to semi-active and active control strategies
(b) semi-active control strategy compared to passive-on and passive-off cases

Fig. 3 compares the inter-story displacements of various control strategies. It is observed that the semi-active controller is capable of keeping a minimal inter-story displacement, while limiting the displacement at the first floor. The passive-on case gives the best performance at mitigating the first floor displacement as it is equivalent to locking the floor, but this is done at the expense of larger inter-story displacements, meaning that the advantage of base isolation is cancelled.
Fig. 3: Maximum floor displacement for each strategy

Table 1 summarizes the displacement reductions for various control strategies. The linear quadratic regulator (LQR) optimal controller is designed using full knowledge of the plant for comparison purposes. The semi-active control strategy shows comparable performance with the active case.

<table>
<thead>
<tr>
<th></th>
<th>Passive-on</th>
<th>Passive-off</th>
<th>Semi-active control</th>
<th>Active control</th>
<th>LQR control</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st floor</td>
<td>93.6%</td>
<td>82.3%</td>
<td>89.4%</td>
<td>92.7%</td>
<td>92.9%</td>
</tr>
<tr>
<td>2nd floor</td>
<td>88.4%</td>
<td>81.7%</td>
<td>89.2%</td>
<td>88.2%</td>
<td>92.4%</td>
</tr>
<tr>
<td>3rd floor</td>
<td>82.3%</td>
<td>80.7%</td>
<td>88.8%</td>
<td>87.1%</td>
<td>88.3%</td>
</tr>
<tr>
<td>max damping force (N)</td>
<td>982</td>
<td>211</td>
<td>731</td>
<td>1100</td>
<td>1100</td>
</tr>
</tbody>
</table>

Table 1: Summary of displacement reductions for each strategy.

Table 2 summarizes the performance of the neurocontroller with different inputs. The proposed neurocontroller uses displacement and velocity feedback, but its performance has been studied for the case of pure acceleration feedback. Notice that acceleration feedback would normally prevent the use of an adaptive observer or numerous sensors, but since the neurocontroller uses a sliding mode control scheme, displacement and velocity measurements will be needed to compute the sliding surface. The importance of carefully choosing the network inputs is highlighted in this analysis.
Table 2: Comparison of displacement/velocity feedback with acceleration feedback

<table>
<thead>
<tr>
<th></th>
<th>1st floor</th>
<th>2nd floor</th>
<th>3rd floor</th>
<th>maximum damping force</th>
</tr>
</thead>
<tbody>
<tr>
<td>displacement and velocity feedback</td>
<td>0.164</td>
<td>0.173</td>
<td>0.182</td>
<td>731</td>
</tr>
<tr>
<td>pure acceleration feedback</td>
<td>0.178</td>
<td>0.206</td>
<td>0.235</td>
<td>740</td>
</tr>
<tr>
<td>relative change</td>
<td>8.5%</td>
<td>19%</td>
<td>29%</td>
<td>1.2%</td>
</tr>
</tbody>
</table>

CONCLUSION

From the simulation and analysis results provided in this paper, it is found that the proposed neurocontroller provides good performance for controlling the base displacement of a smart base isolation scheme by efficiently minimizing the inter-story displacements. Its performance is similar to the active case, and the displacement shape of the uncontrolled case is preserved. Current work is undergoing at MIT to formalize the type of inputs that can optimize the architecture of the neurocontroller.

REFERENCES

ACHIEVING CONSISTENT SMA ACTUATION TIMES UNDER VARYING AMBIENT THERMAL CONDITIONS

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ABSTRACT

Shape Memory Alloys (SMAs) have been implemented as actuators in a wide range of applications spanning fields such as robotics, aeronautics, automotive and medicine. However, controlling SMA actuators is no simple task as they are highly nonlinear due to the inherent hysteresis.

In particular, the thermal nature of the SMA phase transformation means that the surrounding ambient conditions, such as temperature and air flow, have a direct effect on the time needed for the SMA wire to actuate. For example, if the surrounding temperature is high, the wire will contract in a shorter period of time at fixed current, compared to when the surrounding temperature is low. In some applications, such as automotive, this is a very important factor to consider as one key objective in such applications is attaining consistent actuation times across a broad range of ambient conditions.

Thus, the focus of this work is devising a method to actuate an SMA wire in a more consistent time regardless of the ambient conditions and stresses applied to the SMA wire. One way to achieve this is to preheat the SMA wire prior to actuation, a technique referred to as “priming”. We introduce an adaptive priming method, based on resistance feedback, which shows improved consistency in SMA actuation in lab tests under varying air-flow conditions.

Keywords: Shape Memory Alloys, Priming, Probing, PI control.
INTRODUCTION

Shape memory alloys (SMAs) are a group of metallic materials that demonstrate the ability to return back to a previously defined shape or size if subjected to the appropriate thermal procedure. In particular, the contraction of an SMA wire under load is governed by its internal temperature and this temperature can be controlled by regulating current flow through the wire. Primarily because of convective heat losses, larger currents are required for a given contraction under conditions of lower ambient temperature or higher convective medium flow. However, if too much current is applied, the wire will overheat and be damaged or destroyed. Manufacturers generally specify a recommended "safe current" that can theoretically be applied indefinitely to the wire without the risk of overheating, but this specification is given at room temperature and under specific convection conditions; it will not actuate the wire in a consistent time over a range of environmental conditions. The stress applied to a wire also affects the required actuation power, since it alters the transformation temperatures of the material.

The goal of this work is to design and implement a controller that will actuate an SMA wire in a consistent time regardless of wire heat loss due to ambient air flow conditions when a fixed load is applied. This is achieved by making use of the following observation: during heating, an SMA’s resistance increases to a maximum value or cusp prior to contraction, then decreases to a lower austenite resistance value. For example, as shown in Fig. 1, the resistance \( R \) is above 1.5Ω with 0.1A running through the wire but first increases to 1.6Ω then, decreases to under 1.4Ω when the current is ramped up to 1A and the wire is actuated (90s to 100s). Therefore, if a controller can be developed that pre-heats the SMA so that its resistance is at its maximum value prior to actuation, then it is possible to actuate the wire in a consistent time period regardless of the ambient air flow conditions without the danger of overheating it. We call this process “priming”.

Since we anticipate applications where actuator priming is desirable even if actuation is eventually cancelled (e.g., in a safety application where a vehicle is prepared for an impact which then does not occur), the proposed current control strategy will be divided into two parts: a priming controller and an actuation controller. The goal of the priming controller is to heat the SMA so that \( R \) is at (or close to) the cusp. Here, we are using the cusp as a proxy for the austenite start temperature \( (A_s) \) which will be our consistent state before actuation. For the actuation controller, the goal is to apply a current that will drive \( R \) down to its lowest point in a consistent time period. Similarly, we are using the lowest \( R \) value as a proxy for the austenite finish temperature \( (A_f) \) which will aid with the control of our actuation current.
LITERATURE REVIEW

According to Ikuta [1], SMA feedback control can be classified into two categories: feedback with external variables (e.g. force, displacement) or with internal variables (e.g. temperature, resistance). External feedback methods require the use of sensors that increase system size and cost. Moreover, since there is no measure of wire temperature, overheating can occur. Similarly, using an internal variable such as temperature would require the use of a sensor increasing system size and cost. Moreover, accurately measuring the temperature of a thin wire is difficult as highlighted by Kuribayashi [2]. Fortunately, the relationship between $R$ and temperature is only slightly hysteretic so that $R$ can be measured to reasonably predict the temperature of an SMA wire [1]. Furthermore, $R$ is easily computed in real-time without the need for sensors and this approach has been adopted by several researchers [3], [4].

In the literature, there are a number of researchers who have implemented what we call “priming” and “actuation” strategies. For example, Allston et al. [3] implemented control strategies that incorporate priming. They used $R$ as a proxy for temperature in an SMA fuel injector where $R$ was monitored to keep the SMA wire at the martensite start temperature ($M_s$) prior to actuation. In [5], Seldon et al. used actuator priming in the position control of an SMA wire. They divided the wire into segments and treated each segment as an independent actuator. Each segment was held at either $M_s$ or at $A_s$, depending on the segment's last action. As such, if the wire was to be contracted, a segment at $A_s$ would be heated and if the wire was to be expanded, a segment at $M_s$ would be cooled. This method took advantage of the strain-temperature hysteresis allowing for faster actuator response time. However, the individual segment control and the use of a thermocouple to measure wire temperature add significant complexity to the system.

With regard to actuation strategies, Teh et al. [4] implemented an actuation controller that switched from a high current to a safe current depending on the value of $R$ during actuation although environmental conditions were held constant. The actuation control strategy presented in this paper will build on the approach in [4] to account for variable ambient conditions.
EXPERIMENTAL SETUP

The major components of this experiment include a Xantrex XPD 33-16 programmable power supply, a Newport 443 Series linear stage, a 250μm diameter 90°C Flexinol wire from Dynalloy, a Quanser MultiQ3 data acquisition board, a Data General 6070 Force Actuator, an EG&G Current amplifier for the force actuator (model CO502-001), a Honeywell Sensotec Force Cell (model AL311 101b), a Ball Bearing Optical Shaft Encoder from US Digital (model H5-1024-I-S), a Jamicon 12V 0.15A rotary DC mini-fan, and a PC loaded with MATLAB/Simulink.

![Fig. 2: Top view of experimental setup [6]](image)

As shown in Fig. 2, one end of the SMA wire is fastened to the XYZ stage while the other end is fastened to a clamp that is connected to the force cell and the encoder. The Xantrex power supply, the force actuator (through the current amplifier), the force cell, the encoder and the mini-fan are all interfaced to the MQ3 card.

Experiments with a sampling frequency of 100Hz are built in MATLAB/Simulink and compiled for the Real-Time Windows Target using Real-Time Workshop. They are then executed on the hardware via the MultiQ3 (MQ3) interface board. The force cell has an independent closed-loop force controller that applies forces on the SMA wire using feedback from the Honeywell Sensotec force sensor. The austenite length of the SMA wire is hand measured with a micrometer while it is heated under zero-load. The encoder count is then divided by this length to yield strain. The Xantrex power supply is configured to operate in voltage-control mode and is controlled remotely via an analog output channel on the MQ3 card. The power supply outputs a current that heats the SMA wire. This current is measured via a built-in current monitor signal from the power supply, using an analog input channel on the MQ3 card. Voltage across the SMA wire is also measured independently using a separate analog input channel on the MQ3. All these signals are then fed back to the Simulink model for analysis. Resistance is calculated in software by dividing the voltage and current.
measurements. The mini-fan is placed directly in front of the SMA wire and is used to provide varying convection coefficients. It is controlled programatically from Simulink through an analog output channel on MQ3. A large fan is also used to provide stronger air flow. It is controlled manually.

Noise Filtering

Fig. 3 shows the measured current and calculated resistance using a 2Ω Dale power resistor as a test load. Power supply noise of relatively constant amplitude degrades the signal-to-noise ratio (SNR) for lower currents, resulting in unacceptable noise on the calculated resistance. This is made evident in Fig. 3 where the $R$ readings are noisiest at 0.1A.

![Fig 3: Current and Resistance VS Time for 2-Ohm Power Resistor](image)

After experimenting with several filtering techniques, we implemented an exponential weighted moving average (EWMA) as it gave us the best trade-off between noise rejection and delay introduced to the signal. The EMWA is governed by Equation (1) [7]:

\[
EWMA_k = \lambda Y_k + (1-\lambda)EWMA_{k-1} \quad k = 1,2,\ldots,n
\]

$Y_k = \text{current measurement}$

$\lambda = \text{smoothing factor}$

TEST PROCEDURE

Since the current required to actuate an SMA wire is dependent on ambient conditions, we cannot simply apply a fixed current every time and expect the wire to actuate. As such, a method is required to determine an appropriate current profile which will actuate the wire without danger of overheating. Given that our only feedback variable is $R$, we examined the behaviour of $R$ during actuation. As shown in Fig. 1, $R$ increases until it reaches a cusp and starts decreasing as the wire actuates. Note that the wire's contraction is minimal up to the cusp. Thus, we hypothesize that if we slowly ramp current through the wire and monitor $R$, we will be able to find the value of current that causes actuation in the present ambient conditions.
without causing too much contraction. Essentially, we are "probing" the wire to determine a current that will later be used to determine the priming and actuation currents.

The Probing Current

The probing current is a slow ramping current that is used to determine the current that causes wire actuation in the present environmental conditions. We call this current the maintenance current ($I_{mtn}$). $I_{mtn}$ serves the same purpose as the safe current used by Teh et al. [4], i.e. the actuation controller will switch to $I_{mtn}$ after $R$ has dropped below the actuation threshold. Hence, $I_{mtn}$ maintains actuation without risk of overheating the wire.

An example of the probing current is shown in Fig. 4 where $I_{mtn}$ is found to be slightly under 0.8A at approximately 68s. Notice that the probing current does cause some decrease in strain but does not cause the wire to fully actuate, which occurs during the actuation period at 90s. Notice also that the sharp drop in resistance following the cusp seen in Fig. 1 is again seen here around 93s and does accompany the transformation and wire contraction.

Sample Experimental Trial

For the experiments, all the trials used the same timing sequence, force profile, initialization procedure, and ending procedure. An example of an experimental trial is shown in Fig. 5. The sections labelled 1 to 6 in Fig. 5 are the initialization, probing, cooling, priming, actuation, and ending periods respectively. The cooling, priming, and actuation periods are repeated two more times to give three full actuation cycles per trial run.

Ambient Wind Conditions

Ambient air currents affect the wire heating by increasing the coefficient of convective cooling. Thus, for each of the priming strategies, we ran experimental trials under four different wind conditions: Fans Off; Mini-Fan at 50% Duty Cycle (MF50), Mini-Fan at 100% Duty
Cycle (MF100), and Large Fan at Highest Setting (LFHS). The wind speeds were measured with a Kestrel 1000 held-hand anemometer and were as follows: MF50 → 0.4 – 0.5m/s; MF100 → 1.1 – 1.2m/s; LFHS → 3.7 – 3.8m/s.

If a fan was used during a trial run, the fan was turned on after the encoder was zeroed in the initialization period and turned off after the force starts decreasing in the ending period.

**PRIMING AND ACTUATION STRATEGIES**

The experiments ran used 6 different priming strategies, which were labelled 50%*I_mtn, Minus 0.4A, dR/dt 1, Absolute R 1, dR/dt 2, and Absolute R 2. Since the cusp in resistance is identified with the value of I_mtn, the priming strategies are based on this current. We label the current during the priming period as I_prime. The first two strategies are purely heuristic, based on observations from experiments under different wind conditions. The other four build upon the first two by including PI servo controllers.

The idea of the servo controller is to servo around the cusp using either R or dR/dt measurements as feedback. Thus, as R drops below the cusp, the servo controller increases the value of I_prime to bring it back up. However, R decreases below the cusp in two situations: when the wire cools naturally without actuating, and when the wire actuates. In the latter case, increasing I_prime will not bring R back to the cusp. Consequently, this paradox of not knowing the reason behind the decrease in R forced us to use servo controllers that erred on the conservative side to guard against spurious actuation. In future work, controllers will include some heuristic decision making to accommodate this.

For comparison, experiments using the recommended safe current (*Safe Current*), and the actuation strategy with no priming (*No Priming*) were also carried out.

*Safe Current*

The manufacturer-recommended safe current for the SMA wire is 1A [8]. As such, with the experiments using *Safe Current*, the actuation periods consisted of running I_act=1A through...
the wire for 10s. While there is not any real priming in this test, during the priming period $I_{\text{prime}}=0.1\text{A}$ to provide a low-level current for resistance measurement.

**No Priming**

These experiments were run to serve as a benchmark to illustrate the effects of priming. During the priming period, $I_{\text{prime}}=0.1\text{A}$.

50%*$I_{\text{mtn}}$

The 50%*$I_{\text{mtn}}$ strategy ($I_{\text{prime}}=0.5I_{\text{mtn}}$) came about after noticing that at room temperature with the fans off, $I_{\text{mtn}}$ was approximately 0.8A. Under the same conditions, if the current was stepped up periodically by 0.1A rather than ramped, the wire would start to actuate slowly at 0.5A. Hence, we thought a priming strategy that used a constant current equivalent to 50% of $I_{\text{mtn}}$ would help prime the wire without the risk of spurious actuation.

**Minus 0.4A**

Although the 50%*$I_{\text{mtn}}$ strategy worked well with the fans off, the priming results seemed a little conservative when the fans were on. Hence, an absolute difference was examined where $I_{\text{prime}}=I_{\text{mtn}}-0.4$. The 0.4A offset was derived empirically and the results of this heuristic were very good in our wind conditions.

dR/dt 1

We augmented the 50%*$I_{\text{mtn}}$ strategy with the implementation of a PI controller to servo around the cusp where the theoretical value of $dR/dt$ is 0. Unfortunately, $dR/dt$ can also equal 0 at any $R$ value as long as the current is constant and not high enough to cause actuation. To overcome this problem, the current was first ramped until the resistance reached the cusp, and the servo controller was activated afterwards.

The ramp stops at the first occurrence of $dR/dt<=0$, after the current has reached 0.8$I_{\text{mtn}}$. This latter condition is introduced to avoid detection of negative gradients in the noise at low currents. Once the cusp is reached, the servo controller is initiated, with input $e = -dR/dt$ and the output is $I_{\text{prime}} = [K_p*e + K_i*\int e\,dt]$.

As a safety precaution against overheating and to prevent spurious actuation, the upper limit of $I_{\text{prime}}$ is set at $I_{\text{mtn}}$. The lower limit of $I_{\text{prime}}$ is set to 50% of $I_{\text{mtn}}$ since previous experiments showed that 50% of $I_{\text{mtn}}$ was a relatively conservative lower bound. Through experimentation, the values of $K_p$ and $K_i$ values that gave best performance were found to be 15 and 80 respectively.

**Absolute R 1**

Since it is difficult to servo on an inherently noisy signal and since the $R$ readings were relatively clean during high currents, it seemed logical to attempt to servo on an absolute $R$ value. From previous experiments, we found that 1.585Ω was a good value to servo upon without causing spurious actuation. Thus, $e=1.585-R$ and $I_{\text{prime}} = [K_p*e + K_i*\int e\,dt]$. With this setup, the $K_p$ and $K_i$ were both set to 10 and similar to $dR/dt 1$, the upper and lower limits of $I_{\text{prime}}$ were set to $I_{\text{mtn}}$ and 50% of $I_{\text{mtn}}$ respectively.
dR/dt 2

In this strategy, the lower limit for $I_{\text{prime}}$ during servoing was set to $I_{\text{mtn}}-0.4$, since this had shown in the Minus 0.4A trials to be a better lower bound for priming. With the exception of this change, $dR/dt 2$ is the same as $dR/dt 1$.

Absolute R 2

For completeness, the Absolute R technique was also combined with the Minus 0.4A lower bound for $I_{\text{prime}}$ which gave rise to the Absolute R 2 strategy.

Actuation Strategy

All the tests used the same actuation strategy. Through experimentation, the current was set to $I_{\text{act}}=1.25 I_{\text{mtn}}$ during the actuation period. However, once the wire has actuated, the current is lowered to $I_{\text{mtn}}$.

To determine a point where the wire would be considered as actuated, $R$ was again examined. For our wire, we found that 1.38Ω was a consistent threshold that was lower than the low-current $R$ (when the wire is cold) but higher than the austenite $R$ (see Fig. 1). As a result, our actuation strategy was to apply 125% of $I_{\text{mtn}}$ until $R$ was less than 1.38Ω and then switch to $I_{\text{mtn}}$ to maintain the actuation.

EXPERIMENTAL RESULTS

Several experiments were conducted to test the effectiveness of the different priming strategies. To compare the obtained results, two measures were defined: Time-to-Cusp (TTC) and Absolute Strain Time (AST). TTC is defined as the time from the start of the actuation period (if priming is used, then the wire is primed at this point) to the cusp. The shorter the TTC the more effective the priming strategy is (i.e., the closer the priming phase brought the wire to the cusp). AST is defined as the time needed to contract the SMA wire to 1% strain from the start of the actuation period. The goal is to obtain a consistent AST under all wind conditions. One should note that this measure favours the priming strategy that causes more pre-actuation contraction.

The TTC and AST are both functions of the average $I_{\text{prime}}$ (noted by $I_p$ in Fig. 6) and $I_{\text{act}}$ (noted by $I_{\text{act}}$ in Fig. 7). With TTC, a higher $I_{\text{prime}}$ will generally mean $R$ would be closer to the cusp prior to actuation and a higher $I_{\text{act}}$ would generally mean that $R$ will rise faster to the cusp after actuation. Since AST includes the TTC, it depends on both $I_{\text{prime}}$ and $I_{\text{act}}$. Furthermore, a higher $I_{\text{act}}$ would mean that the wire would contract faster leading to a faster AST.

Comparing the TTC across all wind conditions in Fig. 6, one can see that the Absolute R strategies are able to keep their TTC consistently faster than the others. This of course does not include the Fans Off case where the servo controller does not have much of an effect on the priming. Also, adding the tighter lower limit for $I_{\text{prime}}$ on $dR/dt 1$ to form $dR/dt 2$ also seemed to improve the TTC as the wind speeds increase.

Comparing the AST across all wind conditions in Fig. 7, one can see that the Absolute R strategies performed consistently well given the values of $I_{\text{act}}$. The exception is the AST for Absolute R 2 in MF50 but its $I_{\text{act}}$ was noticeably lower. Another trend that should be noted is that the ASTs decreased as the wind speed increased. This seems to suggest that the actuation strategy of using 125%*$I_{\text{mtn}}$ for $I_{\text{act}}$ is a little high. Hence, a lower percentage should
probably be used to try to keep the AST consistent across all wind speeds. Otherwise, if this trend continues, the AST will be even faster as the wind speeds further increase which suggests that the $I_{\text{act}}$ may be too high, which will risk overheating the wire. However, a lower percentage would mean a lower AST across all wind conditions which may or may not be acceptable for a given application. If the lower AST in not acceptable, then another actuation strategy will be needed since the present strategy cannot produce a faster yet consistent AST across all wind speeds without risking overheating the wire.

![Comparison of Time-to-Cusp Across All Wind Conditions](image)

**Fig. 6**: Comparison of Time-to-Cusp Across All Wind Conditions
CONCLUSIONS AND FUTURE WORK

From comparing the results of No Priming with those of the priming strategies, we conclude that priming does help the wire actuation. In addition, from examining the different approaches, we conclude that Absolute R 2 was the best priming strategy of the ones tested. Its tighter lower limit on I_prime helped improve over Absolute R 1 and its inherently cleaner R signal and simpler control scheme allowed it to outperform the dR/dt strategies.

With regards to the actuation strategy, we conclude that using 125%*I_mtn for I_act may be too high to produce consistent ASTs across the wind conditions. This was evident as the ASTs got faster as the wind speeds increased. This is a direct result of higher wind speeds resulting in higher measured I_mtn during probing, and higher resulting actuation currents.

In this work, we used varying winds as a proxy for varying ambient temperatures. However, we anticipate that if we rerun these same experiments under different ambient temperatures, the effect of priming will be greater. We are currently developing the setup to allow measurements in environmentally-controlled conditions across a broader range of temperatures (approximately -20°C to 40°C).

With an analog current driver being used in these experiments, it was found that the resistance measurements were quite noisy at low current values (c.f. Fig. 5). Our current setup will use a PWM-based current driver. SMA voltage and current are sampled during the ON-cycle of the PWM signal ensuring a high SNR.
REFERENCES

A FEEDBACK-BASED DYNAMIC INSTRUMENT FOR MEASURING MECHANICAL PROPERTIES OF SOFT TISSUES FOR MINIMALLY-INVASIVE SURGERY

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ABSTRACT

In this paper, a feedback-based dynamic instrument integrated into a MIS tool to provide mechanical properties of soft tissue is presented. The proposed dynamic instrument captures the resonance frequency shift using a phase-locked loop feedback system. Some important advantages of this method are that it is robust and simple in comparison to other similar instruments as it does not require magnitude information of plant’s displacement output and no force sensor is used; it works fast, and is capable of capturing dynamic mechanical properties of viscoelastic tissues, while most of the works are focused on only static/quasi-static elastic modulus.

Keywords: Minimally-Invasive Surgery (MIS), soft tissue stiffness measurement, resonance frequency shift, Phase-Locked Loop (PLL)
INTRODUCTION

Minimally-Invasive Surgery (MIS), where the surgeon operates through one or small incisions, has become popular due to its improved patient recovery as compared to open surgery. MIS typically involves use of laparoscopic devices and remote-control manipulation of instruments with indirect observation of the surgical field through an endoscope or similar device. This may result in shorter hospital stays, or allow outpatient treatment, reducing trauma, risk of inflammation, and postoperative complications [1,2,3].

However, MIS has several disadvantages including loss of tactile feedback. Performing the surgery with such impaired haptic information can lead to an increase in tissue trauma and vital organic tissue damage, and a reduced chance of detecting expected or unexpected tissue abnormalities. To restore this loss of information, instruments have been developed providing a sense of touch to the surgeon. The size and motion of these instruments are limited. They need to fit inside a cavity with a maximum diameter of 12 mm, and be sterilizable [2].

Tissue characteristics are changed by diseases, most significant of which is cancer responsible for many deaths worldwide. Tumours are generally harder than normal tissues making it possible to be detected by a tactile feedback. Unfortunately, the existing methods cannot detect the tumours accurately. For example, palpation cannot give quantitative measurements and is difficult to administer as it only relies on the surgeon’s expertise. Ongoing studies are improving soft tissue measurement techniques under such conditions as small/large deformation and linear/quasi-linear/nonlinear behaviour, noting that soft tissues exhibit challenging behaviours [4,5,6]. A brief review of relevant tactile sensors particularly adapted or adaptable to MIS is presented below.

Previously, apparatus for measuring the consistency, hardness, or stiffness of soft tissues by measuring the displacement and/or force (strain/stress) in the tissue has been proposed [3,7,8]. Ohashi et al. [9] used suction to deform the tissue instead of indentation. Several methods using one or more sensor arrays to form a tactile unit have also been presented [1,10]. Howe et al. [11] investigated relaying tactile information from surgical site to the surgeon by recreating the tactile stimulus directly on the surgeon’s finger tip. These methods require capturing accurate displacement/force data, which are also under influence of noise. Moreover, this requires higher quality components and assembly altogether increasing the cost of the instrument.

A number of studies suggest techniques based on frequency response analysis to detect tissue mechanical characteristics. These methods have been proposed incorporating vibration amplitude detection [12], open-loop phase detection [13], closed-loop resonance frequency detection [14], and phase-locked loop (PLL) based resonance frequency shift detection [15,16]. A set of studies introduce a phase shift circuit in a feedback network [17,18,19]. Valtorta and Mazza [20] proposed a tortional resonator device (TRD). In [21], a mechanical resonator is designed that allows for low resonance frequencies.
Several issues with these methods are that they are not designed specifically for MIS [13,14,15,16], are only based on simulations [21], cannot measure the viscosity of the tissue, or may introduce significant elasticity measurement errors because of the effective mass of the tissue and the relevant assumptions associated with their high resonance frequencies. To filter out the inertial parameter of the tissue, Murayama [17] has conducted experiments on silicones with different moduli, and calculated a calibration equation. The provided results present high errors in the measurement of actual Young’s modulus of soft tissues. Another common issue is capability of and optimization for measuring tissue properties in-vivo noting that the in-vivo mechanical properties are different from in-vitro, in-situ, or ex-vivo [3,22].

In this work, a feedback-based dynamic instrument will be integrated into a MIS tool to provide mechanical properties of a soft tissue to the surgeon, satisfying limiting conditions of MIS including the size, sterilizability, and ability to capture tissue behaviour in-vivo. The proposed instrument incorporates a resonance or near-resonance frequency shift (RFS) method. The advantages of this method are that it is robust and simple in comparison to other similar instruments as it does not require magnitude information of plant’s displacement output and no force sensor is used. Moreover, this method can directly measure viscoelasticity as an important property for understanding the conditions of tissue.

In the next section, the design of the proposed instrument is presented. The layout of the system is explained, and presented in state-space form. The process of selecting the parameters of the instrument is described, and finally the system controller is designed. The simulation and experimental results for the system are then shown in the subsequent section. The concluding remarks are presented in the final section.

**SYSTEM DESIGN**

**Design principles**

The frequency and phase characteristics of displacement of the contact point of a vibrating instrument changes when in contact with viscoelastic tissues. The RFS, $\Delta \omega_r$, is shown to be related to tissue Young’s modulus, $E$, while the bandwidth of the frequency response is related to viscosity.

$$\Delta \omega_r \propto E, \zeta = \frac{\Delta \omega_{3db}}{2\omega_r}$$

In order to determine the mechanical properties of tissues based on RFS, one approach is to drive the instrument to vibrate in a phase corresponding to system’s resonance frequency (or near it) with a feedback system. When the instrument is in contact with tissue, it will vibrate with a different frequency response. This difference is used to determine mechanical properties of tissue. Another approach, which is used to calibrate the instrument and optimize the response of the first approach, is to drive the instrument with an input that includes a set of frequencies in the desired range. The output frequency response is captured and analyzed, and consequently the viscoelasticity of the examined tissue is determined.

**System layout**
The proposed instrument, shown in Fig. 1, consists of two springs and masses combined together to form two resonance frequencies below 100 Hz. The springs and masses, actuator, and the sensors can be placed in different layouts; however, practical mechanical design issues limit the instrument’s design. The assembly of sensors and actuator, their wires and other electrical components must be considered which lead to this design, similar to the layout considered in [21]. The springs and masses are connected to the actuator to create two low resonance frequencies, and the LVDT sensors are set to measure the displacement of the masses.

![Fig. 1- Design layout for MIS instrument](image)

**Piezoelectric actuator model**

The piezoelectric actuator integrated in the instrument helps minimizes the size and improves positioning performance. It provides an accurate and strong actuation, allowing the instrument to capture the results more reliably.

In order to analyze the behaviour of the instrument more accurately and optimize the system parameters, a simple model for the piezoelectric actuator is considered. The first governing equation of a piezoelectric material can be simplified to be used in a stack actuator, which essentially is composed of $n$ layers of a single-layer actuator. Ideally, a stack actuator only deforms in the axial direction, so a one dimensional governing equation can be used to define stack actuator’s behaviour.

\[
\Delta L = -\alpha F_{\text{produced}} + \beta V
\]

where

\[
\alpha = 1/k_p = s_{E33}L/A
\]

\[
\beta = \frac{F_{p0}}{V_{pm}k_p} = nd_{33}
\]

$L$ is the length of actuator, $A$ is the cross-section of actuator, $V_{pm}$ is maximum possible voltage applied to actuator. $F_{p0}$ is maximum blocked force, and $k_p$ is piezo actuator’s stiffness. A static model of the stack actuator is equivalent to an active spring whose displacement is linearly dependent on the applied force and voltage. For a dynamic response, the mass of the stack, $m_p$, and a damping coefficient, $c_p$, are also considered.

**State-space model**

The lumped parameter model of the system is represented in Fig. 2. When in contact with a tissue, the instrument has a different state-space model generating the RFS. It is desirable to be able to apply these changes into the model in a way that the tissue components are distinguishable. This is met using a Voigt model [5] with mass. The state-space model in this
case is given below. When \( k_m = c_m = m_m = 0 \), the instrument is working in non-contact setup.

\[
\dot{x} = Ax + Bu, \tag{5}
\]

where

\[
A = \begin{bmatrix}
0 & 0 & 0 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 \\
-\frac{k_1 + k_2}{m} & \frac{k_2}{m} & \frac{k_1}{m} & -\frac{c_1 + c_2}{m} & \frac{c_2}{m} & \frac{c_1}{m} \\
-\frac{k_2}{m} & -\frac{k_2 + k_m}{m} & 0 & -\frac{c_2}{m} & \frac{c_2 + c_m}{m} & 0 \\
-\frac{k_1}{m} & 0 & -\frac{k_1 + k_p}{m_p} & \frac{c_1}{m_p} & 0 & -\frac{c_1 + c_p}{m_p}
\end{bmatrix}, \quad
B = \begin{bmatrix}
0 \\
0 \\
0 \\
0 \\
\beta \\
\frac{m_p \alpha}{m_p}
\end{bmatrix},
\]

and

\[ m_0 = m_{tip} + m_m \]

Fig. 2- Lumped-parameter schematic of the system.

**Parameter selection and optimization**

The operating frequency range of the device is selected to approximately lie within 25 to 100 Hz. At very high excitation frequencies, the materials tends to act more like an elastic solid while at very low frequencies, it is very hard to filter out disturbances especially surgeon’s hand tremor. The normal hand tremor is assumed to be less than 0.5 mm for a range of frequencies below 10 Hz [23]. The resonance frequency of the piezoelectric actuator is too high, so additional masses and springs are necessary to tune the resonance frequency to the desired value. A group of studies, e.g. [17], used a very high-frequency resonator which required the tip mass of the instrument to be very small in order not to be influenced by the effect of the mass of the tissue for stiffness measurement.

A number of criteria and constraints are used to select the value of system parameters for best performance. The resonance frequency shifts should be big enough to be detectable by feedback system, and the ratio of the frequency shifts to the resonance frequencies should be as high as possible. The relationship between the first RFS and instrument’s mass and spring parameters is shown in Fig. 3. Full phase information should be available for both resonance frequencies; i.e., the phase of the transfer function of the system, \( G_i(\omega) \), should span \( R_\phi \), where,

\[
R_\phi = [\angle G_i(j\omega_r) - 45^\circ, \angle G_i(j\omega_r) + 45^\circ] \tag{6}
\]
The displacement of each LVDT sensor must remain in the sensor’s range. This displacement is affected by the actuator, the surgeon’s hand when interacting with a tissue, and the force of gravity on the tip and proof mass. The selected sensors, Schaevitz 050-MHR, have a displacement range of ±1.27 mm. The phase and displacement characteristics of the system are mainly affected by the damping of the system, and depend on proper design and fabrication. Therefore, the system parameters are selected in a manner to ensure a satisfactory performance.

![Figure 3: Contours of RFS (Hz) with respect to the instrument’s mass and spring parameters.](image)

One of the critical parameters of the instrument is the viscous damping coefficient. This damping factor influences the dynamic response of the system such that the phase corresponding to the resonance frequency changes. The general damping of the system also affects the amount of the displacement of the masses of the system. Therefore, bench-top experiments have been conducted to measure these factors in the springs. The Fast Fourier Transform (FFT) method was used to find the total damping ratio of a second-order system, while the viscous and coulomb damping coefficients of the springs were separated in a time-domain analysis. For a second-order system, the damping ratio of the system with a sine-wave input is approximated by (1). Similar analysis holds for a system with initial conditions \(x(0) = 0\) and unknown \(x(0)\), which again reduces to the approximate equation (1). For the time-domain analysis, the signal was filtered using a bandpass filter. The overlapping curve to the signal is derived, and fit to the following curve.

\[
F(x) = b_1 e^{b_2 x} + \begin{cases} 
 b_3 x + b_4 & \text{if } b_3 x + b_4 > 0 \\
 0 & \text{if } b_3 x + b_4 < 0
\end{cases}
\]  

(7)

**Controller design**

To measure the desired properties of a tissue, a fast and robust approach based on Phase-Locked Loop (PLL) is proposed. A PLL is an example of a feedback system that satisfies a relationship between the phase of the generated and reference signals. One of the most significant advantages of a PLL system is its noise rejection capabilities. For this application, the feedback system drives the device at its resonance frequency, near resonance, or the corresponding 3-dB frequencies in either non-contact or in-contact setups. The proposed method also ensures a smooth result and eliminates a risk of unexpected behaviour for the PLL system if one of the displacements gets out of the range of the sensor.
For driving the system in its resonance frequency, it is required to set the phase of the system to a phase corresponding to the resonance. In order to capture the bandwidth of the system, the PLL needs to be compared with ±45 degrees shifted phases. An overview of the PLL-based feedback system is shown in Fig. 4. A type-2 PLL is required to obtain a zero-error response, while a type-1 PLL provides a phase that satisfies (8). Therefore, a type-1 PLL can only be used to drive the system in a near-resonance frequency.

\[
\begin{align*}
type1: & \quad PD_{(f_0 + \Delta_f)} \times K_{\text{VCO}} = \Delta_f, \\
type2: & \quad PD|_{f=f_r} = PD_r,
\end{align*}
\]

where \( PD \) represents the output of Phase Difference detection unit, \( PD_r \) is the phase difference occurring at resonance frequency.

In order to obtain an optimized performance, the loop controller of PLL needs to be designed carefully. In order to filter out the disturbances in the system, including PLL and electronic noise and hand tremor of the user, bandpass and lowpass filters are incorporated before and after PD respectively. The loop controller design will be fully explored in subsequent prototype testing.

![Diagram](image.png)

**Fig. 4- System overview**

**SIMULATION & EXPERIMENTAL RESULTS**

MATLAB/SIMULINK was used for simulation. The Bode plot of the outputs of the system is shown in Fig. 5. It is illustrated that the frequency response of the system changes because of the presence of the mechanical impedance of the soft tissue.

The summary of the experiments in order to find the damping ratio of the springs is shown in Table 1. Although the frequency and time-domain results are conducted separately, the results are compatible. The calculated range of the damping ratio is within desired range for the system design.

<table>
<thead>
<tr>
<th></th>
<th>Mean</th>
<th>Variance</th>
<th>Min</th>
<th>Max</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency-domain</td>
<td>0.025</td>
<td>0.00001</td>
<td>0.0099</td>
<td>0.065</td>
</tr>
<tr>
<td>Time-domain</td>
<td>0.032</td>
<td>0.00003</td>
<td>0.0138</td>
<td>0.083</td>
</tr>
</tbody>
</table>
The resonance frequency shift is detected using the PLL-based feedback system, simulated with a moving probe along a soft material. The instrument is first initialized. It is then put into contact with a material consisting of two different sections with different stiffness values. Finally, the instrument is moved from the less stiff side of the material to the stiffer side gradually. A sample result is illustrated in Fig. 6. The required initialization time for the instrument is about 1.2s, and the rising time of the feedback system for a step input is equal to about 0.1s. When the instrument is in contact with a material with a stiffness value approximately equal to 70 (N/m) or equivalently with a Young’s modulus of 6 kPa, the first and second resonance frequencies change about 4% and 0.75% respectively. The resolution of the feedback system, however, is much better. Therefore, the frequency shifts in the instrument can be detected both fast and accurately.

Fig. 5- Bode plot of the system outputs with/without contacting a material
CONCLUSION

A new dynamic minimally-invasive-surgery instrument for measuring mechanical properties of biological soft tissue using a PLL-based feedback system is proposed in this paper. It is shown that it satisfactorily detects the resonance frequency shift. The resonance frequency shift is approximately linearly proportional to the stiffness of the material under indentation. A look-up table can be used to relate the RFS and stiffness values. The system is robust to noise, does not require accurate displacement information, and can determine the stiffness of a soft tissue both accurately and fast enough for MIS purposes.

REFERENCES


EFFECT OF DIFFERENT ADHESIVES ON STRENGTH, ENERGY ABSORPTION AND DAMPING PROPERTIES OF BONDED ALUMINUM STRUCTURES FOR THE TRANSPORTATION INDUSTRIES

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ABSTRACT

The use of aluminum bonded structures is growing with the need for lighter cars and trucks and cheaper assembly techniques. Several adhesive manufacturers propose a broad range of structural adhesives from very strong and stiff epoxies to flexible urethane products. Despite the documentation available, the use of adhesive in the ground transportation especially for structural assemblies still requires knowledge reinforcement.

In this paper, three major types of adhesives, epoxies, acrylics/methacrylates and polyurethanes are first tested using standard samples to establish their properties. Several parameters related to the surface preparation, adhesive application, adhesive thickness are evaluated. Simple structural assemblies made of bonded aluminum extrusions and sheets are produced and tested combined with rivets for energy absorption under large deformations. Finally, dynamic analyzes are carried out to establish the impact of adhesives on vibration and noise attenuation.

Keywords: Aluminum structures, bonding, energy absorption, vibration damping.

INTRODUCTION

Over the last thirty years, adhesives have been increasingly applied for structural assembly. More recently, car manufacturers have used them in combination with spot welds and rivets to increase stiffness and impact energy absorption notably in chassis and body-in-white structural assemblies [1-4]. With the use of aluminum, riveting, MIG, TIG, laser and friction stir welding techniques have been preferred over resistance spot welding and bonding is playing a key role by reinforcing and distributing the stress over a larger surface. Indeed, because aluminum is softer than steel and in reason of thinner walls of many components it is even more important to
better spread the stress over a larger surface to optimize the strength and stiffness and reduce the weight. Some manufacturers of sport cars are testing or began to use adhesives as their main technique for chassis assembly. Other important reasons explaining the widespread utilization of adhesive in automobile and aircraft assembly include (1) the reduction of vibrations and noise, (2) the reduction of galvanic corrosion (e.g. steel and aluminum), (3) its capacity to join dissimilar materials (e.g. plastic and aluminum) and most importantly, (4) the assembly cost that is often lower compare to other techniques.

Although increasingly used for automobile and aircraft assembly, bonding techniques are still slowly emerging in assembly of commercial vehicles or has even failed in other industries. Among the main reasons are: (1) a lack of understanding of the techniques, (2) products poorly designed for bonding, (3) inadequate adhesives and surface preparation methods, (4) degradation of adhesives and lack of durability. The simple replacement of other joining techniques by bonding without adapting the design and making proper calculation and tests has little chance of success.

This paper is mainly intended to research, design and production engineers who want to better understand the use and benefits of adhesives for structural assembly of aluminum components. Its content involves (1) the characterization of different adhesives covering a wide range of structural assembly applications, (2) the study of the impact on energy absorption of assemblies using rivets and/or adhesives, (3) an investigation of the effect of adhesives on vibration reduction and (4) some design recommendations.

**CHARACTERIZATION OF ADHESIVES**

A wide range of products are available for structural assembly. It goes from very stiff epoxies to flexible polyurethanes, and includes acrylics and methacrylates and many other products. These products have specific mechanical properties (e.g. strength, stiffness, ...), durability and environment resistance and proposes specific methods for surface preparation, adhesive curing and thickness of joint. For engineers and designers the selection of proper adhesives and the design adaptation for its application require an extensive knowledge of all these aspects. For example, for a car space frame assembly, it is essential to adapt significantly the design of interfaces, consider the geometric errors in the components and assembly which will determine the gap for the adhesive, define how the surface will be prepared, and consider also the dynamic aspect, the stiffness and energy absorption during a collision.

The first source of information is obviously the manufacturer’s technical sheet. They generally include useful information such as operating temperature and humidity condition, environment resistance, preparation method and indication about the curing. They also give a general overview of the properties of the product, often including shear strength, as tested in laboratory using specifically controlled conditions sometimes more difficult to achieve in industries. Automotive and aerospace industries use extensive facilities especially for surface preparation and curing to achieve optimal performances.

Because of the wide range of conditions and applications found in industries, tests must be carried out. Standard ASTM tests are generally utilized to compare a pre selected set of adhesives and surface conditions. As shown in Figure 1 these include notably single [5] or
double lap shear test [6], the tensile strength test [7], the cleavage test [8] and the peel test [9].

Even if they are very useful to establish the performance of adhesives in specific conditions, the samples are of very simple geometry and tests are carried out under quasi static loading. In practice, complex geometries, irregular surface, geometric and positioning errors and dynamic loading are generally observed.

Characterization of three types of adhesives

Table 1 presents the general characteristics of three types of adhesives popular in the transportation industry [3, 4, 10]: epoxies, acrylics and polyurethanes. Although many other products are available, this study is limited to adhesives that cure at room temperature and humidity. Within each type of adhesive several formulations with different properties exist.

The epoxies have very high strength and are very stiff. However, their properties vary considerably with the adhesive film thickness. A very thin film (0.1 to 0.2mm) is required for optimal performances. Their use is recommended where consistent film thickness can be achieved for example with accurate mating surfaces or with sheet panels that are spot welded or riveted which also hold the parts together during setting. In reason of the thin film and their high strength, the amount of adhesive required is very small.

Polyurethanes are much more flexible but very significantly weaker. The film is generally much thicker (1 to 4mm) but their strength does not vary much with thickness. This adhesive is pertinent for application where components are not very accurate and where good isolation between components is required. A good example is for the assembly of truck box panels and bus sheet panels on structural members made of rolled or extruded aluminum. A gap of 1 or 2 mm can easily be tolerated by the adhesive. Because this type of adhesive is much weaker than epoxies, larger surfaces must be covered. A much larger volume of adhesive is also required.

Acrylics including methacrylates stand between epoxies and polyurethanes. They are less flexible but stronger than polyurethanes and more flexible and weaker than epoxies.
<table>
<thead>
<tr>
<th></th>
<th>Epoxy 2 components</th>
<th>Acrylic (Methacrylate)</th>
<th>Polyurethane 1 component</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shear strength (MPa)</td>
<td>12 - 35</td>
<td>20 - 30</td>
<td>2.5 – 15</td>
</tr>
<tr>
<td>Rigidity</td>
<td>rigid</td>
<td>average</td>
<td>flexible</td>
</tr>
<tr>
<td>Film thickness (mm)</td>
<td>0,1 - 0,5</td>
<td>0,1 – 0,5</td>
<td>0,1 – 5</td>
</tr>
<tr>
<td>Surface preparation</td>
<td>extensive</td>
<td>low</td>
<td>moderate</td>
</tr>
<tr>
<td>Operating temperature</td>
<td>-50 to 120</td>
<td>-70 to 120</td>
<td>-100 to 80</td>
</tr>
<tr>
<td>Water resistance</td>
<td>excellent</td>
<td>good - excellent</td>
<td>excellent</td>
</tr>
<tr>
<td>Oil resistance</td>
<td>excellent</td>
<td>excellent</td>
<td>good</td>
</tr>
<tr>
<td>UV resistance</td>
<td>excellent</td>
<td>excellent</td>
<td>good</td>
</tr>
<tr>
<td>Impact resistance</td>
<td>good</td>
<td>good - excellent</td>
<td>excellent</td>
</tr>
<tr>
<td>Fatigue résistance</td>
<td>excellent</td>
<td>excellent</td>
<td>excellent</td>
</tr>
<tr>
<td>Shrinkage</td>
<td>very low</td>
<td>low</td>
<td>moderate</td>
</tr>
<tr>
<td>Peel resistance (N/cm)</td>
<td>low 20-40</td>
<td>moderate 100</td>
<td>good 50-60</td>
</tr>
</tbody>
</table>

Table 1: Overall characteristics of adhesives (manufacturer and literature data)

Testing conditions. For all adhesives, the easiest and most industrially applicable but not necessarily the best method of surface preparation suggested by the adhesive manufacturer was generally used. The adhesive was applied manually within about ½ of the maximum deposition time recommended. For all ASTM tests, samples made of 6061-T6 sheets and bars and during setting, held in specially made fixtures in order to assure constant film thickness and proper sample alignment.

For two-components epoxies, manufacturers recommend, in order of increasing efficiency, (1) cleaning with a good degreasing agent, (2) mechanical abrasion or (3) chemical etching. The method chosen for testing this adhesive will be to first degrease with methyl ethyl ketone (MEK), second chemical etching in an alkaline solution of 10% Sodium hydroxide (NaOH) at 60°C for 30sec and then rinse in running water for 2min and free air dry. The two components where fully mixed prior to application on the sample.

Polyurethanes are provided in a single component. It can be applied in bond line thickness up to 10mm and cure in 7 days, thinner layer will cure more rapidly. The surface preparation comprises first a cleaning of all surfaces with a glass cleaner, followed by the abrasion of the surfaces with a Scotchbrite™ pad before cleaning again with glass cleaner. Finally, an adherence promoter or activator is applied with a clean absorbent paper and let to dry for 15min.

For acrylics and methacrylates the surface preparation consists in cleaning all surfaces with MEK to remove grease and dirt and optionally to abrade the surface with Scotchbrite™ and cleaned again with MEK. Single component acrylics utilized an activator deposited to the surface before applying the adhesive in layers typically under 0.5mm thick. Two components methacrylates were mixed prior to application on the surface in thicker layers.

Results. Tables 2, 3 and 4 presents respectively the results obtained for double lapped shear strength, tensile and cleavage tests. Announced strength is provided by the manufacturer. The measured strength, adhesive thickness, flexibility and rupture mode in these tables are experimental observations made on at least two samples. In table 2, we can see that most tests
do not reach the announced strength which was probably obtained in optimal laboratory conditions using the best surface preparation and the optimal adhesive thickness.

<table>
<thead>
<tr>
<th>Adhesive</th>
<th>Surface preparation</th>
<th>Announced strength Mpa</th>
<th>Measured strength Mpa</th>
<th>Adh. thickness mm</th>
<th>Elongation mm</th>
<th>Rupture mode</th>
<th>Comments</th>
</tr>
</thead>
</table>
| Epoxy    | MEK, NaOH 60°C, 30 sec | 25                     | 21.6                  | <.18             | 1.1           | Mixed        | - The central plate was 2.5 mm thick  
- The adhesive bond line must be thin to avoid adhesive failure  
- Sensitive to the surface preparation |
| Plexus   | MEK                 | 16 - 19.6              | 10.8 - 11.3           | .254             | 1.15          | Cohesive     | - Constant properties, even with little surface preparation  
- Always cohesive failure |
| Permanbond | MEK               | 20 – 25                | 17.8                  | .127             | 4.7           | Cohesive     | - Constant properties, even with little surface preparation  
- Always cohesive failure |
| Acrylic  | MEK, scotchbrite (SB), MEK | 15 – 25               | 7.8                   | .127             | 1.1           | Adhesive     | - Difficult to obtain cohesive failure with the minimal surface preparation recommended by the manufacturer |
| Polyurethane | GlassCleaner (GC) , SB, GC, Aktivator (Akt.) 20 min | 2.5                  | .99                   | 1.5              | 6.6           | Cohesive     | - With the recommended preparation method, total cohesive failure is rarely obtained  
- Constant cohesion failure  
- Air bubbles easily are trapped in the adhesive layer |
| SikaFlex | GlassCleaner (GC) , SB, GC, Aktivator (Akt.) 20 min | 6                    | 3.4 - 3.7             | 1.5              | 3.8-12        | Cohesive     | - With the recommended preparation method, total cohesive failure is rarely obtained  
- Constant cohesion failure  
- Air bubbles easily are trapped in the adhesive layer |

Table 2: Tests results of the double lap shear strength test (ASTM D 3528). Announced strengths from adhesive manufacturers are generally for AL2024-T3 and not AL6061-T6
**Table 3**: Results of the tension strength test (ASTM standard D 2095)

<table>
<thead>
<tr>
<th>Adhesive</th>
<th>Surface preparation</th>
<th>Measured strength Mpa</th>
<th>Adh. thickness mm</th>
<th>Elogeation mm</th>
<th>Rupiture mode</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Epoxy</td>
<td>Araldite 2012</td>
<td>MEK, NaOH 60°, 30 sec</td>
<td>34,2</td>
<td>0,089</td>
<td>0,4</td>
<td>Cohesive - sensitive to preparation method</td>
</tr>
<tr>
<td></td>
<td>3M DP 420</td>
<td>MEK, NaOH 60°, 30 sec</td>
<td>23,6-25,8</td>
<td>0,127</td>
<td>0,3</td>
<td>mainly adhesive -(note: oct08 cohesive failure and 43.6 Mpa)</td>
</tr>
<tr>
<td></td>
<td>Plexus MA 832</td>
<td>MEK</td>
<td>15,2-16,1</td>
<td>0,241</td>
<td>0,3</td>
<td>Cohesive  - always cohesive</td>
</tr>
<tr>
<td></td>
<td>Loctite H8000</td>
<td>MEK</td>
<td>16,1-19,1</td>
<td>0,065</td>
<td>0,3</td>
<td>Cohesive  - always cohesive</td>
</tr>
<tr>
<td></td>
<td>LORD 406</td>
<td>MEK, SB, MEK</td>
<td>16,6-17,2</td>
<td>0,213</td>
<td>0,3</td>
<td>Cohesive  - always cohesive</td>
</tr>
<tr>
<td>Acrylic</td>
<td>Perma-bond TA 435</td>
<td>MEK, SB, MEK</td>
<td>3,1-3,4</td>
<td>0,109</td>
<td>0,13</td>
<td>Adhesive  - very difficult to obtain 100 % cohesive failure</td>
</tr>
<tr>
<td>Polyurethane</td>
<td>Sika Flex 252</td>
<td>GC, SB, GC, Akt. 20 min</td>
<td>1,6</td>
<td>1,35</td>
<td>2,6</td>
<td>Cohesive  - in this mode of loading, cohesive failure is obtained with the manufacturer recommendations</td>
</tr>
<tr>
<td></td>
<td>Sika Force 7550</td>
<td>GC, SB, GC, Akt. 20 min</td>
<td>4,0-4,5</td>
<td>1,21</td>
<td>0,5-0,8</td>
<td>Cohesive  - Air bubbles are present in the adhesive layer</td>
</tr>
</tbody>
</table>

**ENERGY ABSORPTION UNDER LARGE DEFORMATIONS**

The intelligent design of lighter aluminum structures must also consider the absorption of energy during collisions. Because combined assembly modes are often used, this investigation will consider samples assembled using rivets, adhesive and combined rivets and adhesives. The adhesive utilized is the DP420 epoxy from 3D. First, ASTM double lap shear strength tests were carried out on samples assembled with (1) adhesive alone, (2) two 3.125x7.9mm aluminum rivets and (3) using both rivets and adhesive. For the samples assembled with rivets shown in Figure 2, these rivets were located 12.7mm apart and in the center of the bonding area of 9.53mm wide, i.e. 4.76mm from the edge.

![Fig. 2: Double lap shear strength samples with rivets](image-url)
The results of shear tests are shown in Table 5. Because there is an initial gap of 0.1 mm in diameter between the holes and the rivets and because the rivets and plates deform prior to failure, rivets fail after 2.3mm of global elongation with an equivalent shear strength of 12.7 MPa over the contact face or 5195.3N. The epoxy fails after only 1.1mm of elongation and a force of 8778.2N thus much before the rivets. With combined rivets and adhesive, part of the bonded surface is replaced by the rivets. This and the different of elongation before failure explains why the adhesive plus rivets is somewhat weaker than adhesive alone. For a better design, we should eventually select an adhesive with slightly higher elasticity. A positive impact of adding rivets to adhesive is that rivets can hold firmly the components together during adhesive setting.

Finally 6 samples of tubular structure made of two “U” bent sheet components have been assembled in combinations similar to the samples of Table 4. These tubular assemblies are illustrated in Figure 3. Where applicable, four 3.97x9.53mm aluminum rivets have been used per tube.
Crushing of the tube is realized on a 75tons servo-controlled hydraulic press. The force to displacement curve and more specifically the area under the curve will give the energy absorbed by the tube. One set of three tubes including a bonded tube, a riveted tube and a combined bonded and riveted tube, has been crushed in the vertical position as illustrated in Figures 3b and 4a. The second set of three tubes has been crushed in horizontal position as shown in Figure 4b. This latter figure illustrates also the crushing of the tubes in both orientations.

**Fig. 3:** Tube assembly (a) design made of two “U” bent sheets of 1.59mm (b) press apparatus for crushing the tubes and monitor the force and displacement.

**Table 5:** Effect of rivets of the double lap shear strength

<table>
<thead>
<tr>
<th>Assembly method</th>
<th>Surface preparation</th>
<th>Measured force N</th>
<th>Elongation mm</th>
<th>Rupture mode</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adhesive 3M DP 420 + 2 rivets</td>
<td>MEK, NaOH 60°, 30 sec</td>
<td>7721.6</td>
<td>1.1 mm to adh. failure 2.3 mm to rivet failure</td>
<td>Mixed, Rivets are sectioned in each shear area</td>
<td>- After the adhesive failure, the samples sustain stress up to 8.23 Mpa or 3344N before the rivets broke</td>
</tr>
<tr>
<td>Adhesive 3M DP 420 2 rivets</td>
<td>MEK, NaOH 60°, 30 sec</td>
<td>8778,2</td>
<td>1.1 mm to adhesive failure</td>
<td>Mixed adhesion/cohesion</td>
<td></td>
</tr>
<tr>
<td>2 rivets</td>
<td></td>
<td>5195.3</td>
<td>2.6 mm to adhesive failure</td>
<td>Rivets sheared</td>
<td></td>
</tr>
</tbody>
</table>
Fig. 4: Crushing results (a) progressive crushing of a riveted tube oriented vertically (b) riveted tube lying horizontally prior to crushing, (c) riveted tube after some vertical force which sheared all 4 rivets without deforming the U bent component, (d) riveted and bonded tube after crushing.

Fig. 5: Crushing results in vertical and horizontal positions
The crushing results are shown in Figure 5. In vertical position the area under the force to displacement curve expresses the amount of energy absorbed by the tubes. In vertical position, the bonded and bonded+riveted tubes sustain a maximum force at the beginning of deformations. Rivets of the riveted tube failed partly at the beginning and also after 60 mm. Although relatively similar before 60mm, the energy absorbed after 60mm is much higher for the bonded and riveted tube where part of the bounding and half of the rivets last over the entire stroke. In horizontal position, all four rivets of the riveted tube sheared after only 5mm of press motion. The bonded tube sustained again a higher force but failed on one side with 4 mm and on the other side after about 13 mm. The bonded and riveted tube is by far the best by absorbing more than twice the energy of the bonded tube and many times the energy of the riveted tube. Again, part of the bond and most rivets sustain the entire stroke. The crushing process illustrates very well all the stressing modes occurring over the stroke on the bond joint. For example in horizontal position, shearing is the first mode and cleavage appears in second as the lateral lips bend. Although very illustrative, this example shows all the complexity and the need to better understand and optimize structural designs especially where thin wall and buckling occurs, i.e. in most vehicle structures.

VIBRATION AND NOISE REDUCTION

One advantage of using adhesives is their capacity to increase stiffness and attenuate noise and vibrations. As reported in [2] stiffness in car space frame can be increased by 15 to 30% providing higher modal frequencies of somewhat lower magnitude. Two phenomena can be observed. First, by joining larger surfaces, the adhesive increases the stiffness of assemblies especially where thin walls are involved. This also increases the modal frequencies of the assemblies. Generally higher frequencies also mean lower vibration magnitudes and audible sound levels. Secondly, other researchers have investigated the additional damping due to the adhesive (i.e. absorbed energy in the bond line) when vibrations occur [11]. Using FE simulations, it was observed that modal frequencies and amplitudes change very little as the adhesive properties are modified (see also [12]). The most important effect seems to come from the joining area and the increase of stiffness.

To better understand the effect of adhesive on vibration attenuation, a first series of impact tests have been carried out on the assembled tube of the previous section (before they were crushed!). The apparatus is illustrated in Figure 6. A PCI 353MM77 accelerometer located inside the tube on the bottom face was used for receptance. The impact force was monitored using a B&K8200 accelerometer installed on the hammer. A small impact force was applied on the top surface at the opposite of the receptance accelerometer. Both signals are received by a HP3560A dynamic analyzer and downloaded to a computer for further data processing. Finally,
the tubes were supported by rubber bands inside the top corner in order to have negligible effect on the dynamic response. Several impacts have been tested and only repeatable signals are illustrated in Figure 7 for the riveted, bonded and combined riveted and bonded tubes. The time and spectral responses of the riveted tube shows two harmonics around 500 Hz which generate beats. The magnitudes at low frequencies are much higher. The adhesive used in bonded and bonded+riveted tubes reduces considerably the peaks at low frequencies but generate some smaller high frequency harmonics.

Fig. 7: Dynamic results with an impact hammer

CONCLUSION AND RECOMMENDATIONS

The performance of the adhesives proposed in the paper could in many cases be improved by used of more sophisticated surface preparation, automated application of adhesive or use an optimal thickness for example. However, these results also provide the forces and weaknesses of different adhesives as applied on AA6061-T6 using conditions that will usually take place in industry. The different mechanisms tested (shear, tensile, cleavage) appear in most
applications. Peeling results tested in a previous project and not reported in this paper should be avoided when designing components and assemblies.

An intelligent design of structural assemblies sometime needs to consider larger deformations or impact resistance. Again adhesives especially as combined with other assembly methods can improve significantly the absorption of energy. A good structure should consider the deformation and adjust the design to maximize the effect of adhesives.

Finally, adhesives are useful to increase stiffness and reduce vibrations especially at low frequencies. In the range of properties of the adhesives tested previously, their effect on vibration reduction comes mainly from the surface area covered and the increase of stiffness and not much from the adhesive properties. However, as for other design requirements, it is important to avoid weak bonding areas which could under vibrations, break progressively.

ACKNOWLEDGMENTS

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REFERENCES

TEMPERATURE MEASUREMENT OF SHAPE MEMORY ALLOY WIRES WITH SPOT WELDED THERMOCOUPLES

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ABSTRACT

Shape memory alloys (SMA), materials which convert heat energy (usually through Joule-heating) to mechanical energy, provide compact and effective actuation for a variety of mechanical systems. They are attractive options to be used in an automotive context as lightweight, scalable actuators that have a very high power/weight ratio. However, the materials must be protected from overheating during actuation. This leads to the need for direct temperature measurement methods for use either in direct temperature feedback controllers or indirectly for validating models of the material’s thermo-electric behaviour. Developing a proven experimental method for measurement of wire temperature is the goal of this work.

Various methods were applied and tested, including contact methods using thermocouples and thermistors, as well as non-contact infrared thermal imaging. The latter two are briefly described, while the paper focuses on our results achieved using thermocouples. Several different methods of thermocouple attachment are also investigated: spot welding, adhesive fixturing, and mechanical contact methods. A series of experiments and theoretical calculations conducted with 500\textmu m diameter Flexinol wire showed that the spatial offset of the thermocouple leads and the resulting ohmic drop due to current flowing in the wire causes measurement errors in the thermocouple readings. Experimental and analytical methods are proposed to compensate for these errors, resulting in a set of recommendations for thermocouple-based temperature measurement of current-carrying wires.

Keywords: Shape Memory Alloy, Temperature Measurement, Thermocouple, Actuator.
INTRODUCTION

Shape memory alloy (SMA) actuators can be an ideal substitution for more traditional actuators (e.g., magnetic solenoid) due to their unique “shape memory” property. Light, scalable and having a high power/weight ratio, these actuators are promising in a variety of mechanical systems, particularly in the automotive industry. SMA actuators rely on a reversible, thermally-driven phase transformation which occurs as the alloy temperature is cycled between approximately 30°C and 100°C. The difference in mechanical properties of the two material phases can be used to do mechanical work. Notably, the NiTi SMA commonly used for actuation has a relatively high electrical resistivity, enabling the design of electro-mechanical actuators using SMA. An actuator design typically comprises a biased wire made of SMA, which contracts in the presence of an electrical current, and expands as it cools when the current is removed. The actuator behaviour, then, can be roughly divided into the thermo-electric heating response which converts electrical to thermal energy and generates the phase transformation, and the thermo-mechanical phase transformation which causes the motion.

Research on SMA actuators has focused primarily on modelling and compensating for the nonlinear thermo-mechanical behaviour of the material, with relatively little attention being brought to bear on the heating aspects. This, despite the fact that the material must be protected from overheating during actuation in order to avoid degrading or destroying performance. This lack of attention is perhaps due in part to the inherent difficulty in measuring the temperature of the SMA wires: they are typically less than 500μm in diameter and can carry currents on the order of several amperes.

In this work, we describe our efforts to experimentally investigate appropriate temperature measurement techniques, and the development of reliable thermocouple-based methods. Our end use for the measurements is in the laboratory validation of thermo-electric models for SMA wire heating, so a reasonable level of accuracy and precision is desirable. In addition, however, the eventual approach may also be used in direct temperature measurement in an end application, for direct temperature control of the SMA actuator.

The Background section describes the problem as evidenced by the efforts of other researchers in the area, and briefly describes our investigations with thermistors and non-contact Infrared (IR) sensing. We then provide experimental and analytical evidence of the error induced in spot-welded thermocouple readings due to current flowing in the wire under measurement. Three methods are proposed to compensate for these errors: current reversal and averaging, pulse shut-off measurement, and a zero-offset spot-welding process.

BACKGROUND

Thermocouples are a common choice for temperature measurement because of their self-energization, low cost, robust nature and wide temperature range. With minor calibration corrections, thermocouples can have accuracies of 0.25 – 0.5°C. Michalski et al. [1] conducted research on measuring the temperature of solid bodies with a thermocouple and pointed out that temperature measurement of a solid surface has a series of possible errors
including: *temperature distortion error* of the solid surface when introducing a thermocouple, and *contact thermal resistance error* between the solid surface and the thermocouple.

Volkov [2] proposed that thermocouple readings can be affected by current in such a way as to add an “effect voltage” $\Delta U$ to the thermo-emf of the thermocouple if it has direct contact with the current carrying surface. He concluded that the influence of current on thermocouple readings was a function of the voltage gradient along the current carrying surface. Furthermore, both [3] and [4] propose a method using two thermocouple junctions of opposing polarity, affixed next to each other on a current-carrying conductor, to measure and compensate for the current-induced error.

Kuribayashi [5] used a thermocouple twist junction attached in close proximity to the SMA wire to avoid measurement errors due to current going through the SMA wire. He also used isolation transformers to electrically separate the sensor circuit from the SMA wire heating circuit.

There is little further evidence of consideration of this problem in the literature, perhaps due to the particular nature of the SMA wire. For temperature measurement of larger-diameter conductors, electrically-insulated thermocouples are recommended [2]. However, the insulation increases the volume of the bead relative to the thin wire under test, introducing further uncertainty in the measurement.

Leading up to our thermocouple-based results which are the focus of this paper, we investigated the use of thermistors and non-contact infrared sensing. A thermistor is a device whose electrical resistance varies in a predictable way with temperature. Thus, a thermistor located at a spot under test can be used in a resistance bridge to measure temperature. Typical thermistors are more stable and can be orders of magnitude more precise than thermocouples [6]. However, the smallest thermistors which are readily-available are still larger than a small thermocouple bead [e.g., 7] and have a significant influence on the local temperature of the SMA wire during measurement. IR measurement was attempted and eventually found to be too cumbersome due to the need for specialized lenses given the target size, the difficulty accounting for background effects, the limited accuracy (approximately $\pm 2^\circ C$, [8]), the need to accurately know the SMA wire emissivity and the effects of wire curvature on IR temperature measurement. In addition, while successful IR measurements could be used in the lab for model development and validation, they could never be implemented *in situ* in our eventual application.

**SMA WIRE CURRENT INFLUENCE ON THERMOCOUPLE READINGS**

In an early experiment comparing thermocouple attachment methods, two 500mm long samples of 500$\mu m$ diameter SMA wire (Flexinol, from Dynalloy) were prepared by affixing seven 36AWG type-T thermocouples along the wire length at regular 50mm separations between 100mm and 400mm along the wire. On one wire, the thermocouple beads were spot-welded, while on the other they were affixed with adhesive gel (QuickTite, from Loctite). Examples are shown in Fig. 1.
A current of 250mA was applied to each of the wires, first in one direction then in the opposite, and the steady-state readings from all thermocouples recorded. The results are shown in Fig. 2. For the glued thermocouples, measurement differences along the length of the wire can be easily ascribed to thermocouple accuracy and variability in the gluing process as evidenced by comparing Fig. 1(a) and Fig. 1(b). The difference in readings when current is reversed is minimal. The spot-welded thermocouples, on the other hand, show wider variability between thermocouples. In addition, the current direction has a large influence on readings, producing mirror-image curves which appear symmetric about approximately 26°C. The average reading from the glued thermocouples is also about one degree lower than the average from the spot-welded thermocouples, which is not unexpected given the greater thermal resistance of the adhesive.
wire temperature, this offset should have no influence on the thermocouple emf. However, we hypothesize that the wire current induces an ohmic drop proportional to the lateral separation of the leads. This additional voltage corrupts the thermoelectric voltage generated by the thermal field, leading to an erroneous reading which deviates from the average reading of 26°C. The fact that the error-induced offset reverses its sign when the current is reversed supports this hypothesis.

![Images of TC spot welding](a), TC 1 spot welded  
(b), TC 3 spot welded  
(c), TC 4 spot welded

**Fig. 3:** Spot welded TC

To further verify this idea, we can calculate the lateral offset suggested by the observed errors, by comparing the measured thermocouple voltages with the expected emf for a T-type thermocouple at 26°C. The equation is

\[ U_{26°C} - U_{\text{measured}} = I \cdot \rho \cdot \Delta x \]  

where \( U \) is the T-type thermocouple emf, \( I \) is the wire current (250mA), \( \rho \) is the linear resistance of the wire (6.3 \( \Omega \)/m [9]), and \( \Delta x \) is the thermocouple lead separation. T-type thermocouple emfs for TC 1 readings from **Fig. 2** were computed from thermocouple coefficients in the NIST ITS-90 database [10], and are given in **Table 1**.

**Table 1:** Temperature readings and corresponding voltages for TC1

<table>
<thead>
<tr>
<th>Temp (C)</th>
<th>U (mV)</th>
<th>comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>26.0</td>
<td>1.033</td>
<td>average reading</td>
</tr>
<tr>
<td>26.8</td>
<td>1.066</td>
<td>reverse current</td>
</tr>
<tr>
<td>25.2</td>
<td>1.000</td>
<td>forward current</td>
</tr>
</tbody>
</table>

The lateral offset between the leads in TC1 can then be estimated using (1), giving a separation of 21μm. Similarly, the offsets of TC3, TC4 and TC7 can be estimated at 17μm, 61μm and 9μm respectively. It can be seen from this experiment and analysis that even very small offsets can introduce significant measurement error, particularly at relatively high currents and temperatures close to room temperature. Since currents of 250mA or higher, and room temperature operation are very normal conditions for an SMA actuator, it is important to develop a method to account for these current-induced measurement errors.
COMPENSATING FOR CURRENT-INDUCED MEASUREMENT ERROR

Three methods are proposed and have been verified, to compensate for measurement errors induced in thermocouples spot-welded to current-carrying conductors.

Current reversal and averaging

Since the error component of the measured thermocouple voltage is due to an ohmic drop, its sign is dependent on the direction of current flow, as seen in Fig. 2. One approach to compensate which has been previously proposed is to use a “compensated thermocouple” comprised of three thermocouple leads [4]. Two leads of a similar metal are affixed to the surface under measurement, and the third lead, of a dissimilar metal, is affixed in the centre. This creates two thermocouple junctions which will have opposite current-induced errors, and the average voltage can be read to get the “true” temperature. This approach still relies on the precise relative positioning of the leads on the surface, but residual errors due to lateral spacing differences would be smaller than the original current-induced errors.

Often, the amplitude or pulse-width modulation (PWM) duty cycle of the current in an SMA wire is controlled by a computer or embedded microcontroller, in order to control temperature and hence actuator motion. The greater the amplitude or duty cycle, the more power is delivered to the wire and the faster the heating. In this case, another approach using a single traditional thermocouple can be used to reduce or eliminate current-induced measurement errors. Since current direction does not affect heating, the control electronics can be used to alternate current direction while achieving the same overall desired control. If temperature measurements are synchronized with the current reversals, the average thermocouple emf can be used to eliminate the effects of the ohmic drop on the temperature reading.

Pulse shut-off measurement

The pulse shut-off method takes advantage of the fact that the electrical time constant is much smaller than the thermal time constant of the system: the problematic ohmic drop disappears as soon as the current is removed, while the wire takes longer to cool. This is illustrated in Fig. 4, which shows the response of a 40AWG K-type thermocouple spot-welded to a 500μm diameter Flexinol wire. Thermocouple readings were recorded during two trials, in which 750mA was run through the wire, in opposite directions in each trial. At steady state with current flowing, the measured thermocouple temperatures are -4C with current in one direction, and 78C after the polarity is switched. When current is shut off, the thermocouple reading in both cases jumps almost immediately to 37C, the average of the two. The jump is followed by an identical slow decrease in measured temperature as the wire cools to ambient temperature.
By synchronizing the temperature measurement with the removal of the current in the wire, a single spot-welded thermocouple can be used to get an accurate reading, avoiding errors induced due to ohmic drop. This is particularly suited to SMA actuator applications where heating is controlled using a PWM current signal, as the measurement can be synchronized with the off-cycle of the control signal.

To further confirm the validity of the averaged and pulse shut-off measurements, we can use a first-order heating model accounting for Joule-heating and convection, to approximate the expected steady-state temperature of the wire in our experiments [11]:

\[ T_{ss} = T_\infty + \frac{\rho}{h \cdot \pi \cdot d} I^2 \]  

where \( T_{ss} \) is steady-state temperature, \( T_\infty \) is ambient temperature, \( I \) is the wire current (250mA), \( \rho \) is the linear resistance of the wire (6.3 \( \Omega /m \), [9]), \( d \) is the wire diameter (500mm), and \( h \) is the convection coefficient (75 W/m\(^2\)C, [11]). Equation (2) confirms that at a measured ambient temperature of 23C, the steady state temperatures for our wire at 250mA and 750mA are approximately 25C and 38C respectively. Within thermocouple measurement error, these are the two average temperatures observed in Fig. 2 and Fig. 4.

Zero-offset spot-welding process

Different from the two methods above, zero-offset spot weld method focuses on the joint point where thermocouple is attached on the SMA wire and eliminates the ohmic drop through improved spot welding techniques. In the conventional spot welding of thermocouples, a bead is made from two thermocouple wires and then the bead is welded to the SMA wire. This method cannot eliminate the voltage drop between two thermocouple
wires associated with the current flow in the SMA wire. The challenge is to properly spot weld thermocouple wires on the SMA surface, minimizing lateral offset to reduce the current’s influence, yet providing better thermal contact than a glue-affixed or insulated thermocouple. This problem can be resolved by first attaching one thermocouple wire to the SMA and subsequently attaching the second thermocouple wire to form the measurement bead. Being a purely mechanical solution, this approach does not depend on the presence of a computer or microcontroller being used for SMA actuator control.

After attaching a thermocouple onto the SMA wire by the zero-offset spot welding, either the average reverse current method or pulse shut off method can be used to examine if there is an ohmic drop on the thermocouple leads. For example, Fig. 5 illustrates that four 40AWG E-type thermocouples spot welded to the SMA wire using the zero-offset process, have no ohmic drop when they are subject to the pulse shut off test. In these figures, when temperatures go above 100°C, the power is shut off by computer and all temperatures decrease smoothly (curve in circle in TC 1), without showing a steep drop or jump as observed in Fig. 4. If the observed offset error is too large, the attachment can be removed and redone. If this is not feasible, either of the two previous compensation methods can be used.

Fig. 5: Zero-offset spot welded thermocouples are examined by pulse shut off method
CONCLUSION AND FUTURE WORK

Based on the above experimental and theoretical analysis, it can be concluded that a combination of improved spot welded thermocouple on the SMA wire, pulse shut off method and average reverse current method is the best practice for laboratory temperature measurement of SMA wires using thermocouples. We give guidance on the selection of the appropriate method depending on the application and type of control.

Future work on temperature measurement will focus on using IR imaging to investigate the error induced by thermal conductance down the thermocouple leads (the “fin effect”). The described measurement techniques will also be applied in our efforts to develop more detailed thermo-electric models of SMA and obtain heat-transfer coefficients.

REFERENCES

ACTIVE BUCKLING CONTROL OF BEAMS USING PIEZOELECTRIC ACTUATORS AND STRAIN GAUGE SENSORS

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ABSTRACT

In this paper, a finite element model incorporating active control techniques has been developed to stabilize the first two buckling modes of a simply supported beam. The goal is to increase the corresponding beam buckling loads by using piezoelectric actuators along with optimal feedback control.

The uniform beam is bonded with two pairs of segmented piezoelectric actuators at top and bottom sides. Resistive strain gauges are attached to the centres of the actuator as sensors. Their measurements are used to estimate the system states. The beam is subjected to a slowly increasing axial compressive load. Finite element formulation based on the classical Bernoulli-Euler beam theory in conjunction with linear piezoelectric constitutive equations for the actuators is presented. The associated reduced order modal equations and the state-space equations are derived for the design of a standard linear quadratic regulator (LQR).

The active control simulation results are consistent with both the theoretical results and the experimental data of other researchers. The designed full state feedback LQR controller is shown to be successful in stabilizing the first two buckling modes of the beam.

Keywords: Buckling Loads, Active Control, Piezoelectric Actuators, Strain Gauge Sensors.
INTRODUCTION

The problem we address in this paper is the active control of the first two buckling modes of a simply supported beam. Meressi and Paden [1] have shown that the buckling of such a beam can be postponed beyond the first critical load by means of feedback control using piezoelectric actuators and strain gauge sensors. Hence, the controlled beam can support a load up to the second critical load. Berlin [2] constructed a prototype composite column that was stabilized against buckling through the use of piezoelectric actuators and non-adaptive control strategies. He demonstrated that multiple buckling modes can be stabilized simultaneously. The load-bearing strength of his controlled column was increased by 5.6 times. Some other researchers have also discussed multi-mode control problems [3].

Here we first present a dynamic finite element buckling analysis of an axial compressed simply supported beam. The associated modal equations and the state-space equations of the reduced order system are then derived. Additionally, we design a compensator combining the feedback control law and the dynamic observer to stabilize the first and the second buckling modes of the beam. Lastly, some results and conclusions are given.

SYSTEM MODEL

The model is a simply supported beam subjected to axial compression, Fig. 1. The beam has width $b_b$ and thickness $t_b$, and is bonded with four piezoelectric actuators at the top and bottom sides. A resistive strain gauge is attached at the centre of the surface of each actuator. Each piezoelectric patch is polarized along the Z-axis. A voltage is applied across its thickness as an actuator. The width of each piezo-patch $b_p$ is assumed to be the same as that of the beam. The strain gauges are placed the third points of the beam span, i.e., at $x = L/3$ and $x = 2L/3$.

![Fig. 1: Beam bonded with actuators and gauge sensors subjected to axial compression.](image)

The following parameters are used in simulations to be presented.

**Beam properties:**

$L_b = 152.4\text{mm}$ $b_b = 25.4\text{mm}$ $t_b = 1\text{mm}$ $E_b = 5\text{GPa}$ $\rho_b = 1000\text{kg} / \text{m}^3$ $\nu = 0.3$

**Piezo-actuator properties:**

$L_p = b_p = 25.4\text{mm}$ $t_p = 0.110\text{mm}$ $E_p = 2\text{GPa}$ $\rho_b = 1780\text{kg} / \text{m}^3$ $d_{31} = 23 \times 10^{-12} \text{m/V}$

**Strain gauge constant:**

$k_g = 0.01\text{V}/\mu$
Finite element analysis (FEA) model

For a beam subjected to a lateral bending loading and with a slowly increasing axial compressive load \( P \), the Bernoulli-Euler beam buckling equation is:

\[
\rho A \ddot{w} + EI \frac{d^4 w}{dx^4} + P \frac{d^2 w}{dx^2} = p(x,t)
\]  

where, \( w(x,t) \) is the transverse displacement of the beam, \( \rho \) is the mass density per length, \( A \) is the cross-sectional area, \( EI \) is the beam rigidity, and \( p(x,t) \) is the externally applied lateral dynamic loading.

For FEA, in accordance with the beam theory, a cubic displacement field in the axial direction is chosen: \( w(x,t) = H(x) d(t) = \sum_{i=1}^{4} H_i(x) d_i(t) \), where \( H_i(x) \) are the standard Hermite polynomials of third degree, and \( \{d^e(t)\} \) denotes the time dependent element nodal response. The resulting FE equations for a beam element are:

\[
[M^e]\{\ddot{d}^e\} + [K^e]\{d^e\} - P[K_G^e]\{d^e\} = \{f^e\}
\]  

The other quantities in the above equation are: the consistent mass matrix, the element force vector, the bending stiffness matrix, and the geometric stiffness matrix. These are, respectively [4]:

\[
[M^e] = \frac{\rho Al}{420} \begin{bmatrix} 156 & 22l & 54 & -13l \\ 22l & 4l^2 & 13l & -3l^2 \\ 54 & 13l & 156 & -22l \\ -13l & -3l^2 & -22l & 4l^2 \end{bmatrix}
\]

\[
[K^e] = \frac{EI}{l^3} \begin{bmatrix} 12 & 6l & -12 & 6l \\ 6l & 4l^2 & -6l & 2l^2 \\ -12 & -6l & 12 & -6l \\ 6l & 2l^2 & -6l & 4l^2 \end{bmatrix}
\]

\[
[K_G^e] = \frac{1}{30l} \begin{bmatrix} 3l & 36 & -36 & 3l \\ -3l & 4l^2 & -3l & -l^2 \\ -36 & -3l & 36 & -3l \\ 3l & -l^2 & -3l & 4l^2 \end{bmatrix}
\]

After assembly of element matrices and vectors, the system matrix equation is obtained as

\[
[M]\{\ddot{d}\} + [K]\{d\} - P[K_G]\{d\} = \{F\}
\]

Piezoelectric-mechanical constitutive equations

For the actuator, linear coupled piezoelectric-mechanical constitutive relation is

\[
\{S\} = [c^E]\{T\} - [d]\{E\}
\]
where \{T\}, \{S\}, \{E\}, \{c\} and are respectively the stress, strain, electric field intensity, and the elastic compliance, and piezoelectric constants matrices. The superscript \(E\) means that the compliance matrix is evaluated at a constant electric field.

The coupled equations for the beam and the piezoelectric actuator can be expressed as:

\[
[M]\{\ddot{d}\} + [K]\{d\} - P[K_G]\{d\} = \{F\} + \{F_p\} \tag{5}
\]

\[
\{F_p\} = b_p E_p d_{31}(t_p / 2 + t_b / 2)(V_2 - V_1) \tag{6}
\]

Here, \(\{d\}\) is the generalized nodal displacement vector, \(\{F_p\}\) is the actuator force vector and \(d_{31}\) is the piezoelectric strain constant. \(V_1, V_2\) are the top and bottom voltages applied to the piezo-layers. \(E_p\) and \(t_p\) are the Young’s modulus and the thickness of these layers.

**Reduced-order model equations**

Using the mode superposition method, in which system modal matrix is used to transform the finite element nodal displacement vector to the modal coordinate vector, we obtain an approximate reduced-order model of the system in modal coordinates.

The generalized nodal displacement vector \(\{d(t)\}\) can be approximated by

\[
\{d(t)\} \approx \sum_{i=1}^{r} \phi_i q_i(t) = [\Phi] \{q(t)\} \tag{7}
\]

where \([\Phi]\) is the truncated modal matrix, assembled from the free vibration modes:

\[
[\Phi] = [\Phi_1, \cdots, \Phi_r] \quad (r < n) \tag{8}
\]

\(\{q(t)\}\) is the modal coordinates vector, which is a time dependent vector of order \(r\), the number of retained modes or the number of modes to be controlled.

After introducing damping, the linear decoupled reduced-order modal equations of the feedback control system are:

\[
[\tilde{M}]\{\ddot{q}\} + [\tilde{C}]\{\dot{q}\} + [\tilde{K}]\{q\} = \{\tilde{F}\} + [\tilde{K}_A]\{u_a\} \tag{9}
\]

Here \([\tilde{M}] = [\Phi]^T[M][\Phi], [\tilde{C}] = [\Phi]^T[C][\Phi], [\tilde{K}] = [\Phi]^T([K] - P[K_G])[\Phi]^T\) are the diagonal modal mass, diagonal model damping, diagonal model stiffness matrices. \(\{\tilde{F}\} = [\Phi]^T\{F\}\) is the external disturbance forcing input vector, and \(\{u_a\}\) is the applied actuator voltage which is the control input vector, \([\tilde{K}_A]\) is the modal actuator stiffness matrix or control input influence matrix.

**Sensor modelling**
Modal states are estimated from strain gauge measurements at the chosen discrete locations. We design the modal control based on the first, second, fourth and fifth buckling modes of the beam. Four sensors are placed at both sides at $x = L/3$ and $x = 2L/3$ of the beam, which are the zero points of the third buckling mode. This mode and its multiples are thereby rendered unobservable.

We take these strain gauges measurements as the system output. From Meressi [1], the output of a gauge is given by

$$v_s = 2\sqrt{\frac{2}{L}} k_g \frac{t_b}{t_p} \frac{(2 + t_p)}{L^2} \sum_{n=1}^{\infty} n^2 q_n \sin \left( \frac{n\pi x}{L} \right)$$

in which $k_g$ is the strain gauge constant and $q_n$ is the nth modal coordinate. The system output vector is

$$V_0 = \begin{bmatrix} v_{01} \\ v_{02} \end{bmatrix} = \begin{bmatrix} v_s(x = \frac{L}{3}) \\ v_s(x = \frac{2L}{3}) \end{bmatrix}$$

COMPENSATOR DESIGN

Introduction of the state-space vector $\{x\} = \begin{bmatrix} q \\ \dot{q} \end{bmatrix}$ results in the system equations of the form

$$\{\dot{x}\} = [A]\{x\} + [B]\{u\}$$
$$\{y\} = [C]\{x\}$$

where, the state vector $\{x\} = [q_1 \ \dot{q}_1 \ q_2 \ \dot{q}_2 \ q_4 \ \dot{q}_4 \ q_5 \ \dot{q}_5]^T$, the actuator input $\{u\} = [u_1 \ u_2]^T$, the sensor output $\{y\} = [v_{01} \ v_{02}]^T$.

Next we use linear quadratic regulator (LQR) control technique to design a full state feedback controller and a state observer.

Controller design

The full state feedback control law is:

$$\{u\} = -[G]\{x\}$$

By LQR technique, we can get the feedback gain matrix $[G]$ to make the closed-loop system stable. Thus, the control input of actuator can be obtained by (13). The LQR is designed to minimize a cost function:
\[ J = \int_{0}^{\infty} \left( \alpha \{q\}^T \{Q\} \{q\} + \{u\}^T \{R\} \{u\} \right) dt \]  (14)

where, \( \alpha \) is a scalar and \( \{Q\} \) and \( \{R\} \) are positive semi-definite and positive definite weighting matrices respectively. The gain \( \{G\} \) is obtained from the corresponding Riccati equation.

**Dynamic observer design**

Because the number of sensors is less than that of state variables, the observer is needed to provide the feedback control law with estimated state variables. We can design a dynamic observer to estimate the state variables out of the direct sensor output, which can be prescribed by the following set of equations:

\[
\begin{align*}
\{\hat{x}\} &= [A_I]\{x\} + [B_I]\{u\} + [L]^T \left( \{y\} - \{\hat{y}\} \right) \\
\{\hat{y}\} &= [C_I]\{\hat{x}\}
\end{align*}
\]  (15)

where \([L]\) is the observer gain to be determined, \(\{\hat{x}\}\) is the observer estimated state variable vector, \(\{y\}\) is the sensor strain gauge output vector, and \(\{\hat{y}\}\) is the observer estimated sensor output vector.

We combine the controller and observer into a complete system which uses the estimated state variables from the observer in the feedback control law:

\[ \{u\} = -[G]\{\hat{x}\} \]  (16)

Substituting (16) into (12) and (15), the combined system equations are expressible as:

\[
\begin{align*}
\{\hat{x}\} &= [A_I]\{x\} - [B_I][G]\{\hat{x}\} \\
\{\hat{x}\} &= ([A_I] - [B_I][G] - [L]^T[C_I])\{\hat{x}\} + [L]\{y\}
\end{align*}
\]  (17)

The feedback gain \(\{G\}\) and the observer gain \([L]\) can be designed separately.

**RESULTS AND CONCLUSIONS**

The resulting closed-loop responses to nonzero initial conditions (no lateral loading \(p(x,t)\) applied) and the control input voltages to the actuators for the controlled model with load \(P = 3.8P_{cr,1}\) and \(P = 8.8P_{cr,1}\) are shown in Figs. 2 to 5, where \(P_{cr,1}\) is the first mode buckling load. We see that the designed compensator has stabilized the first two modes for any \(P \leq P_{\text{max}} (= 9P_{cr,1})\) and is robust.

Figure 2 shows that the closed-loop responses are consistent with the results of Meressi and Paden [1] for the load \(P = 3.8P_{cr,1}\). Figure 3 compares their control input voltage with that of the present segmented actuator pairs. The latter shows a better control, leading to the conclusion that optimally located actuators along the beam are more effective.
Spill-over has not posed a serious problem which is consistent with the theoretical analysis and known simulation results. Meressi and Paden [1] have shown that there is no significant effect of the uncontrolled modes on the dynamics of the controlled modes.

The sensor outputs of Figs. 4 and 5 show that the first two buckling modes of the beam are stabilized. The beam can therefore support a load up to the third critical load.

![Fig. 2: Closed-loop responses for $P = 3.8P_{cr,1}$](image1)

![Fig. 3: Control input voltages for $P = 3.8P_{cr,1}$](image2)
Fig. 4: Closed-loop response for $P = 8.8P_{cr,1}$

Fig. 5: Control input voltages for $P = 8.8P_{cr,1}$
REFERENCES

GENERATION AND DETECTION OF GUIDED ACOUSTIC WAVES FOR NDT USING HIGH TEMPERATURE WEDGES

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ABSTRACT

Ultrasonic wedges which can propagate one longitudinal L and two orthogonally polarized shear S waves are presented. In order to use this wedge to generate and received guided plate acoustic waves along a metal plate at high temperatures brass which has a slow L and S wave velocities was chosen. Mode conversion method was used to convert L to S waves in the wedge. The mode conversion efficiency is 90.8%. The wedge has been tested up to 150°C. The L and S waves in the wedge have been converted to symmetrical, anti-symmetrical and shear horizontal plate acoustic waves in a stainless steel plate. Experimental measurements demonstrated that the guided acoustic waves, in particular, shear horizontal plate waves using high temperature wedges may be an excellent approach for non-destructive testing and structural health monitoring of metal structures.

Keywords: Ultrasonic wedge, Structural health monitoring, Non-destructive testing, integrated ultrasonic transducers, Guided plate acoustic waves, Piezoelectric thick films, Sol-gel process.
INTRODUCTION

Guided acoustic waves are of attraction for non-destructive testing (NDT) [1, 2] and structural health monitoring (SHM) applications [3, 4] because they may inspect parts or structures, in particular, made of metals of a large area within a short time period using a few ultrasonic transducers (UTs). For aerospace industry such NDT and SHM may require that the UTs operate from -80°C to 100°C. In other areas the operation temperatures may be required to be higher [5-7]. Guided waves may be plate (PAWs) or surface acoustic waves (SAWs).

Wedges made of Plexiglas are commonly used to convert longitudinal (L) or shear (S) waves in the Plexiglas into guided PAWs and SAWs along metallic plates or pipes for NDT [1, 8] and SHM purposes. The main advantages of Plexiglas are due to its low ultrasonic velocity, $V_{L,Wedge}$, or $V_{S,Wedge}$ and ease for machining, where $V_L$ and $V_S$ are the L and S wave velocity, respectively. The low velocity of a wedge is a necessary to meet the conversion angle according to the Snell’s law (or acoustic phase matching condition). For example, in the case of a plate, $\sin \theta = V_{Wedge}/V_{PAW}$ or $\sin \theta = V_{Wedge}/V_{SAW}$, where $\theta$ is the angle of the wedge and $V_{PAW}$ and $V_{SAW}$ are the phase velocity of specific mode of PAWs and SAWs, respectively of a plate. It also means that $V_{Wedge}$ is slower than $V_{PAW}$ or $V_{SAW}$. However, the operation temperature of the Plexiglas is limited to less than 100°C. In order to operate at high temperature (HT) wedges made of metal may be preferred. In general, brass provides low $V_L$ and $V_S$ and it may function up to more than 600°C, thus it is chosen as the HT wedge material in this investigation.

For NDT and SHM broad bandwidth of the ultrasonic signals is also desired. Normally the broad bandwidth of an ultrasonic transducer (UT) is to have the proper backing. However, it is difficult to construct a robust backing material for HT UT [5-7]. In this study sol-gel fabricated HT integrated L wave UTs [9-11] will be used together with L to S wave mode conversion technique [12, 13] to produce broad bandwidth shear waves in the brass wedge in order to launch PAWs and SAWs in the metal plate for NDT and SHM purposes.

INTEGRATED ULTRASONIC TRANSDUCER AND MODE CONVERSION

Recently, integrated ultrasonic transducers (IUTs) have been made using the sol-gel based fabrication process [10, 11] consists of six main steps: (1) preparing high dielectric constant lead-zirconate-titanate (PZT) solution, (2) ball milling of piezoelectric PZT powders to submicron size, (3) sensor spraying using slurries from steps (1) and (2) to produce a layer of PZT composite (PZT-c) film, (4) heat treatment to produce a solid composite PZT-c film, (5) corona poling to obtain piezoelectricity, and (6) electrode painting or spraying for electrical connections. Steps (3) and (4) are used multiple times to produce proper film (IUT) thickness for optimal ultrasonic operating frequencies. It is the objective to use such IUTs to achieve L UTs. The typical PZT-c film thickness in this study is about 84 µm. Such IUTs has been operated with a center frequency ranging from 4 to 30 MHz. Their ultrasonic signal strength and bandwidth is comparable to those of the commercially available broadband UTs with backing, however, IUTs can be used at high temperatures [11].

The mode conversion from L to S waves due to reflection at a solid-air interface, as shown in Figure 1, was reported [12, 13]. It means that the L wave IUT together with L-S mode conversion caused by the reflection at a solid-air interface can generate S waves effectively and be used as an S wave probe. Our approach considers a simple way to
fabricate the L wave IUT and let the L IUT be in a plane parallel to axial direction of the probe as shown in Figure 1. The only criterion is that in Figure 1 $\cot \theta$ is equal to $V_S / V_L$ in the probe so that the mode converted shear waves will propagate in the direction parallel to the axial direction of the probe. In Figure 1 $L_i$, $L_r$ and $S_r$ are the incident L, reflected L and reflected S wave, respectively. In this study, a brass with the L wave velocity $V_L = 4295$ m/s and S wave velocity $V_S = 2223$ m/s was used as the probe material. Therefore one can obtain $\theta = 62.6^\circ$. Using the energy reflection coefficient formula reported in Reference [13] the calculated energy conversion rate versus $\theta$ is given at Figure 2. In Figure 2 at $\theta = 62.6^\circ$ the energy conversion rate is 90.8% that is 0.4% smaller than the maximum conversion rate at $\theta = 65.7^\circ$. In Figure 1 an L wave IUT is also made at the top flat surface of the wedge and it can generate and receive the L wave along the axial direction of the probe as well. The arrangement shown in Figure 1 enables both L and S waves propagate together along the axial direction of the probe which will be further explained below.

![Fig. 1: Reflection and mode conversion with an incidence of L wave at a solid-air interface.](image1)

![Fig. 2: Reflection and mode conversion with an incidence of L wave at a brass-air interface.](image2)

Using the L to S wave mode conversion principle shown in Figure 1 if one would like to generate two shear waves with orthogonal polarizations (birefringence) simultaneously, then $\theta = 62.6^\circ$ will be made at two orthogonal edges as shown in Figure 3. Figure 3 shows an integrated probe having three L wave IUTs which will generate and receive one L and two S waves with orthogonal polarizations. Let one S be $S_{\perp}$ polarized in Y direction and the other
S\textsubscript{∥} polarized in X direction. L wave is polarized in Z direction. S\textsubscript{⊥} and S\textsubscript{∥} are designated arbitrarily. Figure 4 shows the measured ultrasonic signal \( L^n \) in time domain and pulse-echo mode from the end of the probe at 150°C. \( L^n \) is the nth trip echo through the axial direction of the probe. Figures 5a and 5b show the measured ultrasonic signal \( S_{\perp}^n \) and \( S_{\parallel}^n \), respectively reflected from the end of the probe in time domain and pulse-echo mode at 150°C where the time delay of \( S^n \) is that of the nth trip S echo through the probe length plus that of the L wave travelling through the length from L IUT to the brass/air interface. The measurement results shown in Figures 4, 5a and 5b can be made simultaneously using a three-channel ultrasonic system or the electrical connections of the three L IUTs can be connected to use ultrasonic system of reduced channel such as two or one. The center frequency and 6 dB bandwidth of the \( L^2 \) signal shown in Figure 4 are 6 MHz and 3.7 MHz, respectively. Its SNR is 43.7 dB. The SNR is defined as the ratio of the signal \( L^2 \) over that of the surrounding noises. The center frequency and 6 dB bandwidth of the \( S_{\perp}^2 \) and \( S_{\parallel}^2 \) signals shown in Figures 5a and 5b are 5.6 MHz and 3.1 MHz, and 4.9 MHz and 2.8 MHz, respectively. Their SNRs are 15.5 dB and 23.9 dB, respectively. The echo \( L_{\text{S}} \) indicated in Figures 5a and 5b are undesired and it comes from the reflection of L waves from the edge opposite to the L IUT surface within the probe head area shown in Figure 3 because its arrival time is shorter than \( L^2 \) shown in Figure 4. It is noted that such probe shown in Figure 3 may be used as an ultrasonic interferometer which is sensitive to, for example, the anisotropy of the material to be measured, which induces a difference in particle displacement direction or velocity or both between two shear wave propagations along the material.

Fig. 3: An integrated S wave probe having two polarizations \( (S_{\perp} \text{ and } S_{\parallel}) \) generated and received by two IUTs.

Fig. 4: Ultrasonic signal in time domain of the L wave generated by the L IUTs shown in Fig. 3 reflected from the end of the probe at 150°C.
Fig. 5: Ultrasonic signal in time domain of the (a) $S_{\perp}$ and (b) $S_{//}$ wave generated by the L IUTs shown in Fig. 3 reflected from the end of the probe at 150°C.

CONVERSION FROM L AND S WAVES IN BRASS WEDGE TO PAWS IN A SS PLATE

Figures 6a, 6b and 6c shows the intended NDT and SHM applications in which the probe shown in Figure 3 is cut into a slanted angle $\phi$ to become a wedge so that the L wave in Figure 6a and $S_{\perp}$ wave in Figure 6b and $S_{//}$ in Figure 6c propagating along the axial direction of the wedge will be converted to the desired PAWs along the metal plate which is made of a SS here.

Fig. 6: Configurations of wedges for (a) L waves to $S_n$ and/or $a_n$ (b) $S_{\perp}$ waves to $S_n$ and/or $a_n$ and (c) $S_{//}$ waves to SH$_n$ PAWs.
If the thickness of the plate is more than ten wavelengths, then SAW propagation will be feasible if a proper angle $\phi$ is chosen. Let $S_n$ be the symmetrical, $a_n$ be the anti-symmetrical and $SH_n$ be the shear horizontal (SH) PAWs, where $n$ denotes high order modes. The tips of the wedges in Figure 6a, 6b and 6c are not given but will be shown below. The operation can be in transmission mode which means that the one wedge shown in Figures 6a, 6b and 6c is used as the transmitting transducer and the other the receiver. If the wedge is operated in pulse-echo mode, each of them can be used as both the transmitting and receiving transducer. It is also one major interest of this study that can the same wedge angle $\phi$ be used for $\phi_L$, $\phi_{S\perp}$, and $\phi_{S//}$ to generate different PAWs efficiently and even simultaneously if the L IUTs can be connected electrically.

Theoretical Calculations:

A stainless steel (SS) plate with a thickness of 1.9 mm, a width of 50.8 mm is selected for this study. For this SS the dispersion curves of both phase (dashed lines) and group velocities (solid lines) of $S_n$, $a_n$ and $SH_n$ waves are calculated and given in Figures 7, 8 and 9, respectively, where $n$ indicates the high order modes. The frequency of the PAW and the plate thickness are expressed as $f$ and $h$, respectively.

Fig. 7: Theoretical calculated velocities versus $f^*h$, curves for the $S_n$ PAWs in the 1.9 mm thick SS plate. Dashed and solid lines represent phase and group velocities, respectively.
Fig. 8: Theoretical calculated velocities versus $f h$, curves for the $a_n$ PAWs in the 1.9 mm thick SS plate. Dashed and solid lines represent phase and group velocities, respectively.

Fig. 9: Theoretical calculated velocities versus $f h$, curves for the $SH_n$ PAWs in the 1.9 mm thick SS plate. Dashed and solid lines represent phase and group velocities, respectively.

Experimental Results:

The experimental setup for the $S_{//}$ wave in the wedge to excite and receive SH PAWs in the plate is shown in Figure 10. The angle 45.5° of $\phi_{S_{//}}$ is chosen because of Snell’s law $\phi_{S_{//}} = \sin^{-1} \frac{V_{S,\text{Wedge}}}{V_{SH,\text{PAW}}} = \sin^{-1} (\frac{2223}{3116})$. It is known that $SH_0$ PAW has no dispersion and
is preferred for many NDT and SHM applications. Also the higher order modes of SH\(_n\) (\(n \neq 0\)) have slower group velocities than that of SH\(_0\) and may not be excited and received efficiently. Figure 11 shows the result of the measurement in transmission mode at room temperature. The separation distance between the centers of the two wedges on the SS plate is also 200 mm. The echo P\(_{SH0}\) indicated in Figure 10 does not go through any signal processing. The preliminary analysis indicates that the echo comes mainly from SH\(_0\) mode shown in Figure 9 because little dispersion is exhibited. Its center frequency and 6 dB bandwidth of the P\(_{SH0}\) signal are 5.3 MHz and 3 MHz, respectively.

![Image of a 1.9mm thick and 50.8mm wide SS plate with a 38mm LUT and a 25mm LUT](image)

**Fig. 10:** The experimental setup for the S\(//\) wave in the wedge to excite and receive SH PAWs in the SS plate.

![Graph showing ultrasonic measurement results](image)

**Fig. 11:** Ultrasonic measurement results in transmission mode shown in Fig. 10 at room temperature. The distance between the centers of the two wedges is 200 mm.

Figure 12 shows the experimental setup for the L wave in the wedge to excite and receive PAWs in the SS plate. The angle of \(\phi_L\) equal to 45.5° which is the same as \(\phi_{S/}\) is chosen because it is the intention to see which S\(_n\) or a\(_n\) PAWs can be generated and received. From the Snell’s law if \(\phi_L\) is equal to 45.5°, the \(V_{PAW}\) should be a value of near 6022 m/s, then according to the data shown in Figures 7 and 8 S\(_5\) and/or a\(_5\) PAWs may be excited and received. Figure 13 shows the measurement result in transmission mode at room temperature. The separation distance between the centers of the two wedges is also 200 mm. The echo P\(_{a5}\) indicated in Figure 13 has gone through a band pass filter between 7.5 MHz and 8.5 MHz. The preliminary analysis indicates that this echo with a group velocity of 4160 m/s comes mainly from a\(_5\) mode shown in Figure 8 and S\(_5\) was not efficiently excited and received using this configuration.
Fig. 12: The experimental setup for the L wave in the wedge to excite and receive PAWs in the SS plate.

![Waveform Image](image1)

Fig. 13: Ultrasonic measurement results in transmission mode shown in Fig. 12 at room temperature. The distance between the centers of the two wedges is 200 mm.

![Waveform Image](image2)

Fig. 14: The experimental setup for the $S_{\perp}$ wave in the wedge to excite and receive PAWs in the SS plate.

![Waveform Image](image3)

Fig. 15: Ultrasonic measurement results in transmission mode shown in Fig. 14 at room temperature. The separation distance between the center of the two wedges is 200 mm.
For the $S_\perp$ wave in the wedge to excite and receive $S_n$ and $a_n$ PAWs in the plate the angle of $\phi_{S_\perp}$ equal to 45.5° is again chosen with an intention to see that if the same wedge used in Figures 10 and 12 is used, which PAW can be excited and received. The measurement setup is shown in Figure 14. Figure 15 provides the measurement result in transmission mode at room temperature. The separation distance between the centers of the two wedges is 200 mm as well. The echo $P_{a0}$ indicated in Figure 15 has gone through a band pass filter between 2 MHz and 3 MHz. The preliminary analysis indicates that the echo comes mainly from $a_0$ mode shown in Figure 8 because of the slow velocity of 2941 m/s.

It is interesting to note that the wedges used in the measurement setups shown in Figures 10, 12 and 14 have the same $\phi$. The reason is that in the future work it is the intention to see for a specific plate sample whether one can use wedges of same $\phi$ to excite and receive three different PAWs efficiently using the $L$, $S_\perp$ and $S_//$ waves in the wedge. If this can be achieved, this wedge will be an excellent candidate for NDT and SHM of this plate sample.

**CONCLUSIONS AND DISCUSSIONS**

Wedges in which $L$, $S_\perp$ and $S_//$ waves can propagate simultaneously are presented. In order to use this wedge to generate and received guided PAWs along a SS plate with a thickness of 1.9 mm and a width of 50.8 mm at high temperatures brass which has a slow $L$ and $S$ wave velocities was chosen. Mode conversion method was used to convert $L$ to $S_\perp$ and $L$ to $S_//$ waves in brass wedge. The energy conversion efficiency is 90.8%. The wedge has been tested up to 150°C. By way of proper electrical connections $L$, $S_\perp$ and $S_//$ waves can be generate and received simultaneously or individually. In the experiments it is demonstrated that with a wedge angle of 45.5° $L$, $S_\perp$ and $S_//$ waves in the wedge has generated and received $a_5$, $a_0$ and SH$_0$ PAW, respectively in the SS plate. The same angle $\phi$ of 45.5° is used so that one may use wedges of same $\phi$ to excite and receive three different PAWs efficiently in the wedge via proper electrical connections of all three LIUTs shown in Figure 3. Experimental measurements demonstrated that the guided PAWs using high temperature wedges may be an excellent approach for NDT and SHM of metal structures.

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A TWO-STEP IDENTIFICATION PROCEDURE FOR THE IDENTIFICATION OF DISTRIBUTED MATERIAL CONSTITUTIVE PARAMETERS FROM SURFACE DISPLACEMENT MEASUREMENTS

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ABSTRACT

The present work describes an inverse identification technique based on the finite element model updating for computing the spatial distribution of Young’s modulus using full-field displacement data obtained from digital image correlation. The problem is formulated as a Tikhonov-regularized of a minimization problem of a data discrepancy functional using the equilibrium equations as constraints. The gradient of the objective functional is evaluated using the adjoint method; a spatial filtering is introduced to the gradient image during the iterative process to improve the numerical stability in the presence of noise in displacement data. Numerical example is presented to demonstrate the performance of the presented methodology.

Keywords: Finite element model update, Adjoint method, Inverse problem, Elasticity imaging, Field measurement.

INTRODUCTION

Surface field measurement techniques, such as the digital image correlation, are becoming very popular in experimental mechanics. These techniques are potentially useful for the identification of heterogeneous spatial distribution of material properties. This has become a new and promising branch of research and development attracting widespread academic and industrial interests.
A recent issue of the journal of “Experimental Mechanics” had been dedicated to this specific area of research [1]. In a review paper published in this issue, Avril et al gave an extensive review of recently developed techniques that identify material parameters from full-field measurements [2]. The term full-field measurement, currently, mostly is equivalent to measurement of displacements and/or strain from digital image correlation [1]. It is also interesting to note that this area of research has long been an area undergoing extensive research in bio-medicine and biomechanics, usually under the name elasticity imaging or elastography [3-5].

The identification of distribution of material parameters is much more difficult than the problem of parameter identification of homogeneous material parameters. From parameterization of a continuous distribution would give rise to a huge number of unknown parameters. As a general approach, the problem is usually formulated as a nonlinear optimization problem minimizing the data discrepancy functional between the measured and predicted displacement fields, and control equations are included as constraints. This approach results in partial-differential-equations (PDE) - constrained optimization problem [6]. In the case of elasticity, Navier’s equations are used as PDE constraint. From mathematical point of view, this formulation is an elliptic distributed control problem, where the unknown variables can be viewed as controls and the measured variables (displacements or strain) are viewed as states [7].

In the following, we present an optimization-based finite element model updating technique for the reconstruction of the Young’s modulus from full-field displacement measurements. Many problems in applied mechanics, such as inclusions problem and damage accumulation, are directly related to the identification of distributed Young’s modulus. The solution process of the optimisation makes use of the adjoint method to evaluate the sensitivity parameters for constructing efficient optimization algorithms; a spatial filtering is used to modify the updating gradient vector, and stabilize the updating process with the presence of noise in data.

The adjoint method had been used in conjunction of an elasticity imaging technique developed by Oberai et al [3]. The presented method is formulated in the context of finite element with no reference to the continuous settings. The objective function is regularized using the Tikhonov method. The Tikhonov regularized finite element model update technique is also used by Doyley et al [5], where the sensitivities are evaluated using a direct differentiation approach. Numerical examples show that the presence of noises in the data, the reconstruction deteriorates severely; the development of inverse technique robust to noise in data is still an open question.

**THE RECONSTRUCTION METHODS**

**Problem set-up**

A general approach in solving inverse problem of structural parameter identification is to minimize an objective function defined as a data discrepancy to measure the fit-to-data between computed and measured mechanical response (i.e. either strain or displacements).
While the direct problem is solved using finite element method, this is also termed as finite element model updating.

The commonly used objective function is defined using the sum of squared differences between the measured data and the corresponding simulated values of the displacement field on the surface of the structure. A Tikhonov regularization term is added to overcome the ill-posedness and stabilize the solution. In the case of static equilibrium, the constrained optimization problem is as follows:

$$
\min_\theta J(u, \theta) = \frac{1}{2} \sum_{i=1}^{N} |u_i(\theta) - u_i^m|^2 + \frac{\alpha}{2} \|\theta\|_2^2
$$

where $N$ is the total number of measurements, $\theta$ is the vector of unknown parameters, $u^m$ is the measured data and $u(\theta)$ the corresponding simulated value using a trial distribution of parameters $\theta$. $\|\theta\|_2$ is the Euclidian norm of $\theta$. The constraint in equation (1) is the expression of equilibrium formulated within the finite element. The parameter $\alpha$ is the Tikhonov regularization parameter; the detailed analysis of the Tikhonov regularization and the selection of the regularization parameter can be found in [9].

The adjoint sensitivity analysis

The numerical solution of problem (1) requires the evaluation of the sensitivity of the observable variables (displacement) with regards to the control parameters $\theta$. If the number of control parameters is relatively small, for example less than ten, direct differentiation can be used to evaluate the sensitivity [8]. If the cost of direct problem simulation is small, powerful direct optimization techniques such as the genetic algorithms (GA) can be used without sensitivity analysis; the GA is especially powerful due to its convergence property to find the global optimal solution.

However, the reconstruction of distribution of material parameters is a functional identification problem; the discretization of the field often results in a large number of parameters to be evaluated. For problems with large number of unknowns, the adjoint-method based sensitivity evaluation is more efficient than any other sensitivity-evaluation techniques [8]. The constraint equations are the discretized governing equation using finite element; more generally they are written in the following form:

$$
R(\theta, u(\theta)) = K(\theta)u - f = 0
$$

where $u$ is the state variables (displacements in our problems), which is an implicit function of the unknown material variables $\theta$. We use Lagrangian multiplier to change the constrained optimization to an unconstrained problem. The augmented output function that enforces the governing equations via Lagrange multipliers is:
\[ L(\theta, u) = J(\theta, u) - \lambda^T R(\theta, u) \] (3)

Differentiating the Lagrangian with respect to \( \theta_i \), we have the gradient of the Lagrangian

\[ \frac{dL(\theta, u)}{d\theta_i} = \frac{\partial J}{\partial \theta_i} + \left( \frac{\partial J}{\partial u} - \lambda^T \frac{\partial R}{\partial u} \right) \frac{du}{d\theta_i} + \lambda^T \frac{\partial R}{\partial \theta_i} \] (4)

In this gradient, the term \( du/d\theta_i \) is very difficult term to evaluate (sensitivity). However, if we impose that the term \( \left( \frac{\partial J}{\partial u} - \lambda^T \frac{\partial R}{\partial u} \right) \) be equals to zero, one can calculate the Lagrange multiplier, \( \lambda \), then the sensitivity term, \( du/d\theta_i \), is no longer needed.

\[ \left( \frac{\partial J}{\partial u} - \lambda^T \frac{\partial R}{\partial u} \right) = 0 \] (5)

Equation-(5) is the adjoint equation and it is expressed as:

\[ (U - U^m) = \lambda^T K(\theta) \] (6)

where \( U \) is the displacement vector computed from the finite element equation-(2), and \( U^m \) is the measured displacement vector. Therefore we have:

\[ \frac{dL(\theta, u)}{d\theta_i} = \frac{\partial J}{\partial \theta_i} + \lambda^T \frac{\partial R}{\partial \theta_i} \] (7)

The two derivatives in (7), \( \frac{\partial J}{\partial \theta_i} \), and \( \frac{\partial R}{\partial \theta_i} \), can be easily computed. In computing \( \frac{\partial R}{\partial \theta_i} \),

\[ \frac{\partial R}{\partial \theta_i} = \frac{\partial K(\theta)}{\partial \theta_i} u - \frac{\partial f}{\partial \theta_i} \] (8)

The global stiffness in (8) matrix is an assembly of the elements’ stiffness matrix; and for linear elastic materials, the element stiffness matrix is proportional to the material modulus and geometric coefficient in the element, so that its derivative to \( \theta \) can be easily defined for a given stiffness matrix expression.

The iterative optimization procedure

The iterative optimization algorithm for finding a gradient consists of two consecutive steps: 1) solve (6) for \( \lambda^T \), then substitute \( \lambda^T \) into (7) to calculate the gradient of Lagrangian. With the gradient calculated from adjoint method, classical gradient-based optimization techniques can be used to find the unknown stiffness parameters. In the present paper, the gradient based pseudo-Newton optimization algorithm implemented in MATLAB.
optimization toolbox is used in the simulation [10]. The flowchart of the iterative process is given in Fig. 1.

Fig. 1: Flowchart of the iterative procedure.

The steps given above are general and can be applied to other finite element schemes providing the governing relation $R(0, u(0)) = K(0)u - f = 0$ holds.

Filtering the gradient

Doyley et al [5] proposed the idea of filtering the updated Young’s modulus at each iteration during an iterative reconstruction process. The filtering procedure proposed by the authors is of a type of edge-enhancing filtering; this type of filtering has been proven to
generate better images with abrupt changes of Young’s modulus. The idea is adopted here; but the filtering is performed over the calculated gradient image within each iteration, rather than the overall updated Young’s modulus. Simulations show that this type of filtering stabilizes the solution in the presence of noise.

The filtering adopted for the gradient is a sliding average filtering, which slide a 3-by-3 moving average across the 2-dimensional image of gradients, producing a low-pass filtered version of the original gradient image. This two-dimensional filtering can be implemented using two-dimensional convolution. At the k-th iteration, the gradient to update the Young’s modulus at element A is given as the filtered value

\[ D^k(A) = \frac{1}{6} \sum_{i=1}^{6} D^k(A_i) \]  

where the superscript \( k \) indicates the iteration number; \( A_i \) represent all the elements in the 6-by-6 block centered at the element A (including the element A itself), as shown in Fig. 2.

Within the image processing, this is also an edge-enhancing filtering. Edge-enhancing filtering are useful at reproducing blocky images, and stabilize the image-deblurring process. Numerical experiments show that this filtering is good for reconstructing piecewise-distribution of Young’s modulus in the presence of noise. Its relationship to image-deblurring and mathematical analysis is still an open question and need to be explored.

**Fig. 2**: The 3-by-3 block used in the sliding moving average of the gradient for element-A.

**NUMERICAL EXPERIMENT**

In order to verify the proposed technique, a simple problem is illustrated. The simple plate shown in Fig. 3 is considered. It is a composite specimen made of two different materials. The objective is to reconstruct the distribution of the Young’s modulus using the proposed algorithm. Using the finite element method, the specimen is discretized into \( 10 \times 20 = 200 \) plane stress Q1 elements of equal size. Hence, there are in total 200 unknown parameters representing the distribution of Young’s modulus of the specimen. Prescribed tensile displacement is applied to the right side while the left side is constrained.
The measured displacement field on the surface is simulated using computational analysis results for the given reference parameters. The Tikhonov regularization parameter is set equal to $10^{-4}$. This value is not meant to be the optimal one, i.e. the identification could be improved if a better value of the parameter is found.

Fig. 3: The plane stress bi-material specimen.

A simple additive Gaussian noise has been used to study the stability of the solution when noise corrupts the data. Numerical simulations show that the identification without added noise is satisfactory; however, with added noise, the performance deteriorates severely. At 2% noise level, the identified image of Young’s modulus distribution is shown in Fig. 4. The border between the two different materials is clearly visible, but its exact location is blurred within two to three elements, which corresponds to 10–15% of the total length of the longer side of the specimen. At 5% noise level, the border is hardly recognizable. One should bear in mind that noises are introduced not only from measuring process, but also from the modelling process; both sources are physically inevitable.

Fig. 4: The identified image of Young’s modulus distribution of the bi-material specimen.
A numerical index, the weighted average error (WAE), is introduced here to measure the performance of the algorithm:

\[
WAE = \sqrt{\frac{1}{N} \sum_{i=1}^{N} \left( \frac{E_i^{id} - E_i^{tr}}{E_i^{tr}} \right)^2}
\]  
(10)

where \(E_i^{id}\) and \(E_i^{tr}\) represent the identified and the true values of Young’s modulus at element \(i\), respectively. \(N\) represents the total number of elements in the finite element mesh. Another measure of performance is the maximum deviation of identified values from corresponding true values:

\[
\max_i \left( \frac{E_i^{id} - E_i^{tr}}{E_i^{tr}} \right) \times 100\%
\]  
(11)

Although the solution is not smooth, as shown Fig. 4, one can recognize that there are generally two different regions of the Young’s modulus. To improve the solution process, one can augment the previously presented algorithm by a parametric optimisation identifying the two zones separately. To proceed with a parametric modelling and identification of the modulus distribution. In this way the distribution of Young’s modulus is parameterized by a fewer parameters. For this particular case, \(E_1\), the modulus of the left part of the specimen, \(E_2\), the modulus of the right part of the specimen, and a parameter \(l_1\), defining the distance from the border between the supposed two materials to the left side of the specimen. Mathematically the elasticity modulus is defined as a function, \(E(x) = E(E_1,E_2,l_1)\). That is to say, this function \(E(E_1,E_2,l_1)\) is assumed to describe completely the distribution \(E(x)\). Then a direct optimization process without sensitivity analysis can be used to find the three parameters that provide the best fit between the simulated and measured displacement response. The direct optimization is performed using MATLAB optimization toolbox [10]. The performances of the parametric method and the proposed finite element model updating procedure are listed in Table-1.

<table>
<thead>
<tr>
<th>Method</th>
<th>Noise level</th>
<th>WAE</th>
<th>Maximum deviation from true value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parametric identification</td>
<td>0 %</td>
<td>0.01</td>
<td>0.1 %</td>
</tr>
<tr>
<td></td>
<td>2 %</td>
<td>0.03</td>
<td>0.3 %</td>
</tr>
<tr>
<td></td>
<td>5 %</td>
<td>0.09</td>
<td>1.1 %</td>
</tr>
<tr>
<td>Adjoint-based FE model update</td>
<td>0 %</td>
<td>0.12</td>
<td>5 %</td>
</tr>
<tr>
<td></td>
<td>2 %</td>
<td>0.28</td>
<td>9 %</td>
</tr>
<tr>
<td></td>
<td>5 %</td>
<td>1.01</td>
<td>27 %</td>
</tr>
</tbody>
</table>

Table-1: Performance of the identification
CONCLUSION

A finite element model updating procedure for the inverse reconstruction of the elasticity modulus distribution with surface displacements measurements is proposed. The problem is formulated as a minimization of a data discrepancy functional augmented with Tikhonov regularization term. The optimisation procedure uses gradient based algorithms. The gradient of the objective functional with respect to the unknown parameters can be solved numerically following an adjoint formulation. Tikhonov regularization and a spatial filtering of the gradients in each iterate are adopted to stabilize the problem in the presence of noise in data. Once the gradient information is obtained, we solve the optimization problem using a gradient-based method. Once the different regions are identified, a second step consists at using a parametric optimisation formulation to smoothen the results.

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GUIDED PLATE ACOUSTIC WAVES TRANSDUCERS FOR STRUCTURAL HEALTH MONITORING

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ABSTRACT

Flexible ultrasonic transducers (FUTs) have been fabricated using a sol-gel spray method. The FUTs consist of 75 µm thick titanium membranes, 135 µm thick lead-zirconate-titanate (PZT) composite films and 10 µm thick silver paste top electrodes. They have been bonded onto metal articles for structural health monitoring and non-destructive evaluation purposes. Two artificial line defects with 1 mm width and depth created on the surface of a 6.35 mm thick aluminum (Al) plate have been successfully detected by such guided wave configurations. In addition, artificial defects perpendicular to the guided plate acoustic wave directions and made on an aircraft representative component were also detected, and the results are confirmed by X-ray techniques. In this investigation the center operation frequencies of the FUTs ranged from 8 MHz to 12 MHz.

\textbf{Keywords}: Structural health monitoring, Non-destructive evaluation, Flexible ultrasonic transducers, Guided plate acoustic waves, Piezoelectric thick films, Sol-gel process.
INTRODUCTION

Civil and military aircraft operators around the world have incurred rising maintenance costs due to their aging fleets. They are seeking ways to reduce the fleets maintenance cost while still meeting airworthiness requirements. Structural health monitoring (SHM) is potentially a cost effective emerging area of technology that enables condition-based maintenance in-lieu of the traditional schedule-based non-destructive evaluation (NDE) [1-3]. Guided acoustic waves (GAWs) are of particular interest in SHM and NDE applications because they provide the ability to conduct large area structural inspections within short period of time using readily available ultrasonic transducers (UTs). For the aerospace industry such sensing tool may be required to operate in an environment where the temperature varies from -80°C to 100°C. In this investigation an approach using newly developed piezoelectric broadband high efficiency flexible ultrasonic transducers (FUTs) bonded to an aluminum alloy (Al) plate structures and on an aircraft representative component to generate and detect GAWs for artificially induced defects. The FUT bonding approach using glue cured at room temperature is a promising on-site sensor instrumentation technique. The center frequencies of such FUTs can range from 3 MHz to 25 MHz [4, 5].

FLEXIBLE ULTRASONIC TRANSDUCERS (FUTs)

The fabrication process of FUTs is based on a sol-gel spray technique [6]. It mainly consists of six steps [4, 5]: (1) lead-zirconate-titanate (PZT) solution preparation; (2) mixing and ball milling of PZT powders and PZT solution to submicron size; (3) spraying of the PZT composite (PZT-c) slurry obtained from step (2) onto 75 µm thick titanium (Ti) membranes; (4) heat treating to produce a thin solid PZT-c ceramic film; (5) corona poling of the PZT-c film to obtain piezoelectricity; (6) top electrode painting using silver paste. Steps (3) and (4) are repeated to achieve desired transducer film thickness that is determined by the desired ultrasonic transducer operational centre frequency and performance. The painting in step (6) provides the convenience of obtaining desired sensor configurations at selected locations. It is noted that the ultrasonic performance of such FUTs on Ti membranes used in this study showed in general 5 to 10 dB stronger signal strength than those reported in [4, 5], whereby FUTs were made onto stainless steel (SS) membranes. The improved signal strength comes from the reduced oxidation of the membrane substrates (Ti over SS) during heat treatment and improvement of the sol-gel spray technique. Figure 1 shows a sample of such FUT with three top electrodes.

Fig. 1: A FUT with three top electrodes fabricated on a 75 µm thick Ti membrane.
DAMAGE DETECTION CAPABILITIES OF A FUT

Al 6061-T6 Plate Test Article

An Al 6061-T6 plate used as a test article for the developed ultrasonic transducer is shown in Figure 2. A FUT similar to the one shown in Figure 1 was bonded onto the 50.8 mm by 6.35 mm cross section area of the aluminum test article using room temperature cured adhesive as shown in Figure 3. Such bonding method is an excellent sensor installation technique due to its in-the-field potential. All silver top electrodes have identical dimensions (~ 4.35 mm by 5 mm) with a thickness of about 10 µm as shown in Figure 3. The FUTs have a center frequency of around 10 MHz. Ultrasonic baseline measurements using these FUTs were taken for a virgin article before the introduction of artificial defects (cracks). The edge bonded FUTs will generate and detect the symmetrical-mode like plate acoustic waves (PAWs) [7, 8] propagating along the plate. The guided PAW velocity of 6246 m/s was obtained using the time delay of the strongest first and the strongest second round trip echoes reflected from the end of the plate opposite to the FUTs, at a distance of 406.4 mm.

Fig. 2: An Al 6061-T6 plate test article.

50.8 mm
406.4 mm
6.35 mm

Fig. 3: FUT bonded onto the cross section of the Al 6061-T6 plate test article.

Two artificial notches, D₁ and D₂, of 1 mm width and 1 mm depth were introduced on the Al 6061-T6 plate, as shown in Figure 4. The one way distances from the FUT (at the edge) to the artificial notches D₁ and D₂ and to the opposite edge E₁ are 145 mm, 222 mm, and 406.4 mm, respectively. Notches D₁ and D₂ have a length of 25.4 mm and 50.8 mm, respectively. Employing the pulse-echo mode and a band-pass filter of 5 MHz to 10 MHz, a comparison between ultrasonic signals measured at transducer A and B in the presence of the artificial defects was made. In Figure 5, G_{D1}, G_{D2}, and G_{E1} represent the echoes reflected from D₁, D₂, and E₁, respectively. In order to see G_{D1} and G_{D2} clearly, the echo G_{E1} was deliberately made to be saturated in this Figure. The shown result provides a calculated time delay of 46 µs, 71 µs, and 130 µs for G_{D1}, G_{D2}, and G_{E1}, respectively. It is noted that since the PAWs are symmetrical-like modes, the higher order modes form the spurious (trailing) signals [9]
shown in Figure 5. Such spurious signals may affect the accuracy of the arrival time which is closely related to the defect location. Proper choice of the PAWs modes may reduce or eliminate such spurious signals [7, 8, 10]. In Figure 5 (b) only echoes from notch D₂ appeared when measured at transducer B. This indicated that for this Al plate of 6.35 mm thick and 50.8 mm wide the guided PAW does not diverge much and this may enable such FUT with array configuration to detect the tip position and propagation of the defect such as D₁ and D₂.

![Diagram showing FUT dimensions](image)

**Fig. 4:** Locations of artificially induced cracks and transducers on the Al 6061-T6 plate.

![Ultrasonic signals measured at different sensors](image)

**Fig. 5:** Ultrasonic signals measured at (a) sensor A and (b) sensor B with the presence of artificially induced cracks illustrated in Figure 4.

**Complex Structural Component**

A complex structural component that is representative of aircraft structural complexity was constructed with artificial cracks at selected rivets locations. This structure shown in Figure 6 is designed to illustrate the structural complexity that could be faced in employing such discussed sensing approach. The length of this structure is 510 mm. Employing electrical discharge machining (EDM) notches were introduced. Notches 1, 3, and 4 are of
2.54 mm long and notches 2, 5, and 6 are of 1.27 mm long. The one way distances from the edge, where a FUT was bonded, to the EDM notches are as identified in Figure 6.

Table 1 summarizes the distance of various EDM notches to the edge bonded FUT and the expected time delay of the ultrasonic echo reflected from the notches using the pulse-echo method. The guided PAW velocity of 6234 m/s used for calculating time delay was obtained by dividing the round trip distance, from the edge bonded FUT to the opposite edge, by the time difference of the two strongest consecutive ultrasonic echoes reflected from the opposite edge of the structure. The actual structure and its FUT edge cross section are also shown in Figure 6.

![Fig. 6: Complex structural specimen with artificially induced EDM notches at specified locations.](image)

**Table 1**: EDM notch locations with respect to the edge bonded FUT

<table>
<thead>
<tr>
<th>Notch</th>
<th>Distance (mm)</th>
<th>Calculated Time Delay (µs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Notch 1</td>
<td>15</td>
<td>4.81</td>
</tr>
<tr>
<td>Notch 2</td>
<td>48</td>
<td>15.40</td>
</tr>
<tr>
<td>Notch 3</td>
<td>117</td>
<td>37.54</td>
</tr>
<tr>
<td>Notch 4</td>
<td>135</td>
<td>43.31</td>
</tr>
<tr>
<td>Notch 5</td>
<td>158</td>
<td>50.69</td>
</tr>
<tr>
<td>Notch 6</td>
<td>176</td>
<td>56.46</td>
</tr>
</tbody>
</table>

**X-ray Inspection**: An X-ray inspection was performed on the complex structural component in order to detect the artificially induced EDM notches. An identical schematic to the one in
Figure 6 and the X-ray films are shown in Figure 7. During the inspection, the X-ray probe needed to be tilted at different angles in order to detect several EDM notches. In addition, it is noted that only the 2.54 mm EDM notches were identified using this technique.

![Image](image.png)

**Fig. 7:** X-ray inspection results of the complex structural component.

Ultrasonic Inspection: The commercial UT V129-10MHz-0.125 from Panametrics was placed at various locations on the cross section shown in Figure 6. The ultrasonic signal measurements were taken using the pulse-echo method. Subsequently, the FUT described earlier was bonded onto the cross sectional surface as shown in Figure 8. The center frequency of the FUT is around 10 MHz. The ultrasonic signals obtained from both the commercial UT V129 and the FUT using pulse-echo method were compared at two sensor locations A and B, separated by 10 mm. Figure 9 shows the ultrasonic signals measured at sensor location A. All top electrodes here have an identical 4×5 mm² dimension. The pulser-receiver settings used to obtain ultrasonic signals for both commercial UT (V129) and the FUT were the same. G_{N2}, G_{N3}, and G_{N6} noted in Figure 9 are the round trip echoes from the EDM notches 2, 3, and 6 that are shown in Figure 6, respectively. The actual time delays of the echoes agree well with the calculated values shown in Table 1. G_{R1} and G_{R2} are the echoes reflected from rivets R1 and R2 between notches 2 and 3, as indicated in Figure 6, respectively. It is observed that the SNR of the signal measured by the FUT is higher than the one measured by the commercial UT V129. Although the EDM notches and rivets are not aligned with sensor location A, echoes from these features may still be seen due to the fact that ultrasound beam width increases over distance.

Another set of ultrasonic signals were measured at sensor location B indicated in Figure 8. In reference to Figure 6, sensor location B is aligned with EDM notch 1. The results are shown in Figure 10. G_{N1} represents the round trip echo generated by EDM notch 1, and G_{R3} is the echo reflected from rivet R3 as shown in Figure 6. The signal measured by the FUT is 8 dB weaker than the one obtained by the commercial V129. For sensor location B, only the first 10 µs time traces are shown as the ultrasonic waves were mostly reflected by EDM notch 1 and other echoes cannot be seen at this level of signal gain.
Fig. 8: FUT bonded onto the 6.35 mm thick section of the complex structural component.

Fig. 9: Ultrasonic signal obtained using (a) commercial Panametrics V129 and (b) bonded FUT at sensor location A indicated in Figure 8.

It has been demonstrated at the two sensor locations that the FUT bonded onto the edge of a plate is capable of detecting defects with high sensitivity, down to 1.27 mm long defects. In addition, the FUT was able to distinguish between rivets and actual defects provided that the rivets and the defects are close to the line-of-sight of the FUT.

CONCLUSIONS AND DISCUSSIONS

Thick composite piezoelectric film sprayed onto 75 µm titanium substrate formed a FUT to excite and detect guided plate acoustic waves. Mass production of the FUT and the potential of ease of on-site installation using room temperature cured adhesive present additional advantage of the technology. The operational temperature of the developed and evaluated FUT is between -80°C and 100°C that is suitable for aerospace structural health monitoring applications. This investigation has demonstrated that the bonded FUTs have
ultrasonic performance as good as commercial ultrasonic transducers while providing additional advantages including their operational temperature range. In addition, the FUT is capable of detecting defects as small as 1.27 mm long in an environment as complex as the one encountered in aircraft structures as illustrated in this document.

![Fig. 10: Ultrasonic signal obtained using (a) commercial Panametrics V129 and (b) bonded FUT at sensor location B indicated in Figure 8.](image)

The FUTs, having characteristics of flexibility for curved surfaces, light-weight and small profile (< 150 µm in thickness), were bonded onto a cross-section edge of 6.35 mm thick aluminum (Al) plates of 50.8 mm width and 406.4 mm length to generate and receive plate acoustic waves (PAWs). The advantage of PAWs is its ability to perform large area inspections. Artificial line cracks were produced along the Al plate. When pulse-echo technique was used, the guided PAWs generated by the FUTs in array form were reflected by the cracks and then detected by the same FUTs. The relationship between the crack length and the strength of the ultrasonic echoes was studied. Experimental results were compared with theoretical calculations. In addition, a section of an aircraft representative complex structure (L-shaped lap-joint) was instrumented with the FUTs that have center frequency of 10 MHz. The FUTs were bonded onto the edges of the 6.35 mm thick Al sections, and ultrasonic results were compared with those obtained by the X-ray technique.

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ABSTRACT

Aerospace structural maintenance (fuselage, wings) is a major component of operational costs which requires aircraft to be grounded and some of its parts to be dismantled in order to proceed to inspection. In order to allow in situ monitoring, Structural Health Monitoring (SHM) has been proposed where sensors and actuators are integrated on the structure. To avoid extensive wiring of the nodes, wireless sensors and actuators are attractive but should be self-powered to fully benefit from them. One idea is to convert the mechanical energy (vibrations) available all over an aircraft into electricity using piezoelectric materials. This work investigates the potential of strain-based energy harvesters (as opposed to inertial harvesters) to supply wireless nodes on typical aircraft structures. A simple model is used to describe typical dynamic behavior of aircraft components: a beam representing the whole wing subjected to atmospheric effects and a plate representing a fuselage panel. Various configurations of piezoelectric materials are tested such as bulk PZT, PZT fiber composite and Polyvinylidene Fluoride (PVDF) in order to evaluate the influence of their characteristics (size, polarization, electrodes’ shape, capacitance…) on the harvested power. The results show that for a typical excitation of the beam (10 Hz and 56 µdef), the energy produced is up to 100 mJ with bulk PZT. From the literature, this appears sufficient for measurements (10 nJ) or RF transmission (25 µJ) but not for both at the same time depending on the kind of piezoelectric material. Therefore, strain-based energy harvester could be used for supplying wireless sensor nodes but they would not allow real time measurement. However this approach is a simple and convenient way to scavenge energy compared to other kinds of harvesters (inertial, solar…) since it amounts to bonding a piezoelectric material on a flexible surface.

Keywords: Power harvesting, energy harvesting, piezoelectric, piezoelectric fiber composite, vibration energy, structural health monitoring.
INTRODUCTION

In applications such as Structural Health Monitoring (SHM) of aerospace structures, sensors and actuators are required to be installed permanently and transmit monitoring signals to a base. In order to reduce the footprint of the wiring required for powering these units, power harvesting from ambient vibration sources has been proposed. Power can be harvested either from tonal sources arising from natural modes of vibrations of wings for example, or broadband turbulent boundary layer or jet noise.

Piezoelectric harvesting devices have been proposed for energy harvesting from ambient vibration sources [1]. Two approaches can be identified for mechanical energy harvesting: i) resonant devices where a mass-spring-damper system formed of a piezoelectric material is excited by vibrations of the supporting structure and ii) non-resonant approaches where the strain of the structure is directly transferred to the piezoelectric material.

Bulk or monolithic piezoceramics (PZT) have been used mostly for energy harvesting and, while they offer a large sensitivity to strain, they suffer from brittleness [2]. Polyvinylidene fluoride (PVDF) piezoelectric films offer an interesting alternative to PZT, with increased material flexibility. A number of electrode patterns have been proposed for PVDF, including inter-digital transducers [3]. However, PVDF suffer from low sensitivity and connection problems. Active Fiber Composites (AFC) [4], and then Macro Fiber composites (MFC) [5], also called Piezoelectric Fiber Composites (PFC), have been more recently developed to offer directional sensing and actuation with electrode connection through screen-printing tools. Greater sensitivity is naturally obtained along the axis of the piezoelectric fibers used in the fabrication of the transducer which are embedded in an epoxy matrix and sandwiched between two sets of inter-digital electrodes (IDT). PFC have been compared with bulk PZTs and Quick Pack actuators for charging a battery and for a given polarization direction [6-7].

It is the purpose of this paper to present a thorough comparison of energy harvesting for different piezoelectric harvesting devices and two polarization directions (when applicable), and evaluate the performance for representative aircraft structural loads.

ENERGY HARVESTING

Electric circuit

A number of electrical circuits have been proposed for harvesting power form a piezoelectric device [8]. Two configurations are presented in the following for dissipation of power in a resistive load and for storage of energy in a capacitor. Although more advanced circuits could have been used, these simple circuits were chosen for comparison purposes between the piezoelectric harvesting devices. It is therefore expected that better performance could be achieved with advanced circuits.

Dissipation in a resistive charge: Figure 1a) presents the circuit which was used to study the effect of the resistive load $R_L$ on the power dissipated and to determine an optimal resistive charge.
As the resistive load has an impedance much higher than the measurement device (oscilloscope), a small measurement resistor $R_i$ (typically 1 kΩ or 10 kΩ) is used to obtain the current $I$ in the circuit. Then, by measuring the voltage $U_i$, current and power in the circuit can be found. For the case of optimal voltage and power, where the maximum power is dissipated, the following can be written:

$$I_{opt} = \frac{U_i}{R_i}$$

(1)

$$U_{Lopt} = (R_i + R_{Lopt}) \cdot I_{opt}$$

(2)

$$P_{opt} = \frac{U_{Lopt}^2}{R_i + R_{Lopt}} = (R_i + R_{Lopt}) \cdot I_{opt}^2$$

(3)

Storage in a capacitor: The circuit presented in Figure 1b) is used to charge a capacitor $C_L$ of 93.4 µF, simulating the charging of a battery for real applications. A switch is used in the circuit to allow the capacitor to discharge when connected to a load (not shown in the figure).

Piezoelectric harvesting devices

A number of piezoelectric harvesting devices have been compared within this work, with different polarization directions, piezoelectric coupling coefficients and type. Table I summarizes the properties of the devices used.

**Table I: Properties of the piezoelectric harvesting devices used in the study.**

<table>
<thead>
<tr>
<th>Capacitance, nF</th>
<th>Polarization</th>
<th>Size of the active in mm (Mass in g)</th>
<th>Piezoelectric coefficient $d_{33}$, pC/N</th>
<th>Type</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>SM - P1 - 0.66 nF</td>
<td>0.66</td>
<td>14.2<em>28</em>0.18 (0.785)</td>
<td>400</td>
<td>Composite</td>
<td>Smart Material Corp.</td>
</tr>
<tr>
<td>SM - P1 - 1.13 nF</td>
<td>1.13</td>
<td>14.2<em>28</em>0.18 (0.785)</td>
<td>400</td>
<td>Composite</td>
<td>Smart Material Corp.</td>
</tr>
<tr>
<td>SM - P2 -</td>
<td>23.2</td>
<td>13.5<em>28</em>0.18 (0.683)</td>
<td>-170</td>
<td>Composite</td>
<td>Smart Material Corp.</td>
</tr>
<tr>
<td>ERF</td>
<td>1.53</td>
<td>20.5<em>41</em>0.25 (2.207)</td>
<td>280</td>
<td>Composite</td>
<td>ERF Produktion Würzburg GmbH</td>
</tr>
<tr>
<td>BM 500</td>
<td>15.29</td>
<td>25.4<em>50.8</em>1 (10.519)</td>
<td>-175</td>
<td>Bulk</td>
<td>Sensor Technology Ltd.</td>
</tr>
<tr>
<td>PVDF (Polyvinylidene fluoride)</td>
<td>1.36</td>
<td>50.8<em>50</em>0.11 (0.508)</td>
<td>-23</td>
<td>Polymer</td>
<td>Measurement Specialties</td>
</tr>
</tbody>
</table>
In order to validate the experimental performance of the piezoelectric harvesting devices in power harvesting with respect to both strain and frequency, a few indications are given. For a piezoelectric, the electrical charges $Q$ generated are proportional to the strain experienced by the device. The optimal current $I_{\text{opt}}$ is the time derivative of the charges and thus, as the frequency of the measured signal increases, this current will increase proportionally. Therefore, the optimal current is proportional to both the strain and the frequency. On the other hand, maximum power will be harvested for a load with impedance equal to the magnitude of the impedance of the piezoelectric device. Since the piezoelectric device behaves essentially as a capacitor its impedance is inversely proportional to the frequency. Therefore, the optimal resistor $R_{\text{opt}}$ will also be inversely proportional to frequency. Following these observations, the global behaviour for optimal power harvested can be written as:

$$
\begin{align*}
R_{\text{opt}} & \propto \frac{1}{\text{frequency}} \\
I_{\text{opt}} & \propto \text{strain} \times \text{frequency}
\end{align*}
$$

which leads to the optimal power:

$$
\langle P_{\text{opt}} \rangle \propto \text{frequency} \times \text{strain}^2
$$

Using constitutive relations of piezoelectricity and assuming that a piezoelectric material can be described as a plane capacitor, the optimal power becomes:

$$
U_{\text{opt}} \propto \frac{d_{3x} \cdot S_x \cdot t}{K_T}
$$

$$
\langle P_{\text{opt}} \rangle \propto \frac{C \cdot t^2}{K_T^2} \cdot d_{3x}^2 \cdot S_x^2 \cdot f
$$

where $t$, $S$, $K_T$ are the inter-electrodes space, strain, and dielectric constant, respectively.

**EXPERIMENTAL SETUP**

Non-homogeneous flow encountered by an aircraft will translate into various types of excitation on the structure. Among these excitations, one very energetic one will be in the form of flexural vibration of the wings. It was chosen in this work to simulate such an excitation by using a simple clamped beam with representative strain (56 $\mu$def) and frequency (10 Hz) [9-10].

Figure 2a) presents the beam used while Figure 2b) presents one the piezoelectric harvesting devices used. A number of identical beams with dimensions 542 mm x 50.8 mm x 6.35 mm were used and excited at the free end with a shaker capable of generating large displacement at low frequency. The piezoelectric harvesting devices were bonded close to
the clamped end, where maximum strain levels are obtained for such a configuration, and maximum harvested power will be obtained.

![Image]

**Fig. 2**: a) Clamped beam and b) piezoelectric fiber composite SM P1 – 1.13 nF.

The shaker excites the beam at a given frequency and with a given displacement. The shaker is driven by a frequency generator and connected to the beam through a stinger. In order to compare all piezoelectric harvesting devices under similar conditions, excitation levels are adjusted such that the same strain is measured by the devices. The strain is simulated through analytical calculation from the displacement obtained from time integration of either acceleration or velocity measured at the excitation point using flexural beam theory under clamped-free conditions. The acceleration is measured with an accelerometer bonded at the excitation point while the velocity is measured using a laser vibrometer at the same point. This approach allows each piezoelectric device to experience the same strain level.

**RESULTS**

The experimental results are presented for the piezoelectric harvesting devices listed in Table I. Results are presented for both power dissipation in a resistive load and storage of energy in a capacitor.

**Dissipation in a resistive load**

Figure 3 shows the dissipated power as a function of the resistive load \( R_L \) for the operating point \{10 Hz; 56 \( \mu \)def\}. The first observation is that the similar behaviour is obtained for different piezoelectric harvesting devices, where a clear maximum can be localized, corresponding to the optimal resistive load \( R_{Lopt} \). As already mentioned, this maximum dissipated power is obtained when the magnitude of the impedance of the load, \( R_{Lopt} \), is equal to the magnitude of the impedance of the piezoelectric harvesting device, \( |Z_p| \). Figure 3 also shows that the optimal resistive load \( R_{Lopt} \) varies for different piezoelectric harvesting devices.
First observations show that two groups can be distinguished: i) piezoelectric harvesting devices polarized in {3-3} (Figure 3a) and ii) piezoelectric polarized in {3-1} (Figure 3b). In fact, a factor over 10 has been obtained for the optimal resistive loads $R_{Lopt}$ between both groups. This observation can be explained by the different capacitance associated with both groups, as shown in Table II.

As shown in Figure 4 for the bulk piezoelectric (BM 500), the optimal resistive load also depends on the excitation frequency. The level of excitation itself has no impact on the value of the optimal resistive load but will obviously affect the level of dissipated power, as discussed next.

Figure 5 shows the dissipated power density (per unit volume) as a function of both the strain level (Figure 5a) and the frequency (Figure 5b). The maximum level for dissipated power (650 $\mu$W/cm$^3$) is achieved for an excitation at 18 Hz and a strain level of 56 $\mu$def for the piezoelectric device SM P1 – 1.13 nF.
Figure 5 also compares the various piezoelectric harvesting devices for the same excitation. Again, two groups are distinguished: i) piezoelectric harvesting devices polarized in \{3-3\} and ii) piezoelectric harvesting devices polarized in \{3-1\}.

Among those polarized in \{3-3\}, the SM P1 – 1.13 nF – provides more power than the SM P1 – 0.66 nF – and the ERF. This illustrates the fact that for similar piezoelectric coefficients (SM P1 0.66 and 1.13 nF), the power obtained is greater for a higher capacitance. In fact, for an increase of 71 % (|0.66 - 1.13| / 0.66) in the capacitance, the mean power dissipated increases by 28 % (|265.31 - 339.99| / 265.31). However, the performances of the piezoelectric device SM P1 – 0.66 nF – are better than those of the ERF (1.52 nF) even if the capacitance of the latter is higher by 132 %. It therefore appears that the piezoelectric coupling coefficient has a greater effect on the dissipated power than the capacitance of the piezoelectric material polarized in \{3-3\}.

As for the devices polarized in \{3-1\}, the SM P2 provides the most dissipated power, in front of the BM 500. Their piezoelectric coupling coefficient is similar but their capacitance is different. The capacitance is higher for the SM P2 (23.2 nF) than for the BM 500 (15.29 nF). This also confirms that higher dissipated power is obtained with higher capacitance values.

Finally, the two groups of polarization can be compared. It appears that the SM P1 – 1.13 nF – provides the highest power dissipated power, in front of the SM P2 and the SM P1 – 0.66 nF -. The selected PFCs produce the most power among the piezoelectric harvesting devices tested. On the other hand, the ERF provides the least dissipated power. One explanation for this would be the quality of fabrication which seems to be beyond the quality of fabrication of other PFCs: fibers are not properly aligned and do not have the same length. Another explanation is that the fibers for ERF are circular instead of rectangular, which impairs the contact between the electrodes and the piezoelectric material itself. It should however be mentioned that the PFCs from ERF are not to be used in small patches like the ones used in this paper but rather be integrated into larger structures (such as wings or fuselage) for excitation and measurement.
Table II presents the estimated power density calculated using Eq. (7) for the same operating point \{10 Hz; 56 \mu\text{def}\}. The observations made earlier correlate well with the predictions in Table II. The SM P1 – 1.13 nF – has the highest power density while the ERF is the lowest one. The other three piezoelectric harvesting devices have similar power density.

<table>
<thead>
<tr>
<th></th>
<th>Relative permittivity</th>
<th>Piezoelectric coefficient $d_{33}$, pC/N</th>
<th>Capacitance, nF</th>
<th>Inter-digital spacing, mm</th>
<th>Volume, mm$^3$</th>
<th>$C_{\text{eff}} d_{33} / K$</th>
</tr>
</thead>
<tbody>
<tr>
<td>SM - P1 - 0.66 nF</td>
<td>1850</td>
<td>400</td>
<td>0.66</td>
<td>0.5</td>
<td>70.56</td>
<td>0.1093</td>
</tr>
<tr>
<td>SM - P1 - 1.13 nF</td>
<td>1850</td>
<td>400</td>
<td>1.13</td>
<td>0.5</td>
<td>70.56</td>
<td>0.1872</td>
</tr>
<tr>
<td>SM - P2 -</td>
<td>1850</td>
<td>-170</td>
<td>23.2</td>
<td>0.18</td>
<td>65.52</td>
<td>0.0969</td>
</tr>
<tr>
<td>ERF</td>
<td>1850</td>
<td>280</td>
<td>1.53</td>
<td>0.5</td>
<td>210.21</td>
<td>0.0417</td>
</tr>
<tr>
<td>BM 500</td>
<td>1750</td>
<td>-175</td>
<td>15.29</td>
<td>1</td>
<td>1290.32</td>
<td>0.1185</td>
</tr>
</tbody>
</table>

Storage in a capacitor

In this part of the work, the resistive load is replaced by a capacitor to investigate the performance of the piezoelectric harvesting devices in an energy storage configuration, using the electrical circuit presented in Figure 1b). The capacitor used in the following was chosen to have the value 93.4 \mu F, for comparison purposes between the piezoelectric harvesting devices. In a practical application, the capacitor would be used to store energy that would later be used to power a sensor or an actuator for structural health monitoring.

In the following, the time required to charge the capacitor is presented together with the power transferred by the piezoelectric harvesting device to the capacitor and the energy stored in the capacitor.

Fig. 6: Charging of the capacitor (93.4 \mu F).

The charging time of the capacitor is first presented in Figure 6. It appears that the charging time varies significantly with the type of polarization. Piezoelectric harvesting devices polarized in \{3-1\} (SM P2 and BM 500) charge much faster than the piezoelectric harvesting devices polarized in \{3-3\}. This is explained by the fact that the current generated by the devices polarized in \{3-1\} is much larger then the current generated by those polarized in \{3-3\}. This means that for a given time period, more electrical charges are collected at the
electrodes. Nevertheless, once normalized by the volume of the piezoelectric material, the final voltage reached by the piezoelectric harvesting devices polarized in \{3-3\} is much higher. This was expected since the final voltage is proportional to the piezoelectric coupling coefficients \(d_{3x}\) and the piezoelectric harvesting devices polarized in \{3-1\} have larger coefficients.

Figure 7 presents the energy density. Higher final voltage is associated to higher energy density \(\mathcal{E}_v\) for piezoelectric harvesting devices polarized in \{3-3\}:

\[
\mathcal{E}_v = \frac{1}{2} \cdot C_{st} \cdot U^2
\]  

However, when plotting the power transmitted by the piezoelectric harvesting device at 95% of the maximum voltage (Figure 8), the piezoelectric harvesting devices tend to behave in a more similar manner. This is due to the faster charging of the piezoelectric harvesting devices polarized in \{3-1\}. Thus, although the quantity of energy is smaller for this latter case, the power transmitted is nearly equivalent due to shorter charging time.

CONCLUSIONS

This paper presented an experimental comparison of different piezoelectric energy harvesting devices for application in structural health monitoring. An experimental setup was proposed to reproduce simple vibration behaviour of an aircraft wing with respect to frequency and strain level.

Two distinct groups of piezoelectric harvesting devices were investigated: i) piezoelectric harvesting devices polarized in \{3-3\} and ii) piezoelectric harvesting devices polarized in \{3-1\}. Two configurations were tested: i) power dissipation in a resistive load and ii) energy storage in a capacitance.

It was noted that the optimal voltage is linearly dependent on the strain, but independent of the frequency. The optimal current was shown to be linearly dependent on both the frequency and the strain level. The power dissipated in a resistive load was shown to be linearly dependent on the frequency but quadratically related to the strain level. As for the
piezoelectric material itself, the dissipated power is a linear function of the capacitance and inversely proportional to the relative permittivity. This dissipated power is also a quadratic function of the inter-digital spacing and the piezoelectric coupling coefficient.

Literature indicates that power density in the order of $100 \mu W/cm^3$ is satisfying for many applications, including structural health monitoring. The results presented in this work are in accordance with this. To increase the harvested energy, larger devices will be required.

Future work will investigate the use of more advanced harvesting circuits and integration with a structural health monitoring device capable of generating waves into the structure and measuring its response.

ACKNOWLEDGEMENTS

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REFERENCES

ENERGY HARVESTING AND EMBEDDED RF WIRELESS SENSOR SYSTEMS FOR HELICOPTER BLADES

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ABSTRACT

Helicopter blades bending out of the rotating plane, due to the long, narrow geometry and aerodynamic forces, cause undesirable vibration to be transmitted through the hub and into the fuselage of the aircraft. Current measurement techniques take data from accelerometers mounted in the fuselage. The signals are effectively “filtered” from the source in the rotating frame and control inputs often do not accomplish vibration suppression as predicted. With advances in the microcontroller design, miniature sensor fabrication, as well as low-power wireless transmission technology, a self-powered RF wireless sensor system can be realized and installed on a helicopter blade to provide measurements directly from the source.

A proposed aeroelastic energy harvesting device is modeled at the trailing edge of the blade and initial attempts at relating its geometric parameters to vibratory frequency of the piezoelectric energy harvesters are reported. In addition, a typical RF wireless sensor package is assumed and a power budget is analyzed in order to gauge the amount of power required from an energy harvesting design. Lastly, flight conditions in hover and forward flight are examined for their effect on power extraction.

Keywords: Abstract, Papers, Materials, Structures, Helicopters, RF, Wireless, Energy Harvesting
INTRODUCTION

The motivation to attach a piezoelectric energy harvesting device to the trailing edge of a helicopter blade comes from the desire to power embedded systems locally on the helicopter blade instead of routing wires up from a fixed frame to the rotating frame. The exclusion of a slip ring greatly simplifies a system and reduces the amount of maintenance as slip rings require continual upkeep to avoid signal degradation of the rotating electrical connections caused by normal wear and debris contamination. If a piezoelectric bimorph energy harvester with a flat plate component is mounted on a helicopter blade in the rotating frame, it can be a local power source for an embedded wireless sensor system; however, the energy from flow induced vibrations will require power management and switching techniques for intermittent operation of the load due to the relatively large power requirements. This type of energy-autonomous battery-less sensor operation has been discuss in conjunction with a low-power radio frequency (RF) transmitter for short-range links (M. Ferrari et al. 2009). Additionally, the electromechanical modeling of a piezoelectric bimorph has been in discussed in several papers (Sodano et al. 2004, Roundy et al. 2004) and is combined with an aeroelastic model for analyzing flutter frequency and flutter airspeeds (Peters et al. 1995, Hodges 2002) in order to relate induced velocity from the rotation of the blade to the frequency at which the energy harvester will oscillate at its first bending mode.

This paper aims to look at the bending frequencies during the method of power harvesting for hover conditions as well as forward flight and discusses the power requirements of modern day RF wireless sensor nodes. Naturally, the inclusion of wireless transceivers is favorable in order to avoid sending data through wires from the fuselage to the blades in the rotor system. By introducing the concept of a self-powered wireless embedded system to a helicopter blade, data can be processed locally and sent at preset intervals to monitor the blades’ performance and health.

ENERGY HARVESTING DEVICE MODELING

A novel energy harvesting device has been researched at the Laboratory of Intelligent Machine Systems (LIMS) in Cornell University. An elastic beam with piezoelectric bimorphs attached at the root was combined with a wing section such as a flat plate. The connection is a revolute or flexural hinge joint. **Fig. 1** shows a side view of the device and its intended purpose at the trailing edge of a helicopter blade. The length of the elastic beam element has been exaggerated to show the first bending mode of the proposed thin beam. Using Euler-Bernoulli beam theory and the Rayleigh Ritz method, an electromechanical model was developed (Sodano 2004) in conjunction with using a p-method flutter analysis in an unsteady aerodynamics model which accounts for circulatory and non-circulatory flow terms (Hodges 2002, Peters 1995). The governing equations of the combined wing section and piezoelectric bimorphs were derived by applying Lagrange’s equations (Bryant et al. 2009).
Fig. 1: Side view of aeroelastic energy harvesting device attached to trailing edge of helicopter blade. The figure shows three different bending states of the elastic beam.

Fig. 2: Top view of aeroelastic energy harvester with flexural joints.

The wing section shown in Fig. 3 can be modeled as a simple 2 degree of freedom (DOF) lumped mass from (Hodges et al. 2002) with springs and dampers acting in both the $i_2$ direction called “heaving” and in the $i_1$ and $i_2$ plane called “pitching”. The variables $h$ and $\theta$ are used to respectively as coordinate variables. $b$ is the airfoil semichord, $a$ is the dimensionless parameter for identifying the elastic axis point, $P$, and $e$ is the dimensionless parameter for identifying point $C$, the center of gravity position. Equations of motion can then be formulated as Langrange equations to capture the kinetic and potential energies of the entire energy harvester electrical and mechanical system.
Standard aeroelastic analytical equations are paired with Euler-Bernoulli piezoelectric bimorph equations (Hodges et al. 2002, Sodano et al. 2004) to produce:

\[ T = \frac{1}{2} m_w \dot{h}^2 + m_c h \dot{\theta} + \frac{1}{2} I_p \dot{\theta}^2 + \int \rho_s u_s^T \dot{u} dV_s + \int \rho_p u_p^T \dot{u} dV_p \]  

(1)

\[ V = \frac{1}{2} k_p \theta^2 + \int S^T c_S S dV_S + \int S^T c_E S dV_E - \int E^T e_p S dV_p - \int E^T \varepsilon^S E dV_p \]  

(2)

where \( T \) and \( V \) are the kinetic and potential energies of the entire system. \( m_w \) is the mass of the wing section while \( m_c \) represents the total mass of the flat plate and flexural hinge structure. \( I_p \) is the mass moment of inertia of the wing section about point P. \( \rho_s \) and \( \rho_p \) are elastic beam and piezo film densities while \( V_s \) and \( V_p \) are elastic beam and piezo film volumes. Beam displacement is the variable \( u \) while S and E are elastic beam strain and electric field respectively. The term \( c_S \) is the substrate elastic modulus, \( c_E \) is the piezo elastic modulus at short circuit, \( e_p \) is the piezo coupling coefficient, \( \varepsilon^S \) is the piezo dielectric or permittivity constant at constant strain. The parameter \( x_\theta \) is the airfoil static unbalance given by:

\[ x_\theta = e - a \]  

(3)

Next, we express the first bending mode of the elastic beam as a Rayleigh-Ritz modal summation

\[ u(x,t) = \sum_{i=1}^{N} \phi_i(x) r_i(t) = \phi(x) r(t) \]  

(4)

where \( r(t) \) is the beam deflection coordinate and \( \phi(x) \) is the first beam mode shape. The wing section heave coordinate \( h(t) \) can be assumed to be equivalent to beam deflection at the hinge. Therefore, \( h(t) \) can be expressed in terms of the beam deflection coordinate and the mode shape evaluated at the beam tip as:
\[ h(t) = u(t, t) = \phi(t) r(t) \]  \hspace{1cm} (5)

The generalized forces acting on the system in the heave and pitch directions can be expressed as:

\[ Q_h = -\phi(t)L(t) \]  \hspace{1cm} (6)

\[ Q_o = M_{1/4}(t) + b\left(\frac{1}{2} + a\right) L(t) \]  \hspace{1cm} (7)

where \( L(t) \) is the total lift force and \( M_{1/4}(t) \) is the total pitching moment about the \( \frac{1}{4} \) chord of the airfoil. Lagrange’s equation is then applied to Equations (1) and (2) with the variable transformations given by (4) and (5) and the generalized forces given by (6) and (7). After grouping terms, the governing equations for the electromechanical system can be written as:

\[
\left( M_S + M_P + m_L \phi(t)^2 \right) \ddot{r}(t) + m_L h x_0 \phi(t) \ddot{\theta}(t) + (K_S + K_P) r(t) - \Theta C_p^{-1} q(t) = -\phi(t) L(t) \]  \hspace{1cm} (8)

\[
m_L h x_0 \phi(t) \ddot{r}(t) + I_\phi \ddot{\theta}(t) + k_\theta \theta(t) = M_{1/4}(t) + b\left(\frac{1}{2} + a\right) L(t) \]  \hspace{1cm} (9)

\[
R \dot{q}(t) - C_p^{-1} \dot{\Omega} r(t) + C_p^{-1} q(t) = 0 \]  \hspace{1cm} (10)

where the generalized coordinates are beam deflection \( r(t) \), wing pitch deflection \( \theta(t) \), and charge \( q(t) \). The reader is referred to (Bryant et al. 2009) for further derivation of the aeroelastic modeling for \( L(t) \) and \( M_{1/4}(t) \) and (Peters et al. 1995) for the calculation of induced flow states and induced velocities using unsteady aerodynamic modeling.

![Flutter Speed vs Frequency](image)

Fig. 4: Plot of predicted airspeed at onset of flutter and bending frequency of elastic beam

With an electromechanical and aerodynamic model in place, simulations were run to approximate frequencies of the elastic beam’s first bending mode at the onset of flutter as shown in Fig. 4. The geometric parameters of the energy harvester were set to approximate the dimensions of a prototype that’s yet to be tested. Additionally, the length of the beam was varied in order to get a qualitative understanding of the relationship between oscillating
frequency and airspeed at the onset of flutter. We assumed the induced airspeed at the trailing edge of a helicopter blade will exceed critical flutter airspeed and excite the elastic beam due to a placement of the device approximately 3.0m from the center of rotation. The energy harvesting device will experience induced airspeeds of approximately 80 m/s in steady hover and vary sinusoidally between 0 m/s and 160 m/s at 80 m/s (155 knots) forward flight as seen in Fig. 5 and Fig. 6. The piezoelectric films attached at the base of the elastic beam, convert mechanical to electrical energy where strain levels are the highest. Results show that the elastic beam bending frequency at the onset of flutter is approximately linear with induced flow airspeed. In hover, we can expect oscillations around 120 Hz while in forward flight, we can expect a range from 0 Hz to approximately 242 Hz.

![Induced Airspeed vs. Azimuth Angle](image)

**Fig. 5:** Induced airspeed as a function of azimuth angle during blade rotation

Currently, power required for each individual helicopter blade is passed from the fuselage up to the rotor head and through a slip ring to the blades. While the use of batteries can be employed, performance in the cold environments experienced by helicopter blades and the maintenance issue of replacing the batteries decreases their appeal. By applying the piezoelectric energy harvesting device to the trailing edge of helicopter blades, strain energy can provide power to on-blade embedded systems.
RF wireless development boards are now readily available commercially and some offer extremely low power consumption to the point where one can take into account alternate ways to power them that was previously impossible. As an example, an embedded wireless sensor system comprising of a Texas Instruments (TI) MSP430F2274 microcontroller chip, a TI RF2500 2.4 GHz radio frequency antenna, and a MEMS manufactured triaxial digital output accelerometer is considered. For the purposes of monitoring acceleration at certain intervals, we do not task the microcontroller with heavy processing. In such a case, the MSP430F2274 processing frequency was lowered to 1MHz and current consumption subsequently decreased. In addition, the radio chip transceiver transmits signals at 250 Kbytes per second so transmission for a 32 Kbyte packet takes approximately 100-300 milliseconds. Setting the communication rate to transmit and receive a packet every 5 seconds allows monitoring of the condition and performance of the blades. Unfortunately, the radio chip demands quite a lot of energy during active operations and contributes most to the total system cost of 57.44 mW to 64.64 mW depending on transmitting or receiving. When the system is in standby mode, the microprocessor is in the quiescent mode and the radio chip is in low-power mode. The power requirement drops to a modest 8.4 μW. Table 1 shows a summary of the system in active and standby power savings mode and duty cycles based on a 4 hour flight as most commercial and military helicopters have endurance limits of less than 4 hours. Future tests are planned to validate the power budget with experimental results and compare them with energy harvesting findings at the trailing edge of a helicopter blade.
### Table 1: Power Budget for RF Wireless Embedded Accelerometer System

<table>
<thead>
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<th>Mode</th>
<th>Power (μwatts)</th>
<th>Minutes (per 4 hour flight)</th>
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**CONCLUSIONS**

This paper described the concept of incorporating a self-powered embedded wireless sensor system on a helicopter blade. The idea of using a novel aeroelastic piezoelectric energy harvester as a means to generate power has been presented. Simulations using a combination of electromechanical and aeroelastic modeling techniques resulted in predictions for bending frequencies of the elastic bender. We predict that given the placement of an energy harvester on the blade, beam bending frequencies of approximately 120 Hz can be realized. In forward flight, due to varying induced velocities as a function of azimuth angle of the blade rotation, bending frequencies will vary approximately between 0 Hz and 242 Hz. Additionally, a discussion on an example RF wireless sensor system was introduced with an estimated power budget. The transmission of data using the radio chip is shown to be responsible for the majority of the power required for such a system.

**REFERENCES**

A DYNAMICAL SENSING MODEL FOR MAGNETOSTRICTIVE GALFENOL WITH APPLICATIONS TO ENERGY HARVESTING

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Keywords: Energy Harvesting, Galfenol, Rate Equations

ABSTRACT

A computational model for the quasi-static and dynamical response of Galfenol based magnetostrictive devices in the sensing configuration is developed. The model calculates the fraction of magnetic moments oriented along each of the energetically preferred directions of the crystal as a function of time using a self-consistent rate equation technique. These magnetic moment fractions can then be used to determine the total magnetization as a function of time. The model is compared to experiments in the case of uniaxial, compressive and quasi-static loading. Using the parameters obtained by the comparison to experiment, predictions for magnetization and energy harvesting under dynamical loading conditions are presented.
INTRODUCTION

Magnetostrictive devices have attracted much attention with the emergence of iron-gallium alloys (Galfenol), which have a number of favourable mechanical properties [1, 2]. This article presents a dynamical numerical model for the sensing (Villari) effect of a Galfenol sensor and an energy harvesting system based on it.

The simplest dynamical sensing model is one where the stress and magnetization are assumed to be linearly related [3,4]. While this is quick and efficient to implement, it cannot incorporate hysteretic and non-linear effects and would therefore have significant limitations in higher frequency applications. Atulasimha et al. [1,5,6] calculate the magnetization and strain with a thermodynamic average of a Boltzmann distribution using the free energy described by Armstrong[7]. Their simulations agree well with experiments, but only in the quasi-static regime and they do not include hysteresis effects. Smith et al.[8,9] used a “homogenized energy” model that incorporates thermal and magnetic after-effects. It is also restricted to the quasi-static case and only accounts for moments aligned parallel or anti-parallel to the crystal axis. Evans and Dapino [10,11] based their models on Smith et al.[8] to create a comprehensive model of the actuation application of Galfenol. They have also presented some simulated sensing results which include hysteretic and dynamical effects [11]. A self-consistent iteration is performed which requires a recalculation of the thermal equilibrium distribution at each iteration. This is a computationally lengthy process, especially if the model is applied to a fully three-dimensional system with stresses and fields applied off-axis.

In this work, a dynamical sensing model based on rate equations is constructed that will include hysteresis and eddy-currents. The moment distributions and transition times of Evans and Dapino [10,11] are simplified to reduce computation time, which will make it more amenable for future simulations of energy harvesting devices. First, a 1D sensing model that is appropriate for materials with a uniaxial symmetry will be presented [12]. Next, a 3D sensing model will be presented that has more flexibility in geometry and loading conditions. Both of these models assume that all magnetic moments are aligned along the preferred orientations with no thermal effects. Finally, as a sample application to energy harvesting, a simple Galfenol based AC energy harvester is modelled which will couple the 1D sensing model to equations for the harvesting circuitry.

The sample has a low aspect ratio being 6.35 cm in diameter and 2.54 cm in length. These dimensions, a Young’s modulus of approximately 60GPa and a density of 77.1kg/m³ give a resonant frequency of 244 kHz. Since we are simulating response for frequencies well below this value, coupling to the mechanical system is therefore neglected in the model.

1. A 1D MODEL WITH NO THERMAL EFFECTS, THEORY AND SIMULATIONS

A. Free energy of a single magnetic moment

The free energy formulation given by Atulasimha et. al. and Evans et. al [5,10] based on Armstrong[7] is used which incorporates magnetocrystalline, magnetoelastic, magnetic and exchange interactions. For a magnetic moment of magnitude Mₛ and with normal stress σ and
magnetic field $H_z$ both applied along the [001] ($\hat{z}$) direction, the free energy can be approximately written as

$$E = K_1 \cos^2(\theta) \sin^2(\theta) - 3\alpha_{100} \cos^2(\theta)/2 - \mu_o M_z H_z \cos(\theta) - J_{\text{exch}} M_z M_z \cos(\theta)$$

(1)

where $\theta$ is the angle of the magnetic moment with respect to the $\hat{z}$ direction, $K_1$ is the crystalline anisotropy in the $\hat{z}$ direction, $J_{\text{exch}}$ is the phenomenologically determined exchange coefficient, $\mu_o$ is the permeability of free space and $M_z$ is the average magnetization in the $\hat{z}$ direction (which will be determined iteratively using equation (2)).

Local minimums of the free energy of equation (1) represent the preferred moment orientations. In this crystal there are three minimums at low stress and field bias, but higher fields and stresses can overpower the crystal anisotropy to reduce the number of local minimums.

### B. Distribution of moment orientations and transition probabilities

In order to find the total magnetization of a material, it is necessary to sum the magnetic moments residing at each orientation. Assuming no thermal effects, each magnetic moment will reside at one of the preferred moment orientations. The fraction of the total number of moments aligned with the $i^{th}$ preferred orientation will be denoted by $\xi_i$, and referred to as the preferred orientation fraction. In this notation, the total magnetization in the system is then

$$M_z = M_z \sum_i \xi_i \cos(\theta_i)$$

(2)

where $\theta_i$ is the $i^{th}$ preferred orientation angle. Due to axial symmetry, the average magnetization components in the $\hat{x}$ and $\hat{y}$ directions will vanish. In this approximation, the transition times between the orientations (as shown schematically in Figure. 1) are best modelled as $t_{ij} = \tau_s (E_{\text{barr},i} - E_{\text{well},j})^2$, where $\tau_s$ is a phenomenological parameter for the thermal scattering time.

### C. Evolution of volume fractions: Rate equations

Now that the transition times have been defined, rate equations can be created to describe the evolution of $\xi_i(t)$. The rate equations come from the master equation and are used to describe, phenomenologically and to first order, the transition rates between a discrete set of states[13]. They are commonly used in many fields such as chemical reaction kinetics[14] and photonics[15]. With transitions restricted to adjacent orientations and a maximum of three preferred orientations the rate equations are

$$\dot{\xi}_1 = -\xi_1(t)/t_{12} + \xi_2(t)/t_{21}, \quad \dot{\xi}_2 = \xi_1(t)/t_{12} - \xi_2(t)/t_{21} + \xi_3(t)/t_{32} - \xi_3(t)/t_{31}$$

and

$$\dot{\xi}_3 = -\xi_3(t)/t_{32} + \xi_2(t)/t_{23}$$

(3)
Fig. 1. The free energy as a function of orientation angle and the locations of wells, barriers and transition times.

D. Eddy-current losses

The set of equations defined so far are only appropriate for the quasi-static regime (where stress is slowly varying). Eddy current losses occur because in a dynamic system, a change in magnetic field will induce an electric field and current flow, which will in turn induce its own magnetic field that opposes the original change. From Maxwell’s equations, for an ohmic material, the eddy-currents induced are proportional to the rate of change of the magnetic field and the eddy-current induced magnetic field is proportional to the eddy-current. Therefore the effect is modelled phenomenologically as

$$\delta H_z \approx A_{edd} \dot{M}_z, \quad \dot{M}_z = M_s \sum_i \dot{\xi}_i \cos(\theta_i)$$

(4)

Where $\delta H_z$ is the eddy-current induced magnetic field and the term $A_{edd}$ is used as a fitting parameter, which would be proportional to the geometry and conductivity of the material.

E. 1D Results

The exchange interactions and the eddy-current effects make the system of equations non-linear. Equations (1) to (4) must therefore be solved self-consistently. Parameters $K_1 = 1.75 \times 10^4$ J/m$^3$, $\lambda_{100} = 225$ microns, $M_s = 1.66$T, $J_{exch} = 6 \times 10^{-4}$, $\tau_s = 1.0 \times 10^{-11}$ s/J$^2$ and $A_{edd} = 3.0 \times 10^{-3}$ are used to simulate a uniaxially compressively loaded <100> oriented Galfenol crystal with 19% atomic weight Ga. These parameters are chosen to fit as close as possible to the experimental results of Atulasimha and Flatau [1,5,16].

In Figure 2 the magnetization over one period versus stress is plotted for current simulations and the experiments of Atulasimha and Flatau [1,5,16] in the quasi-static regime.
(0.01Hz) loading condition at various magnetic field biases. The figure displays hysteretic losses in the system even in quasi-static loading, which is most likely due to the magnetic circuit effects of the experimental apparatus. As shown, it can be incorporated into the model with appropriate choices of parameters.

At low bias fields, the simulations and experiment compare well but they deviate significantly for higher bias fields. It will be necessary to incorporate thermodynamic affects more accurately in the higher bias range. However, for many practical applications such as energy harvesting, it is not of interest to use high bias fields.

**Fig. 2.** The total magnetization versus uniaxial compressive stress in the quasi static regime. Solid lines are the 1D simulations, dashed lines experimental [5,16]. The curves are shown for increasing levels of bias field at 23, 44, 66, 89, 111, 167, 223 and 446 Oe.

For the energy harvesting application, the most efficient harvester would be one where all changes in stress result in a change in magnetization. In figure 2 one observes regions where there is “wasted stress”. For example, on the 111 Oe curve for stress less than 30MPa and for stress greater than 55MPa, there is little change in magnetization as the stress changes. The model presented therefore provides an insight into setting appropriate conditions for maximum harvesting efficiency.

In figure 3 the magnetization versus compressive stress for the 23 Oe bias field is plotted again. This time the maximum compressive stress is chosen to be 25MPa, which is around where the magnetization stops changing significantly. Also shown in this graph are the effects of eddy-currents and scattering time on hysteresis as the frequency increases. As the frequency increases, the hysteresis increases and the amplitude of the magnetization decreases. These effects are mainly due to the thermal scattering time.
The change in magnetization with respect to time is important in energy harvesting because this will be proportional to the induced voltage in a coil wrapped around the magnetostrictive material. In figure 4, the change in magnetization times the period is plotted over the period of stress variation. It is observed that the peak decreases with increasing frequency which suggests there will be a maximum change in magnetization (and hence generated voltage) obtained. Note that the curves in figure 4 are periodic, but are not sinusoidal because of the non-linear nature of our model. For simplicity, it is sometimes assumed that there is a linear relation between the stress and the induced magnetization \[3,4\]. In that case, if the stress variation was sinusoidal, then the magnetization variation would also be sinusoidal.

**Fig. 3.** Total magnetization versus uniaxial compressive stress at 23 Oe magnetic field bias for various frequencies.

**Fig. 4.** The period times change of magnetization over one period of stress variation at maximum stress 25 MPa and 23 Oe magnetic field bias for various frequencies.
2. A 3D MODEL WITH NO THERMAL EFFECTS, THEORY AND SIMULATIONS

A) Free energy of a single magnetic moment

The free energy of a cubic crystal extended to 3D is given in given by Atulasimha et al. and Evans et al. [5,10] based on Armstrong [7]

\[
E = K_1 \left[ \sin^4 \phi \cos^2 \theta \sin^2 \theta + \sin^2 \phi \cos^2 \phi \right] + K_2 \sin^4 \phi \cos^2 \theta \sin^2 \theta \\
- \frac{3}{2} \lambda_{111} \left[ \sin^2 \phi \cos^2 \theta \sigma_{xx} + \sin^2 \phi \sin^2 \theta \sigma_{yy} + \cos^2 \phi \sigma_{zz} \right] \\
- 3 \lambda_{111} \left[ \sin^2 \phi \sin \theta \cos \theta \sigma_{xy} + \sin \phi \cos \phi \sin \theta \sigma_{yz} + \sin \phi \cos \phi \cos \theta \sigma_{xz} \right] \\
- \frac{M_z}{\mu_0} \left[ \sin \phi \cos \theta (H_x + J_{exch} \sigma_{xx}) + \sin \phi \sin \theta (H_y + J_{exch} \sigma_{yy}) + \cos \phi (H_z + J_{exch} \sigma_{zz}) \right]
\]  

(5)

where \( \sigma_{ii} \) are the axial stresses, \( \sigma_{ij}, i \neq j \) are the shear stresses, \( K_i \) is the second crystalline anisotropy coefficient and \( \lambda_{111} \) is the shear magnetostriction coefficient.

B) Transition Times to Adjacent Preferred Orientations

Once again, to reduce computation time, it is desirable to only consider moment rotations between adjacent preferred orientations. In 3D, this involves more care than the 1D case. One must:

a) Find all minimums in free-energy (preferred orientations).

b) Decide which minimums are adjacent (nearest neighbours) to each other.

c) Calculate the energy barriers and thus transition times between adjacent orientations.

i) Minimums

The minimums of free energy can be found using the criteria for minimums of functions of two variables \( E_{\phi\phi} E_{\theta\theta} - E_{\phi\theta}^2 > 0 \), \( E_{\phi\theta} > 0 \) and \( E_{\phi\phi} > 0 \).

ii) Adjacent Orientations

To explain the criteria for assigning adjacent orientations, refer to figure 5. In the figure free energy is plotted versus \( \theta \) and \( \phi \). The arrows on the left denote those minimum that are adjacent to one another. The right side of figure 5 magnifies a region of the free energy surface where, it is observed that adjacent orientations (energy wells) are those that share a common “ridgeline” (and therefore a saddlepoint). In this case, wells i and j are adjacent and share saddlepoint k. The algorithm to therefore determine which minimums are nearest neighbours is as follows:

a) Find and index all minima in the system.

b) Find and index all saddle points in the system. Saddle points are found by the
condition $E_{\phi\phi}E_{\theta\theta} - E^2_{\phi\phi} < 0$.

c) Choose starting points for the moments surrounding each minimum, follow the
direction of steepest descent, find which minima the moment “falls” into and index these.
For example, if moments placed around saddle point k fell into wells i and j, the wells i
and j are nearest neighbours, related by saddle point k (figure 6).

It is mathematically possible to have saddle points that connect more than two minima.
However, in the system and free energy presented, saddle points will only connect two
minima.

Fig. 5. The free energy surface of the 3D model. Left: the total surface, the arrows signify
adjacent wells. Right: a magnification of the area in the dashed box.

Fig. 6. A schematic representation of the technique used to find adjacent wells and their
associated saddlepoint.
iii) Energy Barriers and Transition Times

Now that the adjacent orientations have been indexed, it is required to determine the transition times between these orientations. Here, the same method is used as before, relating the transition times to the square of the difference in energy of the well to the barrier between the adjacent well. In this case, there is not just one possible path (and therefore energy barrier) between two minima. Physically, the highest probability of transition will be over the path that has the least energy barrier between them. Figure 5 shows that this is clearly the saddlepoint. Thus for transition from adjacent well i to j, with saddle point k separating them, the transition time will be defined as \( t_{ij} = \tau_s (E_{\text{saddle}, k} - E_{\text{well}, i})^2 \).

C) Rate equations in 3D

Now that the nearest neighbours and transition times have been defined, it is straightforward to write the rate equations in the compact form

\[
\frac{d\xi_i(t)}{dt} = \sum_{\text{NN}, j} \left( \frac{\xi_j(t)}{t_{ji}} - \frac{\xi_i(t)}{t_{ij}} \right),
\]

where the sum is over all the wells that are adjacent to i. Once the rate equations have been defined, a self-consistency algorithm can once again be followed to determine the fraction of moments in aligned with each orientation as a function of time and therefore the total magnetization as a function of time.

D) 3D Results

The 3D model is applied to the same system as used in section 1.E. As the results will look quite similar to those in one dimension, only one figure is shown to highlight the differences. Figure 7 plots again magnetization versus stress in the quasi-static case for various initial applied field biases. As one would expect, the results look very similar to those in figure 2. However, in this case, the lowest bias curve is fitted better with the 3D model. Differences arise because the two different free energy models of equations (1) and (7) will give different values for the energy of the wells and barriers. It was also found, as one would expect in this case, the total magnetization in the x and y directions are zero due to symmetry. The results presented are for a uniaxial compressive stress along the axis of symmetry of the crystal. Calculations for other loading conditions will be presented in a later publication.
Fig. 7. The total magnetization versus uniaxial compressive stress in the quasi static regime. Solid lines are the 3D simulations, dashed lines experimental [5,16]. The curves are shown for increasing levels of bias field at 23, 44, 66, 89, 111, 167, 223 and 446 Oe.

3. THEORY AND MODELLING OF AN AC ENERGY HARVESTER

The energy harvester is shown schematically in figure 8. If coil of wire is wrapped around the Galfenol sample described in sections 1 and 2, an application of stress to the sample will induce an electric potential $V_m$ in the wire. The coil of wire will have an associated resistance $R_c$ and inductance $L_c$. The load on the system is represented by the resistor $R_{load}$. A capacitor (C) has been added to this circuit for reactive load balancing of the inductance from the generating coil.

A) Faraday’s Law of Induction

Once the magnetization as a function of time in equation (2) is obtained, the total magnetic field produced then will be $B(t) = \mu_iH + M(t)$. Faraday’s law of induction states the $V_m = NAB = NAM_z$ for a wire of N turns tightly wrapped around a material of constant
area A (the negative sign has been dropped by convention). Therefore, using this relation and the sensing model of section 1 (or 2), one has a method to calculate the induced potential as a function of applied time-varying stress.

B) AC Energy Harvesting

Kirchoff’s loop rule for this circuit is

\[ V_m - q/C + (R_c + R_{load}) \dot{q} - L_c \ddot{q} = 0 \] (6)

Expanding the magnetization as \( M_z = \sum M_{az} e^{i\omega t} \), the current \( I = \dot{q} \) is found from equation (6):

\[ I = \sum_{\omega \neq 0} \frac{NA \omega M_{az} e^{i\omega t}}{j(R_c + R_{load}) + (L_c \omega - (C \omega)^{-1})} \] (7)

Since \( V_{load} = IR_{load} \), the average power over one cycle delivered to the load is

\[ < P > = \frac{(NA \omega)^2}{2} \sum_{\omega \neq 0} \frac{M_{az}^2 R_{load}}{(R_c + R_{load})^2 + (L_c \omega - (C \omega)^{-1})^2} \] (8)

The length of the wire wrapped around the coil of radius \( r_c \) is \( l_w = 2\pi r_c N \), so that the inductance and resistance of the coil is \( L_c = \mu N^2 A_c / l_c = \mu \pi N^2 r_c^2 / l_c \) and \( R_c = \rho l_w / A_w = 2\rho r_c N / r_w^2 \) respectively. Current flowing through the coil will induce a magnetic field that affects moment rotation in the magnetostrictive device. This is given by \( B_{ind} = \mu_s NI \) for a cylindrical coil and must be added to the bias field in a self-consistent manner similar to the eddy-current effects.

C) Results

An energy harvester based on the Galfenol sample described in section 1 is examined for 230e applied bias field and 25MPa maximum compressive stress. A copper wire is wrapped around the Galfenol in the AC harvesting configuration. The length and radius of the coil are assumed to be 1cm each. The radius and resistivity of the wire are 0.1mm and 1.7x10^{-8} \( \Omega \)m (Copper) respectively. Results will be shown for various frequencies as labelled on the figures.

In figures 9 and 10, the voltage and average power across the load for 20 and 50 windings/cm are shown. In this regime, as the winding number and frequencies increase, so does the output power. Figure 11 demonstrates the effects of the coil induced magnetic field. As the number of coils around the Galfenol increases, so does the induced field (which acts
to inhibit the magnetic moment rotations). This results in increasing hysteresis as the number of coils increase.

![Output Voltage at 50 Hz](image)

**Fig. 9.** Voltage across the resistive load versus applied stress frequency.

![Power out (mW)](image)

**Fig. 10.** AC power delivered to the resistive load versus applied stress frequency.
Fig. 11. The magnetization versus stress in the Galfenol sample, 23 Oe applied bias, with and without the energy harvesting circuitry.

5. CONCLUSIONS

The 1D and 3D sensing models show similar results for this sample for the same loading condition as would be expected. These results partially validate the 3D model. If one was only to model systems with uniaxial stresses and symmetries, the 1D model would be sufficient. The 3D model however has much more flexibility to model non-symmetric systems and different loading conditions.

It was demonstrated that the sensing model can be easily integrated into an energy harvesting model. Either the 1D or 3D sensing models can be used. The self-consistency iteration must be modified to account for the magnetic field induced by the current flow through the harvesting coil.

Future work will consist of applying different types of loads such as shear, or axial loads not along axis of crystal symmetry. In addition, the model will be expanded to take into account thermodynamic effects, which will modify the transition probabilities between orientations.

ACKNOWLEDGEMENT

The financial support of Defence Research and Development Canada-Atlantic is gratefully acknowledged.
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DEVELOPMENT OF ENERGY HARVESTING MODULES BASED ON PIEZOCERAMICS

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ABSTRACT

Energy harvesting devices capture the ambient energy surrounding a system and convert it into usable electrical energy. With increasing demand for wearable electronics and an increased interest in the implementation of arrays of wireless sensors in a number of rising sectors such as health monitoring in civil infrastructure, MEMS sensor arrays for automotive and aerospace applications, and sensor arrays for environmental control, there is a surge in research in the area of power harvesting. One of the most effective methods of implementing a power harvesting system is to use ambient vibration in conjunction with a piezoelectric device to generate electric energy based on the direct piezoelectric effect.

This paper presents an assessment of recent developments in piezoelectric based energy harvesting and presents the various modules needed for successful implementation. The paper also presents a summary of the patent activity and applications on piezoceramic energy harvesting.

Keywords: energy harvesting, energy scavenging, piezoceramic
INTRODUCTION

With recent developments in personal electronics and micro-electronics, much research has been dedicated to the development of self-powered devices that can overcome the current reliance and limitations of finite-supply batteries. Such devices hold much potential in developing versatile next-generation wireless electronics for a wide variety of applications such as portable electronics and self-monitoring structures. To this end, the core of current research has focused on utilizing ambient energy from a device’s surroundings and converting it into usable electrical energy. While methods of harvesting ambient solar, magnetic and thermal energy have all been proposed, one of the most common methods of power harvesting is to convert ambient mechanical vibrations into electricity through the use of piezoelectric materials. These materials exhibit the direct piezoelectric effect that produces an electrical charge across the material corresponding to a mechanical deformation. Although many materials can exhibit such properties, efficient energy harvesting requires the use of those with specific material properties, such as large electromechanical coupling factors ($k$) and high mechanical quality factors [1]. As a result, piezoceramics such as PZT are most often used for energy harvesting [1].

This paper highlights some of the recent developments in piezoceramic energy harvesting along with proposed circuits that can improve the performance of energy harvesters. It also presents a summary of recent patent activity on piezoelectric energy harvesting.

ENERGY HARVESTING DEVICES

Due to significant research interest in piezoceramic energy harvesting, a wide variety of harvesting devices have been proposed in recent years. To simplify categorization, these can be divided into two broad groups: resonating structures and impulse driven generators.

Resonating structures are the most common type of energy harvesting device and make use of a central structure that is excited by ambient vibrations approaching its resonant frequencies. The oscillating structure then stresses a bonded piezoceramic element generating a time-varying voltage. The most common example of such a resonating structure is a cantilevered beam with piezoelectric elements bonded to the top and bottom surfaces of the beam. For every deflection of the beam, the piezoelectric element is stressed in a transverse direction resulting in voltage generation across electrodes placed at the top and bottom surfaces (see Figure 1). Often, a proof mass is also placed near the tip of the beam to improve its displacement and control the frequency response of the beam.

Fig. 1: Schematic of one cantilever beam harvester [2]
Due to their simplistic design and predictable response, cantilevered harvesters are the subject of extensive modelling and testing [2-10]. One example of such work is by Roundy and Wright [2]. Using a 1 cm³ volume constraint for a proposed device, the group developed an analytical model of the system and subsequently validated it through two optimized prototypes. The prototypes were then tested on a shaker at a frequency of 120 Hz producing a peak total of 375 µW and were able to successfully power a custom-built radio.

Other research on straight cantilever beams includes the work by Sodano et al. in analytically modelling cantilever beams [3-8] and predicting effects such as vibration dampening in the beam due to energy harvesting [6]. Sodano et al. have also investigated the amount of power generated by a cantilevered plate and storing it in batteries or capacitors [7]. In this, it was found that at resonant frequencies, the tested plate was able to produce 2 mW of power and was capable of recharging batteries [8].

Alternatives to straight cantilever beams have also been explored. Zheng et al. have developed an “air-spaced” cantilever beam designed with a tip mass to increase the amplitude of the generated signal [9], and thus improve the efficiency of the AC-DC conversion for subsequent storage. This design uses a fixed base and a proof mass that is attached using a mechanical beam and a separate piezoelectric beam (see Figure 2). The resulting asymmetric structure can undergo two modes of bending: pure bending and S-shaped bending. For energy harvesting applications, the S-shaped bending mode must be avoided and the researchers use an analytical model as well as a FEM model to develop criteria to determine the dominant mode of bending. The model was successfully validated using a prototype that was able to produce 32.5 µW of power at a frequency of 150 Hz.

**Fig. 2:** Schematic of the air-spaced cantilever beam. The bottom beam is the mechanical beam while the top beam is piezoelectric [9].

Other alternative cantilever beam designs also include a tapered cantilever developed by Glynne-Jones et al. to produce uniform strain over the length of the beam [10]. A tested prototype using PZT piezoceramics was able to produce 3 µW at its fundamental frequency of 80 Hz with a load resistance of 333.1 kΩ. The device was made from thick film printing of the piezoceramic material and as a result, suffered from reduced piezoelectric properties when compared to bulk materials.

Despite their simplicity, one of the major drawbacks of the cantilevered designs is their poor frequency response. For optimum performance, cantilevered structures must often be excited close to resonant frequencies to maximize deflections and thus maximize generated
power. As a result, cantilevered structures must be placed in environments where the ambient frequencies match the device’s operating frequencies. While the operating frequency of an energy harvester can be tuned to a certain extent through variations of the proof mass or beam properties, the bandwidth of the device remains more or less fixed. In order to address this issue, some novel solutions have been proposed.

Marinkovich and Koser have proposed a concept for Smart-Sand, a wide bandwidth piezoelectric energy harvester that makes use of four cantilever tethers with bonded piezoceramic materials to support a central proof mass (see Figure 3(b)) [11]. The non-linear dynamics of the device result in a large amount of bending as well as stretching of the tethers and consequently, a 3D finite element model of the device was created to analyze its behaviour. This model was then validated through a tested prototype that was able to successfully operate between the frequencies of 160-400 Hz without the need for any tuning while producing a peak 1 µW of power. The authors note that the critical frequencies of the device could be decreased by creating thinner and more compliant tethers or by increasing the mass of the proof mass.

![Fig. 3: The frequency response of (a) a cantilever harvester compared to (b) a Smart-Sand harvester [11]](image)

As can be observed from the above examples, most resonating energy harvesters operate at frequencies greater than 100 Hz. For low frequency application, energy harvesting must usually be conducted through impulse driven generators. Rastegar et al. have proposed one such energy harvesting platform for very low frequency vibration operating conditions (on the order of 0.1 – 0.5 Hz) such as ships and trains [12]. This design consists of a primary travelling mass that oscillates with the platform at very low frequencies moving back and forth within the device enclosure. As it does so, it strikes a number of secondary pendulums with attached piezoelectric elements that can then oscillate at their natural frequency. The pendulum frequency can be tuned with a tip mass to optimize power output with respect to the overall platform frequency. Prototypes of this device were still under construction at the time of the paper’s publication.

A similar approach of using a moving mass to strike secondary piezoelectric elements has also been proposed by Renaud et al. in [13]. Using human motion to create mechanical vibrations, the proposed device used a sliding mass within a rigid frame to strike piezoelectric elements at either end of the frame. To improve the device’s power outputs, magnets were also attached at either end of the device to increase the force of impact.
Modelling results predicted that the device would be able to produce a peak 40 µW of power at 1 Hz and 0.1 m/s² excitation amplitudes.

One of the main limitations of impulse driven harvesters is that the majority of the energy by the impacting object on the piezoelectric harvester is returned to the object in the collisions. Umeda et al. investigated the effects of such collisions between a steel ball and a piezoceramic plate and developed an analytical model for the system in [14]. Through subsequent testing, they found that the device was only 9.4% efficient in extracting energy from the steel ball. Goldfarb and Jones [15] found similar results with a stack piezoceramic configuration and observed that the majority of the energy was returned back to the excitation source. Consequently, the stack configuration was found to be most effective at low input frequencies around 5 Hz. Xu et al. [16] compared the effects of impact loading to slowly applied compressive loading for a piezoceramic material and found slow compressive loading to produce more energy than impact loading due to the brittle nature of piezoceramics and poor energy transfer between the impacting object and the material.

<table>
<thead>
<tr>
<th>Reference</th>
<th>Device Type</th>
<th>Power [µW]</th>
<th>Frequency [Hz]</th>
<th>Overall Size [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roundy [2]</td>
<td>Resonant - Cantilever Beam</td>
<td>375</td>
<td>120</td>
<td>30 x 3.6 x 7.7</td>
</tr>
<tr>
<td>Sodano [7]</td>
<td>Resonant - Cantilever Plate</td>
<td>2000</td>
<td>50</td>
<td>80 x 40 x 1.0</td>
</tr>
<tr>
<td>Zheng [9]</td>
<td>Resonant - Cantilever Beam</td>
<td>32.5</td>
<td>150</td>
<td>42 x 22 x 0.85a</td>
</tr>
<tr>
<td>Glynne Jones [10]</td>
<td>Resonant - Cantilever Beam</td>
<td>3</td>
<td>80</td>
<td>23 x 20 x 0.1</td>
</tr>
<tr>
<td>Marinkovich [11]</td>
<td>Resonant - Tethered Mass</td>
<td>1</td>
<td>160-400</td>
<td>4 x 4 x 0.5a</td>
</tr>
<tr>
<td>Renaud [13]</td>
<td>Impulse Driven</td>
<td>40</td>
<td>1</td>
<td>12 x 10 x 5</td>
</tr>
</tbody>
</table>

* Estimated from given values

**Table 1: Summary of energy-harvesting devices**

**ENERGY HARVESTING CIRCUITRY**

The successful storage and use of energy generated by various harvesting devices requires the use of specific circuitry to rectify and optimize the output from the devices. These are characterized below.

**Energy Storage**

The power produced by piezoelectric elements from ambient energy is too low to directly power most devices. As a result, the energy must first be stored using appropriate methods and then be utilized for higher power applications. The two most common choices for energy storage are capacitors and rechargeable batteries. In comparing the two, Umeda et al. [17] found that capacitors were the more efficient choice for their tested device, but their storage capacity was too low for many applications. Similarly, Starner [18] found batteries more suitable for high power applications with capacitors more efficient for low-excitation piezoelectric elements. Capacitors were also considered the storage means of choice for many in-vivo applications due to their less intrusive nature.
AC/DC Converter

The nature of harvesting energy from mechanical vibrations means that the charge generated by the piezoelectric element is oscillatory in nature and hence must be rectified through an AC/DC converter before it can be stored. The most common implementation of this circuit component is a simple diode bridge rectifier connected to the electrodes of the piezoelectric element. However, to overcome the bias voltages of the diodes, the piezoelectric element must often undergo significant excitation magnitudes [19]. Liu et al. [19] have proposed an alternative active rectifier circuit that makes use of a MOSFET based full inverter circuit to apply an average-value voltage across the piezoelectric element. The resultant circuit is described as being 78% efficient and the tested device was able to produce 7 mW of power as opposed to 5 mW obtained from a similar diode bridge rectifier.

DC-DC Step Down Converter

One method of optimizing the power flow from energy harvesting device to the battery is through the use of a DC-DC step down converter. Since the voltages generated by a piezoelectric element can be very high, a DC-DC converter can regulate the voltage to an acceptable level for the battery or the load. To further optimize the circuit’s performance for varying voltages, the converter can be controlled through a controller circuit. Ottman et al. [20] proposed one such circuit that controls the DC-DC converter through the use of duty cycles to hold the optimal voltage at the rectifier output. At high excitation levels of the piezoelectric element, the circuit can be effectively implemented through fixed-duty PWM signals and a switching MOSFET. At low excitation levels, however, the duty cycles vary over a wide range and this requires complex controllers that are power inefficient for implementation.

Non-Linear Voltage Processing

Another method of improving the power flow from the piezoelectric element is through the use of non-linear voltage processing. In this, the rectified voltage of the piezoelectric device is subjected through a non-linear circuit component such as a switched inductor to improve power flow. While many forms of non-linear voltage processing have been developed, the most common of these are synchronous charge extraction and Synchronized Switching Harvesting on Inductor (SSHI) [21]. Lefeuvre et al. compared these methods to a baseline rectified energy harvesting device to observe their performance in [22]. Based on their experiments, synchronous charge extraction was found to be the most effective optimization technique and its performance was independent of the circuit load. However, this method peaked at low electromechanical coupling factors ($k^2 < 0.006$) and its effectiveness decreased as $k$ increased. The two SSHI methods (parallel and series) delivered lower peak power outputs when compared to the synchronous charge extraction but still offered improvements over the standard interface circuit.
ENERGY HARVESTING PATENT ACTIVITY

An examination of international patent databases shows that the number of patent families published each year on piezoelectric-based energy harvesting has increased steadily since 2003 (Figure 4). Aerospace is a prominent application area cited in the patent literature, and automotive and medical applications are also significant. The Boeing Company had the largest number of patent families of any single corporate entity for the search strategy used (the search strategy is shown in the caption of Figure 4). A discussion of some of the main patents and the direction of the technology is presented below.

![Graph showing the number of patent families that contain piezo*, as well as (energy or power) in close proximity to (harvest* or scaveng*), in the title, abstract or claims (* is a wild card). The point for 2009 is based on an extrapolation of data for January 1 to July 10, 2009. A patent family includes all patents or applications for a given invention for multiple countries and issuing authorities, including the US, Europe and Japan. Data was obtained using PatBase® (Registered trademark of RWS Information Ltd. and Minesoft Ltd - www.Patbase.com).](image)

Harvesting of vibrational energy in aircraft or space structures is of particular interest for powering devices such as wireless sensors used in integrity and performance monitoring. Wireless designs reduce weight and complexity by eliminating cabling. Furthermore, energy scavenging at a remote site solves accessibility problems when the device location makes battery replacement difficult. Recent patents by Boeing focus on mechanical designs that provide broadband response [23-26], small vibration amplitude response [23, 24], and those which have reduced size and weight [24]. Patent [24] cites all three of these features and uses initially parallel piezoelectric and biasing beams that are slightly bowed by mechanical stress. Notably, the designs described in [23, 24] both respond to low frequencies (a few Hz or less), even though they are not impulse driven. Boeing also has intellectual property on energy storage circuitry that improves energy conversion efficiency using diode-based passive switching [27]. However the practicality of inductance values used in the circuit may restrict its use to frequencies in the kHz range.

A notable application for piezoelectric energy harvesting is to power sensors with wireless communications in a vehicle’s tires. Patents and applications in this area are held by Michelin [28] and Bridgestone [29]. Parameters such as temperature, pressure, and number of rotations could be monitored and used in warning systems or possibly used for feedback.
control of tire pressure. Transmission of manufacturing information is also contemplated. In the medical field, piezoelectric energy harvesting may be used to power various implanted devices. Recent examples found in the patent literature include a stent that can detect blockage and can also carry out other functions such as vibrating to prevent blockage or deliver drugs [30]. Another example uses piezo-harvested power to drive pH sensors to detect and communicate the presence of organ ischemia [31].

CONCLUSIONS

Piezoceramic energy harvesting holds much promise in the development of new versatile electronics. The ability to capture ambient energy from a device’s surroundings and convert it into usable electrical energy is an attractive prospect in many fields and hence has been the subject of much academic and industrial interest. However, despite significant research, piezoceramic energy harvesting remains an emerging technology that requires much advancement before it can be commercially viable. The power generated by current piezoelectric harvesters remains too low for many applications even with the development of optimizing circuitry. Also, the frequency response of most prototypes is very limited for many practical applications. As a result, further research must be conducted to improve the power generated by piezoceramic elements. This can be achieved through developing new harvester designs that can maximize the amount of stressed material and consequently improve the generated power. Novel structural designs can also help improve the frequency response of the device. Finally, alternative piezoceramic materials and their characteristics must be investigated to find those most appropriate for the specified applications. The development of such piezoceramic energy harvesting platforms will help achieve the many potential benefits of this promising technology.

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POWER FOR WIRELESS SENSORS

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ABSTRACT

The recent economic downturn, the high cost of energy and its environmental impact coupled with continued drive to operate efficient military and commercial infrastructure have intensified the development of several technologies including flexible electronics, wireless sensors, wireless sensor networks, and advanced power generation. Additionally, with the green energy revolution, and among several green energy initiatives, energy harvesting technology development has taken front stage particularly in the area of environmental energy harvesting. Energy harvesting is the process by which energy is captured and stored. A variety of different methods exist for harvesting energy, such as solar power, piezoelectricity, thermoelectricity, and physical motion. If pervasive networks of wireless sensors and communication devices are to achieve their full potential, practical solutions for self-powering these autonomous electronic devices must be sought. This document reports on an experimental parametric analysis of piezoelectric vibration-based energy scavenging technique to potentially power wireless sensors and sensor networks. The potential of this energy harvesting approach is demonstrated and experimental results are presented.

Keywords: Wireless sensors, Sensor networks, Energy harvesting, Energy scavenging, Structural health monitoring.
INTRODUCTION

In recent years, due to weight, cost, redundancy and remote access benefits, Wireless Sensors (WS) and Wireless Sensor Networks (WSN) have gained significant interest from both military and commercial sectors. A wireless sensor, also known as a mote, smart dust, smart sensor or sensor node, consists of a sensor and instrument packages that are microprocessor driven and include advanced features such as communication and limited data storage and processing. A wireless sensor network usually consists of a large number of small scale nodes and has limited data processing and storage capacity, wireless data communication and transmission ability, and advanced sensing capabilities. The WSN is also known as a network of Radio Frequency (RF) transceivers, sensors, machine controllers, microcontrollers and user interface devices with at least two nodes communicating by means of wireless transmission. Figures 1 and 2 present a simplified block diagram of a wireless sensor node and a conceptual configuration of wireless sensor network for aircraft Structural Health Monitoring (SHM). Applications that could benefit from such wireless technology include aerospace and civil structural health monitoring, environmental and industrial systems control, components and assets tracking and monitoring. It has been reported recently [1] that the wireless sensors and transmitters market is fast growing, with 200% growth worldwide in the last 5 years and 45% projected growth in the next 3 years, seeing a market forecast reaching 1,800 million US dollars by 2012 with major market growth in North America (up to 600 million dollars), Europe (500 million dollars) and Asia Pacific (400 million dollars).

With the continuous challenges facing the Canadian Forces (CF) in providing increased operational availability and reduced maintenance cost of its aircraft fleet, integration of WS and WSN is expected to reduce the on-going fiscal and operational pressures and provides the CF with on-demand decision making capabilities for its fleet’s life cycle management. In fact it is expected that wireless technology will contribute significantly to addressing the Canadian auditor general recommendations on improving operational performance and asset tracking. While providing added flexibility for data, information, knowledge accumulation and decision making, power requirement and generation continue to be in the mind of the end-user as they consider integration of such wireless technology.

A variety of different methods exist for power generation [2] exploiting the environment while protecting it. This work assesses the feasibility and suitability of one approach based on the use of piezoelectric material and ambient vibration for the generation (harvesting or scavenging) of energy to potentially power wireless sensors and sensor networks requiring power ranging from few μW to few mW (e.g. Ultra-low power consumption of 60µW to 300µW depending on the channel number). The potential of this energy harvesting approach is investigated and experimentally analysed.

POWER SOURCES

The conventional approach to power sources is the use of electrochemical batteries. These batteries can not only increase the size and weight of autonomous wireless sensors, but also suffer from the limitations of a brief service life and the need for costly replacement. Additionally, it has been experienced that at extreme temperatures (high), the use of these
storage devices is unsafe. The conventional approach is not highly practical or cost effective for many applications including those faced by the CF. Table 1[3] illustrates the characteristics of different commercial battery types; whereas; Figure 3 [4] shows the life cycle of rechargeable and non-rechargeable batteries.

![Fig. 1: Simplified block diagram of a wireless sensor node](image1)

![Fig. 2: A conceptual configuration of wireless sensor network for SHM](image2)

![Table 1: Characteristics of different commercial battery type](image3)

<table>
<thead>
<tr>
<th>Battery Type</th>
<th>Volumetric Energy Density (Wh/dm³)</th>
<th>Gravitational Energy Density (Wh/kg)</th>
<th>Self-Discharge % per year</th>
<th>Cycle Life No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alkaline</td>
<td>300</td>
<td>125</td>
<td>4</td>
<td>1</td>
</tr>
<tr>
<td>Ni-Cd</td>
<td>100</td>
<td>30-35</td>
<td>15-20</td>
<td>300</td>
</tr>
<tr>
<td>Ni-MH</td>
<td>175</td>
<td>50</td>
<td>20</td>
<td>300</td>
</tr>
<tr>
<td>Li-ion</td>
<td>200</td>
<td>90</td>
<td>5-10</td>
<td>500</td>
</tr>
</tbody>
</table>

![Fig. 3: Life cycle of one cubic centimetres rechargeable and non-rechargeable batteries](image4)
Even though recent efforts have focused on the development of super and ultra capacitors, fuel cells, and other fixed energy alternatives with impressive large capacitance and higher number of cycles, these remain impractical for wireless devices with an expected lifetime of more than 10 years. For wireless sensors and communication nodes to achieve their full potential, practical options of power generation for these autonomous electronic devices must be investigated. A widely investigated attractive alternative is to use devices that generate power by scavenging ambient environment energy. Energy harvesting or scavenging is the process by which energy is captured and stored. Frequently this term is applied when speaking about small autonomous devices, like those used in wireless sensor networks. A variety of different methods and approaches exist for harvesting energy, such as solar power, ocean tides, piezoelectricity, thermoelectricity, and physical motion. Figure 4 [4] and Table 2 [5] show the output from a variety of energy harvesting schemes.

![Energy harvesting schemes](image)

**Fig. 4:** Power output using different energy harvesting schemes

<table>
<thead>
<tr>
<th>Energy Source</th>
<th>Challenge</th>
<th>Estimated Power (in $1 \text{ cm}^3$ or $1 \text{ cm}^2$)</th>
</tr>
</thead>
</table>
| Light         | Conform to small surface area | $10\mu W$-$15mW$  
Outdoors: $0.15mW$-$15mW$; Indoors: $<10\mu W$ |
| Vibrations    | Variability of vibration | $1\mu W$-$200\mu W$  
Piezoelectric: $\sim 200\mu W$; Electrostatic: $50\mu W$-$100\mu W$; Electromagnetic: $<1\mu W$ |
| Thermal       | Small thermal gradients | $15\mu W$; $10^\circ C$ gradient |

**Table 2:** Estimated power and challenges for different energy sources

With the green energy revolution, energy harvesting technology development has intensifies, in recent years [6]. The global market for energy harvesting devices for small electronic and electrical equipment is expected to reach 10,000 million devices by 2019 from the current (2009) half a million [7]. Figures 5 [7], illustrates the efficiency-power density and cost-life options for currently popular forms of energy harvesting approaches. Vibration-based energy harvesting techniques has emerged as one of the promising approaches to address the needs for low power wireless sensors and sensor networks. Figure 6 illustrates the wide range of power output ($10 \mu W$ to 1 kW) for vibration and heat based devices at

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different frequencies. Piezoelectric vibration-based energy converters are shown to deliver the highest efficiency at lowest cost and increased life cycle.

![Fig. 5: Cost-life options for currently popular forms of energy harvesting approaches](image)

**Fig. 5:** Cost-life options for currently popular forms of energy harvesting approaches

**Fig. 6:** Power output for vibration-based devices

**ENERGY HARVESTING SYSTEM**

The Energy Harvesting (EH) system evaluated in this study is an advanced ceramicitec (ACI) energy harvesting system. The portable system consists of four main components as shown in Figure 7. The Piezoelectric Fiber Composite (PFC) beam, with wired electrodes and snap connector terminals is used to generate/harvest energy when exposed to vibration. The mechanical amplification device is employed to increase the output power of the PFCB composite through increased beam deflection. The energy harvesting circuit card (DB-2) with regulated (EH-R) or unregulated (EH-UR) power output is used to store the harvested energy. The ruggedized enclosure is used to house the EH system components.
This piezoelectric energy harvester employs a Piezo-electric Fiber Composite (PFC) cantilevered beam that is developed from a technology called Viscose Suspension Spinning Process (VSSP) [8]. Conventional piezoelectric ceramic materials are rigid, heavy, and produced in block form. This low-cost technology process can produce fibers ranging in diameter from 10 microns to 250 microns. When formed into user defined shape, the ceramic fibers possess all the desirable properties of ceramics (electrical, thermal, chemical) but eliminate its detrimental characteristics (brittleness, weight). It has been reported [8] that the VSSP generates fibers with 20-30% more efficient energy conversion than traditional bulk ceramics. To put this into perspective, mechanical to electrical transduction efficiency can reach 70% compared with the 16-18% common to solar energy harvesting. These fine ceramic fiber composites provide increased specific strength over monolithic materials, as a result of the fiber load sharing mechanism. For improved toughness, durable polyethylene sheets are used when potted or laminating. The orthotropic nature of the unidirectional fibers and interdigital electrode permits effective design of modal actuators and sensors.

The system presented in Figure 7 is not optimized for any specific self-powered application, but is suitable for applications where mechanical vibration exist (e.g. automobiles, aircraft, trucks, etc.). This system is used to conduct a systematic analysis of the feasibility and suitability of piezoelectric vibration-based energy harvesting approach for potential use with wireless sensors and other low power electronics within a military environment.

Fig. 7a: Portable energy harvesting system

Fig. 7b: PCFC piezo beam and power conditioning and energy storage unit (DB-2)
SYSTEM ANALYSIS

To enhance the understanding of the influencing parameters on the performance of vibration-based energy harvesting using piezoelectric materials, to contribute to future selection of optimum parameters for efficient EH system performance, and to assess the feasibility and the suitability of this technology for potential use with wireless low power electronics, an experimental parametric assessment is conducted. Using an electromechanical shaker, an accelerometer, a LabVIEW-based data acquisition (DAQ) system and the ACI energy harvesting system, the impact of the variation of excitation frequency, amplitude, material thickness and system load on the system’s power output is assessed.

Experimental results show that at a fixed excitation amplitude of 0.5 V a maximum power output was obtained at an optimal excitation frequency equals to that of the PFC beam’s resonance frequency. According to the characteristics of the DB-2 power conditioning unit, when the capacitors are fully charged, the energy stored should be within 2.67 mJ and 6.27 mJ. The energy obtained during the experiment after time \( t = 167 \) seconds, for the same beam, is 2.9 mJ (Table 3). As shown in Figure 8 it has been reported [8] that using random base excitation, a 1.44 mJ can be stored in the capacitor after 33 seconds for material type 3 and 57 seconds for material type 1. Figure 8b illustrates the higher time required to store the same energy using the same materials in a different environment.

Additionally, at the PFC beam’s resonance frequency, a higher system power output was obtained with increased excitation amplitudes (Table 4), with linear and exponential increase for beam configurations with and without proof mass, respectively. Reduced capacitor charging time was also obtained for higher excitation amplitudes. Output saturation was experienced at excitation amplitude of 1.8 V (10 mJ) due to the 10 V DAQ output limit. The energy increase with excitation amplitude provides flexibility in the design of the system for known energy need (e.g. specific energy needs results in specific excitation amplitude).

| Table 2: EH system output for excitation frequency variation at time 167 seconds |
|---------------------------------|----------------|----------------|
|                        | Beam with proof mass | Beam without proof mass |
| Resonant frequency (Hz)      | 55              | 40              |
| Maximum energy stored (mJ)  | 2.9             | 0.13            |
| Maximum power transferred (µW) | 160            | 28              |

At excitation frequency of 40 Hz and amplitude of 0.5 V, it was shown that with increased PFC beam thickness, the overall beam capacitance increased linearly, the material resistance decreased from \( G\Omega \) to \( k\Omega \), and the energy stored in the capacitor increased exponentially with highest value obtained with 5-layers PFC beam (e.g. 1 mJ at 8 Seconds). The PFC beams are stacked mechanically in series and electrically in parallel, resulting in Piezoelectric Multilayer Composites (PMC). Figure 9 illustrates the latter at time 10 seconds. Additional experiments, demonstrated that to obtain optimum power, a load with optimum resistance must be coupled to the capacitor. This resistance is expressed as \( R_{OPT} \),

\[
R_{OPT} = \frac{\pi}{2C_0\omega}, \quad \text{where} \quad \omega \quad \text{and} \quad C_0 \quad \text{are the resonant frequency and the capacitance of the piezoelectric material, respectively.} \]

For 2-PMC beam, an excitation frequency of 55 Hz,
excitation amplitude of 1 V, a capacitor capacitance of 7.62 nF and an optimal resistance of 600 kΩ are obtained with maximum power output of 75 μW. It is noted that the power output without the 600 kΩ load is 1.25 mW.

Fig. 8: Energy stored within capacitors in two environments under random vibrations

Table 4: EH system output for excitation amplitude variation at time 167 seconds

<table>
<thead>
<tr>
<th>Beam with proof mass (W PM)</th>
<th>Beam without proof mass (w/o PM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resonant frequency (Hz)</td>
<td>55</td>
</tr>
<tr>
<td>Maximum energy stored (mJ)</td>
<td>10*</td>
</tr>
<tr>
<td>Maximum power transferred (μW)</td>
<td>628*</td>
</tr>
<tr>
<td>Optimal excitation amplitude</td>
<td>None since energy and power linearly increase with an increase of amplitude</td>
</tr>
</tbody>
</table>

*: DAQ range limitation  
+: After time t = 50 seconds at resonant frequency f = 55 Hz and amplitude of 1V  
~: After time t = 6 minutes at resonant frequency f = 40 Hz and amplitude of 1V

Additional experiments were conducted, using controlled random vibrations, and it was observed that (at time 110 seconds) a 115 μW is obtained at the capacitor in the unloaded configuration (No resistive load), whereas a 4.5 μW (at time 135 Seconds) is obtained when the capacitor is exposed to the optimal resistance of 600 kΩ.

CONCLUSIONS

The feasibility and the suitability of a vibration-based energy scavenging approach using piezoelectric material that can scavenge power from low-level ambient vibration sources were assessed. The analysis considered the impact of the variation of excitation frequency, amplitude, material thickness and system load on the system’s power output. Given appropriate data acquisition system, efficient power storage and conditioning, suitable piezoelectric material, beam configuration, and innovative power amplification the resulting
vibration-based power source is sufficient to support networks of ultra-low-power and peer-peer wireless sensor nodes.

![Graph showing energy stored for different beam thickness](image)

**Fig. 9:** Energy stored for different beam thickness

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ELECTROMECHANICAL PERFORMANCES OF DIFFERENT SHAPES OF PIEZOELECTRIC ENERGY HARVESTERS

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ABSTRACT

In recent years, researchers have shown an interest in the possibility to harvest mechanical energy from vibrating structures. A common way to proceed consists of using the direct piezoelectric effect of a bimorph cantilever beam with integrated piezoceramic elements. Several studies focused on the development of analytical models describing the electromechanical coupling. However, these models were limited to simple structures such as constant cross-section cantilever beam harvester.

This paper studies the effect of the harvester geometry on its electromechanical performance. A specific geometry will be of interest in this paper: a tapered beam. A semi-analytical model is developed using Rayleigh-Ritz approximations and a trigonometric functions set. Numerical simulations are then performed for three different cases: a standard rectangular harvester, an equivalent mass/stiffness tapered beam harvester and an equivalent maximal strain tapered beam harvester. It will be shown that tapered beams lead to a more uniform strain distribution across the piezoelectric material and could increase the harvesting performance by 69%. Tapered beam harvesters are very interesting since they are low-cost and easy to manufacture.

Keywords: Piezoelectric energy harvesting, strain distribution, harvester shape, electromechanical modeling
INTRODUCTION

Energy harvesting from vibrating mechanical structures has been studied by several researchers in the last decade [1][2]. The energy harvesting performances was predicted for simple geometries such as constant thickness beams and plates. Williams and Yates [3] established a single degree-of-freedom mechanical model having a damper as a dissipative element to represent the energy harvester. Then, Roundy et al. [4] analytically represented piezoelectric bimorph harvester by an electrical circuit were the mechanical elements where represented by their electrical equivalent and solved using Kirchoff law. Later, Sodano et al. [5] developed the constitutive equations for a piezoelectric bimorph harvester using a variational approach. With their model, the electromechanical performance of the harvester (which is a continuous system) is described by a set of two coupled matrix equations.

All these work are based on a constant cross-section cantilever beam. Baker et al. [6] proposed to vary the width of the beam (trapezoidal shape) in order to increase the efficiency. By doing so, they reported a 30% increase in energy harvesting. There is no reason for which the geometry should be limited to rectangular and trapezoidal configurations. In this paper, the thickness of the beam is varied (triangular shape) in order to obtain a more uniform strain distribution along the beam, hence to maximize the harvested energy. In the following, a semi-analytic model is briefly described. The model takes both, the mechanical and electromechanical effects into account. A short numerical study is then presented for three different harvester shapes.

MECHANICAL MODELING PROCEDURE

Fig. 1 shows the harvester under study. It consists of a cantilever tapered beam, two bonded piezoelectric ceramics and two masses at its end. \( t_0 \) and \( t_f \) are the thicknesses of the beam respectively at the fixed-end and at the free-end. \( t_p \) is the thickness of each piezoceramic plate and \( t_m \) is the thickness of the masses. \( L \) is the length of the beam, \( L_1 \) is the tapered length and \( L_2 \) is the length along the beam for which the thickness is constant. Finally, \( \theta \) is the taper angle.

![Fig. 1: Geometrical model of the tapered beam energy harvester.](image)
Mechanical dynamic model

Flexural vibrations of the beam are modeled using Bernoulli’s assumptions. The displacement field \( \{u_x, u_y, u_z\} \) can thus be written as:

\[
\begin{align*}
    u_x &= -y \frac{\partial w(x,t)}{\partial x} \\
    u_y &= w(x,t) \\
    u_z &= 0
\end{align*}
\]  

(1)

The vector \( \{u_x, u_y, u_z\} \) represents the displacement of a point either on the beam, the piezoceramic elements or the tip masses. Perfect bonding is considered between the beam and the piezoelectric elements leading to a continuity of displacements at these interfaces. Rayleigh-Ritz approximations are used to solve the system with the general following series expansion:

\[
w(x,t) = \sum_{i=1}^{N} F_i(x)G_i(t)
\]

(2)

where \( F_i(x) \) are the spatial functions and \( G_i(t) \) are the time functions used to describe the displacement.

The analytical formulation is based on the variational approach, in which the energy of the whole system is minimized using the Lagrangian. Using the coefficient \( G_i(t) \) as the generalized coordinates, Lagrange equations can be written in the general form:

\[
\frac{d}{dt} \left( \frac{\partial T}{\partial \dot{G}_i} \right) - \frac{\partial T}{\partial G_i} + \frac{\partial U}{\partial G_i} = 0 \quad (i = 1, 2, ..., N)
\]

(3)

where \( T \) represents the total kinetic energy of the system and \( U \) the total potential energy of the system.

Kinetic energy of each element (beam, piezoceramics and tip masses) is described by:

\[
T_i = \frac{1}{2} \int_0^L \rho_i A_i(x) \left( \frac{\partial w(x,t)}{\partial t} \right)^2 \, dx + \frac{1}{2} \int_0^L \rho_i I_i(x) \left( \frac{\partial^2 w(x,t)}{\partial x \partial t} \right)^2 \, dx
\]

(4)

where \( \rho_i \) represent the density, \( A_i(x) \) and \( I_i(x) \) are respectively the cross-section area and the area moment of inertia of each element.

The first term of (4) represents the kinetic translational energy of each component while the last term represents the kinetic rotational energy of each component. Resolution by Rayleigh-Ritz method allows the introduction of the kinetic rotational energy functions which increases the accuracy of the model. For a tapered beam, rotational energy is greater at the fixed-end than at the free-end of the beam. Thus, rotational energy does have a significant impact on the strain distribution and must be taken into account. These rotational energy functions assume that piezoceramic cross-sections remain perpendicular to the neutral axis. Since the slope angle \( \theta \) will remain small, this assumption does not influence significantly the accuracy of the model.
For the whole system, the total kinetic energy is:

\[ T = T_b + T_p + T_m \] (5)

where the subscripts \( b \), \( p \) and \( m \) indicate respectively the beam, the piezoceramics and the tip masses. Potential energy of each element can be defined as:

\[ U_i = \frac{1}{2} \int_0^L \int_A \sigma_{xx} \varepsilon_{xx} dA dx \] (6)

with

\[ \varepsilon_{xx} = -y \frac{\partial^2 w(x,t)}{\partial x^2} \quad \text{and} \quad \sigma_{xx} = E_i \varepsilon_{xx} \] (7)

where \( E_i \) is the modulus of elasticity of each element. Using (7) and (6) leads to a simplified form of the potential energy of each element:

\[ U_i = \frac{1}{2} \int_0^L E_i I_i(x) \left( \frac{\partial^2 w(x,t)}{\partial x^2} \right)^2 dx \] (8)

For the whole system, the total potential energy is therefore given by:

\[ U = U_b + U_p + U_m \] (9)

Inserting (5) and (9) into Lagrange equations (3) gives the following differential equations system:

\[ M \ddot{\mathbf{G}} + KG = 0 \] (10)

where \( M \) and \( K \) are respectively the mass and the stiffness matrix. \( M \) and \( K \) depend on the spatial functions \( F_i(x) \) which are defined in the next section.

By solving the corresponding eigenvalue problem, it is possible to find the natural frequencies \( \omega_1, \omega_2, ..., \omega_N \) and the mode shapes \( [p_1, p_2, ..., p_N] \) of the system. The first mode shape \( p_1 \) will be later used to simplify the model to one degree-of-freedom.

**Trigonometric functions set**

The spatial functions set used in this paper is trigonometric and defined as follow:

\[ F_i(\xi) = 1 - \cos \left( \frac{(2i - 1)}{2} \pi \xi \right) \] (11)

where the adimensional axial position \( \xi = x/L \) is used.

Fig. 2 shows the first four functions of this set. This functions set has many advantages including: its simplicity decreases the required computing time, it is bounded between 0 and 2 which avoid truncation errors, it respects both geometrical \( F_i'(\xi=0) = F_i'(\xi=0) = 0 \) and one natural \( F_i''(\xi=1) = 0 \) boundary conditions. Moreover, the shape of the first function \( F_i(x) \) is very close to the first mode of a cantilever beam which helps in reducing the number of functions required to well approximate the first mode.
ENERGY HARVESTING THEORY

The basic principle of vibration energy harvesting is to tune the first natural frequency of the harvester to the frequency of the vibration source. As described by Liao et al. [7], the dynamic model presented in the previous section can be reduced to one degree-of-freedom by considering the first mode only. The direct piezoelectric effect must be taken into account as illustrated in Fig. 3.

Fig. 3: Single degree of freedom electromechanical model.

The spatial function \( \varphi(x) \) corresponding to the first mode can be determined with the spatial functions set \( F_i(x) \) and the first mode shape \( p_1 \), i.e.:

\[
\varphi(x) = [F_1(x) \quad F_2(x) \quad \cdots \quad F_N(x)] \times p_1
\]

The transverse displacement can then be written as:

\[
w(x,t) = \varphi(x)h(t)
\]

where \( h(t) \) is the time response of the first mode.

By considering the direct piezoelectric effect, one can show that the electromechanical behaviour of the harvester can be modeled with the two following equations [7]:

\[
M_{eq}\ddot{h}(t) + C_{eq}\dot{h}(t) + K_{eq}h(t) - \Omega v(t) = Da(t)
\]

(13)

\[
\Omega h(t) + C_p v(t) = q(t)
\]

(14)

with

\[
M_{eq} = M_b + M_p + M_m
\]

\[
= \int_0^L \rho_b A_b(x)\varphi(x)\varphi(x)dx + \int_0^L \rho_b I_b(x)\varphi'(x)\varphi'(x)dx + \int_0^{L_i} \rho_p A_p(x)\varphi(x)\varphi(x)dx + \int_0^{L_i} \rho_p I_p(x)\varphi'(x)\varphi'(x)dx + \int_0^{L_m} \rho_m A_m(x)\varphi(x)\varphi(x)dx + \int_0^{L_m} \rho_m I_m(x)\varphi'(x)\varphi'(x)dx
\]
\[ K_{eq} = K_b + K_p + K_m \]
\[ = \int_0^L E_b I_b(x)\varphi''(x)\varphi''(x)dx + \int_0^L E_p I_p(x)\varphi''(x)\varphi''(x)dx + \int_0^L E_m I_m(x)\varphi''(x)\varphi''(x)dx \]
\[ C_{eq} = \zeta 2\sqrt{K_{eq} M_{eq}} \]
\[ D = \int_0^L \rho_b A_b(x)\varphi(x)dx + \int_0^L \rho_p A_p(x)\varphi(x)dx + \int_0^L \rho_m A_m(x)\varphi(x)dx \]
\[ \Omega = 2d_{31} E_p b \int_0^{L_1} \varphi''(x)\left(\frac{t_f - t_o}{2L_1} - \frac{x + t_o + t_p}{2}\right)dx \]
\[ C_p = \frac{2K_{eq}^S \varepsilon_0}{t_p^2} \int_0^{L_1} A_p(x)dx \]

In these expressions, \(a(t)\), \(v(t)\) and \(q(t)\) are respectively the base acceleration, the output voltage and the electrical charge. \(M_{eq}, C_{eq}\) and \(K_{eq}\) are the equivalent mass, damping and stiffness of the system while \(D\) is the inertial loading applied on the system. \(\Omega\) and \(C_p\) are respectively the electromechanical coupling coefficient and the piezoceramics equivalent capacitance. Finally, \(\zeta\) is the mechanical damping ratio, \(K_{eq}^S \varepsilon_0\) is the permittivity of the piezoceramics and \(d_{31}\) is the piezoelectric charge constant.

The harvested energy is dissipated through a resistive load \(R\). Using Ohm’s law, one obtains the following additional equations:
\[ v(t) = -Rq(t) \] (16)

By combining (13), (14) and (16), it is possible to obtain a third order linear differential equation and then to determine the power dissipated across the resistive load.

**PERFORMANCE STUDY OF DIFFERENT SHAPES OF HARVESTER**

In this section, numerical simulations are presented to demonstrate the increase in performances when using tapered beam harvester instead of constant cross section harvester. The general harvester under study consists of a tapered brass beam, two steel masses and two piezoceramics (PIC 151 manufactured by PhysikInstruments) as shown previously in Fig. 1. Material properties of each element are summarized in table 1. For all case under study, the width \(b\) and length \(L\) of the beam are respectively 25 mm and 90 mm, while the piezoceramics dimensions are (70x25x0.5) mm³. A typical value of 2% is used for the mechanical damping ratio. The input excitation is a base acceleration \(a(t)\) having an amplitude of 9.81 m/s². Table 2 summarizes the three cases under study: a standard rectangular harvester, an equivalent mass/stiffness tapered beam harvester and an equivalent maximal strain tapered beam harvester. They all share a common property: the first short circuit natural frequency is 100Hz which is an arbitrary design frequency.

In order to evaluate model quality and the results accuracy, the harvester structure was modeled using finite element with NxNastran 5. The system being symmetrical with respect to the \(x\) axis, a 2D grid composed of CQUAD8 was used to perform a modal analysis. Finite
element results agreed very well with the prediction of our model. Due to length restriction, those results are unfortunately not presented in this paper.

**Table 1: Materials properties.**

<table>
<thead>
<tr>
<th>Properties</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brass modulus of elasticity</td>
<td>$E_b$</td>
<td>100 GPa</td>
</tr>
<tr>
<td>PIC 151 elasticity</td>
<td>$E_p$</td>
<td>66.7 GPa</td>
</tr>
<tr>
<td>Steel elasticity</td>
<td>$E_m$</td>
<td>200 GPa</td>
</tr>
<tr>
<td>Brass density</td>
<td>$\rho_b$</td>
<td>8740 kg.m$^{-3}$</td>
</tr>
<tr>
<td>PIC 151 density</td>
<td>$\rho_p$</td>
<td>7800 kg.m$^{-3}$</td>
</tr>
<tr>
<td>Steel density</td>
<td>$\rho_m$</td>
<td>7870 kg.m$^{-3}$</td>
</tr>
<tr>
<td>Relative permittivity of ceramics</td>
<td>$K_S^3$</td>
<td>2068</td>
</tr>
<tr>
<td>Piezoelectric charge constant</td>
<td>$d_{31}$</td>
<td>-210 pC.N$^{-1}$</td>
</tr>
</tbody>
</table>

**Table 2: Energy harvesters geometrical parameters.**

<table>
<thead>
<tr>
<th>Geometrical parameter</th>
<th>Symbol</th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beam thickness at x=0</td>
<td>$t_o$</td>
<td>0.90 mm</td>
<td>1.10 mm</td>
<td>1.63 mm</td>
</tr>
<tr>
<td>Beam thickness at x=L</td>
<td>$t_f$</td>
<td>0.90 mm</td>
<td>0.40 mm</td>
<td>0.93 mm</td>
</tr>
<tr>
<td>Tip mass thickness</td>
<td>$t_m$</td>
<td>0.85 mm</td>
<td>1.21 mm</td>
<td>9.25 mm</td>
</tr>
<tr>
<td>Slope angle of the thickness</td>
<td>$\theta$</td>
<td>0°</td>
<td>0.3°</td>
<td>0.3°</td>
</tr>
</tbody>
</table>

Case I: Constant thickness beam

The first case under study consists of a constant thickness beam energy harvester ($\theta=0^\circ$). This case represents the one generally proposed in the literature. Fig. 4a) shows the dissipated power as a function of the excitation frequency and the electrical resistance. A maximal dissipated power of 11.5 mW is observed for a 100.74 Hz excitation and a 6.08 kΩ resistive load. Fig. 4b) shows the strain distribution across the piezocermics for these values. A maximal strain of $1.37 \times 10^{-4}$ mm/mm is observed at the fixed-end of the beam (x=0) and the strain decreases in a quasi linear manner from the fixed-end to the free-end.

![Fig. 4: Case I: a) Harvested power as a function of the frequency and the resistive load and b) piezoceramic strain distribution for optimal resistance and frequency.](image-url)
Case II: Tapered beam with $\theta = 0.3^\circ$ (same $M_{eq}$ and $K_{eq}$)

The second case under study is a tapered beam energy harvester having a taper angle of $0.3^\circ$. Dimensions of the beam were chosen such that the equivalent mass $M_{eq}$, the equivalent stiffness $K_{eq}$ and the first natural frequency $\omega_1$ are identical to Case I. Fig. 5a) shows the dissipated power as a function of both the excitation frequency and the resistive load. A maximal dissipated power of 10.2 mW is achieved for a 100.68 Hz excitation and a 4.96 kΩ resistive load. Amazingly, the dissipated power is less than case I. Fig. 5b) compares strain distribution for case I (dashed line) and case II (continuous line), both for optimal resistive load and optimal excitation frequency. It can be observed that the maximal strain is 23% less for case II ($1.05 \times 10^{-4}$ mm/mm instead of $1.37 \times 10^{-4}$ mm/mm) and that the strain distribution is more uniform.

![Fig. 5: Case II: a) Harvested power as a function of the frequency and the resistive load and b) piezoceramic strain distribution for optimal resistance and frequency.](image)

Case III: Tapered beam with $\theta = 0.3^\circ$ (same maximal strain)

In order to rigorously compare the performance of two harvesters, they must have the same maximal deformation. Therefore, we modified case II by increasing both the equivalent mass and stiffness in order to keep the same natural frequency but increase the maximal strain to the one of case I. As shown in Fig. 6a), a maximal dissipated power of 19.4 mW (69% energy harvesting increases) can be obtain for a 101.06 Hz excitation frequency and a 6.12 kΩ resistive load. Fig. 6b) compares the strains distributions for case I (dashed line) and case III (continuous line), both for optimal resistive load and optimal excitation frequency. For case III, the strain distribution is more uniform which increases the performance of the harvester. From these results, one can observe that the strain distribution can be optimized by modifying the geometry of the harvester.

![Graph showing dissipated power and strain distribution](image)
Fig. 6: Case III: a) Harvested power as a function of the frequency and the resistive load and b) piezoceramic strain distribution for optimal resistance and frequency.

CONCLUSION

The main objective of this paper was to demonstrate that the efficiency of a cantilever beam vibration energy harvester can be increased by using a variable thickness. A semi-analytical model that takes both the dynamic and the electromechanical behaviour of the harvester into account was first described. Using this model, it has been shown that the energy harvested can be increased by 69% when the thickness of the beam is varied (0.3° slope angle). It also leads to a more uniform strain distribution and thus increases the amount of energy that can be harvested. Moreover, the proposed geometry is very easy to manufacture and standard commercially available rectangular piezoceramic elements can be used. Further work will investigate the optimal slope angle, i.e. the one leading to the most uniform strain distribution and will propose design guidelines for cantilever beam vibration energy harvester.

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REFERENCES

HIGH TEMPERATURE INTEGRATED ULTRASONIC TRANSDUCERS FOR ENGINE CONDITION MONITORING

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ABSTRACT

Employing a portable on-site fabrication kit, high temperature integrated ultrasonic transducers (IUTs) made of bismuth titanate piezoelectric film, of thickness greater than 50 µm, have been coated directly onto a modified CF700 turbojet engine outer casing, oil sump and supply lines and gaskets using sol-gel spray technology. Transducers top electrodes, electrical wires, conducting adhesive bond, connectors and cables have all been successfully tested for temperatures up to 500°C. The excellent ultrasonic performance of these IUTs is demonstrated in this paper and the potential applications for the non-intrusive real-time temperature and lubricant oil quality and metal debris monitoring have been identified and discussed.

Keywords: Structural health monitoring, Non-destructive evaluation, Integrated ultrasonic transducers, Turbojet engine, Piezoelectric thick films, Sol-gel process.
INTRODUCTION

The increasing demand to improve the performance, reduce downtime, increase reliability and extend the life of engines requires the use of sensors for continuous engine conditions monitoring during development and service. It is established that methods employing piezoelectric ultrasonic transducers (UTs) are widely used for real-time, in-situ or off-line non-destructive evaluation (NDE) of large metallic structures including airplanes, automobiles, ships, pressure vessels, pipelines, etc. because of their subsurface inspection capability, fast inspection speed, simplicity and cost-effectiveness [1-3]. In this investigation the objective is to develop and evaluate effective integrated UT (IUT) technology to perform non-intrusive engine NDE and structural health monitoring (SHM). The intended applications include non-intrusive real-time temperature, lubricant oil quality, and metal debris monitoring within the turbojet engine environment. For this purpose a portable on-site IUT fabrication technique is most desirable. Additionally, because engines operate at elevated temperatures, the developed IUT technology including piezoelectric UTs together with electrical wire, conductive bonding agent, connectors and cables must be assessed. In this study the assessment is aimed and limited to temperatures up to 500°C.

PORTABLE ON-SITE INTEGRATED ULTRASONIC TRANSDUCER FABRICATION

The engine to be used is a modified (fan module removed) CF700 turbojet engine as shown in Figure 1 at the Institute for Aerospace Research (IAR), of the National Research Council (NRC) of Canada. In order to coat high temperature IUT directly onto the engine components and on-site, a portable IUT fabrication kit has been developed. The fabrication of IUTs involves a sol-gel based sensor fabrication process [4-6]. Such process consists of six main steps: (1) preparing high dielectric constant lead-zirconate-titanate (PZT) solution, (2) ball milling of piezoelectric bismuth titanate (BIT) powders to submicron size, (3) sensor spraying using slurries from steps (1) and (2) to produce the thin film UT, (4) heat treating to produce a solid BIT composite (BIT-c) thin film UT, (5) Corona poling to obtain piezoelectricity, and (6) electrode painting for electrical connectivity. Steps (3) and (4) are used multiple times to produce optimal film thickness for specified ultrasonic operating frequency and performance. Silver or platinum paste was used to fabricate top electrodes. BIT-c was used because of its high temperature (500°C) endurance [5, 6]. Figure 2 shows the lower and upper levels of the developed portable IUT fabrication kit. The kit with dimensions of 0.8×0.53×0.3 m³, consists of the following item:

a. BIT powders and PZT solution (used in step 1).
b. One ball milling device, sand papers, detergent, acetone and methanol for sample cleaning (used in step 2).
c. One air brush, one compressor air device, and two glass beakers for cleaning of air brush (used in step 3).
d. One heat gun, one propane torch, two high temperature gloves, and one thermo-couple (used in step 4).
e. One Corona poling device (used in step 5).
f. One silver and one platinum paste pen, and one multi-meter to measure resistance of the electrode and temperature together with a thermocouple. Special high temperature electrical connection accessories are included (used in step 6).
g. One suite case for housing all the components of the portable kit.

Fig. 1: Modified CF700 turbojet engine.

Fig. 2: A portable IUT fabrication kit; (a) interior lower level and (b) interior upper level.

Fig. 3: A portable fabrication kit.

Such a developed portable IUT fabrication kit (Figure 3) can be conveniently carried for on-site application of the technology onto target components. Figure 4 illustrates the instrumentation of two IUTs, consisting of BIT-c films with 500°C platinum paste top electrodes, onto the outer casing of the modified CF700 turbojet engine. This Figure further
shows the bonding conductive material that was applied onto the platinum paste, and the 500°C signal transmitting wires (Figure 4(b)). In general one wire would be sufficient for one IUT; however, for redundancy two wires were installed. The principle of the use of the IUT for the non-intrusive real-time and continuous internal engine temperature measurements is based on using ultrasonic pulses and determining the time of flight as reported in [7, 8].

![Fig. 4: Two IUTs installed on-site at the top of the engine outer casing and (b) 500°C wires bonded to the platinum electrodes.](image)

Additionally, for ultrasonic temperature monitoring of 500°C, IUTs were also coated onto two different engine access covers as shown in Figure 5. These access covers have also been instrumented with conventional thermocouples. In such a configuration, temperature measured by ultrasound can be compared with those measured by thermocouples. These sensors have been installed onto the modified CF700 turbojet engine as shown in Figure 6. Figure 7 shows the typical measured ultrasonic signals traversed back and forth within the gasket thickness in pulse/echo mode at room temperature for the IUT coated onto the gaskets as shown in Fig. 6. The center frequency of these IUTs is around 11 MHz. \( L^n \) is the \( n^{th} \) round trip echo through the thickness of the gasket.

![Fig. 5: Two different engine access covers equipped with 500°C IUTs.](image)
HIGH TEMPERATURE TESTS ON THE ELECTRICAL IUT CONNECTORS

In order to test the fabricated IUTs made of BIT-c film and platinum top electrodes, electrical connectors, and conductive adhesive, a 25.4 mm diameter 150 mm long steel rod, simulating engine component, has been used for the deposition of IUTs (Figure 8(a)). The center frequency of these IUTs is about 10 MHz at room temperature. Such an assembly was placed into a furnace (Figure 8(b)) and thermally cycled from room temperature to 500°C. The results of two thermal cycles are shown in Figure 9. The upper two pictures on the left and two on the right indicate the ultrasonic signals with high signal to noise ratio (SNR) which traveled a distance of 300 mm and 600 mm, respectively. The lowest and right part in Figure 9 indicates the measured ultrasonic velocity profile in the steel rod during the thermal cycles. In the middle of Figure 9, the attenuation number (Attn) indicates the total loss of IUT, electrical connections and ultrasonic attenuation propagating via a round trip of the 150 mm long steel rod. At 500°C this attenuation was only 6 dB.
Fig. 8: (a) One IUT made of BIT-c films on top of a steel rod (delay line) together with top electrode, wire connection, conducting adhesive bond, electrical connector and cable; (b) 500°C thermal cycle test chamber.

Fig. 9: Thermal cycle tests of 500°C BIT-c film IUT directly coated on top of a 150 mm long steel rod including top electrode, conducting bond, wire, connector and cable.

IUTS FOR LUBRICANT OIL QUALITY AND METAL DEBRIS MONITORING

It is also the intention of this work to perform non-intrusive real-time continuous monitoring of the lubricant oil quality and metal debris in the engine oil system. For such purpose one oil sump line and one oil supply line of the turbojet engine, shown in Figure 10, are instrumented. Figure 11 illustrates the instrumented IUTs together with the necessary prototype electrical connections for the oil sump and supply line, respectively. The cables used can only sustain temperatures of up to 200°C; hence they were properly installed as shown in figure 12. Figure
13 illustrates typical measured ultrasonic signals traversed back and forth within the pipe thickness in pulse/echo mode at room temperature for the IUT coated onto the oil sump and supply lines, respectively. The center frequencies of the $L^1$ echo, shown in Fig. 13, are at 10 and 17 MHz, respectively. $L^n$ is the $n$th round trip echo through the wall thickness of the pipe. During actual operations, the ultrasonic measurements will be carried out in the transmission mode and ultrasonic signals with higher SNR than those shown in Fig. 13 are expected. Such an expectation comes from previous experiments which simulated the monitoring of lubricant quality and metal debris [9].

**Fig. 10:** (a) Standard engine oil sump and supply lines, (b) IUTs coated onto special oil sump and supply lines.

**Fig. 11:** (a) Oil sump line (b) Oil supply line equipped with IUTs and prototype electrical connections.
CONCLUSIONS AND DISCUSSIONS

Employing a developed portable on-site fabrication kit, high temperature integrated ultrasonic transducers (IUTs) made of thick BIT composite piezoelectric film have been coated directly onto a modified CF700 turbojet engine outer casing, oil sump and supply lines and gaskets using a sol-gel spray technology that consists of six main steps. The top electrodes, electrical wires, conducting adhesive bond, connectors and cables have been also tested successfully at temperatures of up to 500°C. The center frequencies of these IUTs were around 10 to 17 MHz. Ultrasonic signals obtained in pulse/echo measurements of these IUTs are excellent and it is expected that high temperature ultrasonic performance will be obtained in the transmission mode as well. The potential applications of the developed IUTs include non-intrusive real-time temperature, lubricant oil quality and metal debris monitoring which will be carried out in pulse/echo and transmission mode, respectively.

ACKNOWLEDGMENT

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REFERENCES

IDENTIFICATION OF STIFFNESS DISTRIBUTION AND DAMAGE IN EULER-BERNOUILLI BEAMS USING STATIC RESPONSE

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ABSTRACT

The paper presents two computational inverse procedures for reconstructing the stiffness distribution and for detecting damage in beams using static responses. The complete deflection profile is obtained by processing digital images of the beam and it is utilized as the input for the inverse computational procedures. The first procedure formulation is based on the equilibrium gap through Euler-Bernoulli beam finite element discretization; the second is formulated as a minimization a data discrepancy functional. Examples of a simply-supported beam are simulated to demonstrate the performance of the two techniques. Numerical simulations show that the equilibrium gap method has good performance in locating and quantifying local damage and data discrepancy minimum functional is better used for reconstructing distributed global stiffness. Both two techniques can be integrated to be used as general identification of randomly distributed damage.

Keywords: Equilibrium gap method, adjoint optimization, Inverse problem, Damage identification, Beam stiffness.

INTRODUCTION

The identification of damage in beams have attracted intensive research efforts due to the increasing demands for assessing integrity and reliability of structures in service. Damage accumulation in structures can be associated to stiffness reduction, and expressed mathematically as a stiffness factor or a damage variable related to the stiffness [1-2]. The damage is naturally related to the change in the internal structure of materials within the cross-section of the beam. Damage identification is usually performed based on experimental data of actual beams subjected to external loadings.
Depending on the response data collected, damage identification methods can be divided into two categories: dynamic based and static based. The former category employs dynamic characteristics such as natural frequencies [1][3], displacement modal shapes [4][5], modal shape derivatives [6][7], wavelet analysis of dynamic signals [8], harmonic responses [2][9-10]. A recent overview of dynamic based methods is given in [11]. The literature on static response-based identification is much more limited. Banan et al. gave a formulation estimating the constitutive properties of a finite-element model from measured displacements under a known static loading through minimizing the error in displacements at the measurement sites [12]. Other works include locating crack through wavelet analysis of the static deflection profile [13], using strain integral [14], and single or double crack identification from static loading and displacement measurements at several points [15].

Another way to differentiate identification methods is whether the problem is to locate and quantify damage in the form of simple discrete cracks, or to completely reconstruct a spatially distributed stiffness. Most of the damage detection methods aim at finding the location and quantifying localised damaged zones [14-17]. In this category of approaches, cracks are usually parameterized beforehand, which assumes knowledge of stiffness distribution of the structure. For methods dealing with distributed stiffness, the problem consists at recovering a continuous distribution of the beam’s stiffness. Mathematically, this problem involves a minimisation of a functional, and it is generally a more difficult inverse problem. Usually, the identification of distributed stiffness requires much more information collected testing of the beam. For example, Liu and Chen [9], Kokot and Zembaty [10] used harmonic response at a wide frequency range and spatially distributed points (all the nodes of a finite element model) as input for stiffness reconstruction of beams. Procedures efficient in localizing and quantifying concentrated cracks usually assumes a good knowledge of stiffness distribution of the undamaged structure, while in reconstructing stiffness distribution the damage is assumed continuous across the beam. Numerical algorithms performing well in one method may be inefficient for the other.

Recent progresses in image processing techniques and digital cameras make it possible to measure continuous deflection of structures under static loading from digital images [18]. This progress offers new possibilities in structural parameter identification and damage detection and localization. Unlike point-based testing sensors (displacement gauge, strain gauge, LVDT), digital images (Digital Image Correlation and photogrammetry) are able to provide measurement at a large number of spatially distributed points. The quasi continuous measurement of the beam deflection is more valuable as input for inverse procedure leading to damage detection and health monitoring of structures.

In the present paper, two inverse computational techniques are discussed. One is based the equilibrium gap method formulated on the basis of the equilibrium of internal forces; and the second method is based on a data discrepancy functional to minimize the difference between measured and simulated deflection profile. The total variation (TV) regularization technique is used to overcome the ill-posedness inherent in this type of inverse problems. Examples of a simply-supported beam are calculated showing the performances of the presented two methodologies.
EQUILIBRIUM GAP METHOD

Claire et al [19] presented a formulation based on the equilibrium gap to identify damage in 2D structures. In case of 2D elasticity and in the absence of volumetric loading away from the boundary, the continues format of the equilibrium equations is \( \text{div}(\sigma(E, \nu), \mathbf{u}(\mathbf{x})) = 0 \); where the stress \( \sigma \) is function of material parameters \( E, \nu \) and displacement \( \mathbf{u}(\mathbf{x}) \). Given a measured displacement field, a piecewise-constant material parameter distribution is assumed and the distribution is evaluated through a finite element formulation.

This idea of equilibrium gap formulation can be written specifically for beams where the equilibrium equations can be written as function of the generalised forces at each section:

\[
F_i + F_j = P \quad \text{and} \quad M_i + M_j = T
\]

where the index \( i, j \) denote the internal force to the left and right of a given section, respectively. Variables \( P \) and \( T \) denote the nodal shear forces and moment, respectively at the section under consideration. In contrast with the 2D equilibrium, the external loading is included in the equilibrium expression of beams. The equilibrium of the internal forces and external loads must be satisfied along the beam.

The equilibrium conditions are directly applied to a FE formulation of the Euler-Bernoulli beams. Neglecting the axial deformation and considering an isotropic damage definition, the force displacement equations at the element level of the Euler-Bernoulli beam are given by:

\[
k_d = (1 - D) \frac{E I_0}{L^3} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix} \begin{bmatrix} v_1 \\ \theta_1 \\ v_2 \\ \theta_2 \end{bmatrix} = \begin{bmatrix} V_1 \\ M_1 \\ V_2 \\ M_2 \end{bmatrix}
\]

\[
= (1 - D)k_0d = f
\]

where \((v, \theta)\) are the nodal degrees of freedoms (vertical deflection, rotation angle), and \((V, M)\) are the corresponding nodal forces (shear, moment). Variable \( D \in [0,1] \) is a scalar considered as an index indicating the damage (reduction of stiffness). The reference stiffness is defined as \( EI_0 \), and can vary across the beam, i.e. it stands for \( EI_0(x) \). A possible definition of the damage variable is, \( D = 1 - \frac{EI}{EI_0} \), where \( EI \) is the actual stiffness of the cross section. Using such definition, the beam’s stiffness is expressed as a function of the damage index \( D \), \((k = (1 - D)k_0)\), where \( k_0 \) is the stiffness matrix of an undamaged or reference element. For each element, a constant damage parameter \( D \) is considered, which corresponds to a piecewise-constant definition of damage field along a beam. The parameter \((1 - D)\) corresponds to the stiffness reduction factor.
We will assume that the deflection profile is available from measurements through digital images, for each pair of two adjacent elements \(i\) and \(j\), we have the following expression:

\[
(1 - D_i)(k_0^3)_{i}d_i + (1 - D_j)(k_0^3)_{j}d_j = (V_2)_i + (V_1)_j = P_{ij}
\]

\[
(1 - D_i)(k_0^2)_{i}d_i + (1 - D_j)(k_0^2)_{j}d_j = (M_2)_i + (M_1)_j = T_{ij}
\]

where \((k_0^3)_{i}\) is the third row in the reference stiffness matrix of element \(i\); and \(D_i\) is the damage index of element \(i\). Similarly, \((k_0^2)_{j}\) is the first row of the reference stiffness matrix of element \(j\). The generalised forces \(P_{ij}\) and \(T_{ij}\) are the concentrated load and moment applied at the node connecting element \(i\) and \(j\), respectively. Writing these equilibrium equations for each pair of adjacent elements, we have a system of equations:

\[
A\theta = R
\]

\[
A = \begin{pmatrix}
(k_0^3)_{i}d_i & (k_0^3)_{i}d_2 \\
(k_0^2)_{i}d_i & (k_0^2)_{i}d_2 \\
(k_0^2)_{2}d_2 & (k_0^3)_{2}d_3 \\
(k_0^2)_{2}d_2 & (k_0^2)_{3}d_3 & \cdots & \cdots \\
(k_0^2)_{3}d_3 & \cdots & \cdots & \cdots \\
\vdots & \vdots & \vdots & \vdots
\end{pmatrix}
\]

\[
\theta = (1 - D_1, 1 - D_2, \ldots, 1 - D_{n_{el}})^T, \quad R = (P_1, M_1, \ldots, P_i, M_i, \ldots)^T
\]

The system of linear equations (4) is over-determinate. With the unavoidable noise in measured displacements, the standard least-square solution can be unstable. A total-variation (TV) regularization scheme is used to solve this system [20].

On the practical side, the measurement of the rotation angles of beams is very difficult. Different techniques can be used to overcome this difficulty: (i) a static condensation can be used to eliminate the degrees of freedoms associated to the rotations, (ii) estimate the angles from measured displacements using a stable numerical differentiation such as mollification [21], or (ii) through a minimization of total strain energy while treating the angles as unknown variables.

The equilibrium conditions at the supports or end points are excluded; thus this formulation can be applied for both statically determinate and indeterminate beams. Moreover, the concentrated loads applied to the beam need to be excluded in writing the equations (3) and (4); therefore the regions close to the loads must be excluded from the identification.
DATA DISCEPANCY BASED FORMULATION

Having measurements of the beam’s deflection, the reconstruction of the stiffness of a beam can be formulated as:

\[
\min_\theta J(u, \theta) = \frac{1}{2} \sum_{i=1}^{N} [u_i(\theta) - u_i^m]^2 + \alpha \Phi(\theta)
\]

\[s.t. \quad K(\theta)u = f\]

(5)

where \(N\) is the total number of measurements. \(\theta\) is the vector of unknown parameters (the beam’s stiffness distribution in the present case). The vector \(u^m\) is the measured data displacement and \(u(\theta)\) the corresponding simulated corresponding to as set of distribution parameters \(\theta\). The constraint appearing the minimization problem (5) is the solution by finite element of a direct problem corresponding to a set of parameters \(\theta\).

The cost function contains the sum of squared differences between measured and simulated deflection profile of the beam and a regularization term. Regularization is necessary to stabilize the numerical solution, and to ensure uniqueness. Similar to the equilibrium gap method, the total-variation (TV) regularization is also used here; numerical simulations showed that TV regularization gives better results than the classical Tikhonov method. However, the implementation of TV-regularization is different when dealing with linear system of equations or an optimization problem; implementation details can be found in [20][22].

To solve the minimization problem, we need to evaluate the gradient of the functional. The adjoint method is an efficient algorithm that allows computing the gradient of a cost functional. It is employed in the present work to reconstructs the stiffness distribution through the solution of the constrained optimization problem given in (5). The Adjoint method is used here to compute the gradient vector of the objective function to the unknown stiffness distribution parameters.

The constraint equations are the equilibrium governing equations of the discretized beam using finite element; they are written in the following form:

\[
R(\theta, u(\theta)) = K(\theta)u - f = 0
\]

(6)

where \(u\) is the state variables (displacements in the present case), which is an implicit function of the unknown material variables \(\theta\). We introduce Lagrangian multipliers to modify the constrained optimization problem to an unconstrained optimization. The augmented functional that enforces the governing equations via Lagrange multipliers is expressed as:

\[
L(\theta, u) = J(\theta, u) - \lambda^T R(\theta, u)
\]

(7)
Differentiating the Lagrangian with respect to $\theta_i$, the gradient of the Lagrangian is given by:

$$\frac{dL(\theta, u)}{d\theta_i} = \frac{\partial J}{\partial \theta_i} + \left( \frac{\partial J}{\partial \dot{u}} - \lambda^T \frac{\partial R}{\partial u} \right) \frac{du}{d\theta_i} + \lambda^T \frac{\partial R}{\partial \theta_i} \quad (8)$$

In general, the term $\frac{du}{d\theta_i}$, the derivative of displacement to the material parameters, is difficult to evaluate. By observing that by a suitable choice of $\lambda$, it is possible to make the term in the bracket equals zero, then avoid the need to evaluate the gradient $\frac{du}{d\theta_i}$. A new equation can be written:

$$\left( \frac{\partial J}{\partial u} - \lambda^T \frac{\partial R}{\partial u} \right) = 0 \quad (9)$$

Equation-(9) is the adjoint equation; we solve this equation for $\lambda^T$. In this particular case, equation (9) can be expressed as:

$$(U - U^m) = \lambda^T K(\theta) \quad (10)$$

where $U$ is the displacement vector computed from the finite element equation-(6), and $U^m$ is the measured displacement vector. Therefore we have:

$$\frac{dL(\theta, u)}{d\theta_i} = \frac{\partial J}{\partial \theta_i} + \lambda^T \frac{\partial R}{\partial \theta_i} \quad (11)$$

in which the two derivatives can be easily computed:

$$\frac{\partial J}{\partial \theta_i} = \alpha \frac{\partial \Phi}{\partial \theta} \quad (12)$$

$$\frac{\partial R}{\partial \theta_i} = \frac{\partial K(\theta)}{\partial \theta_i} - \frac{\partial f}{\partial \theta_i} \quad (13)$$

The term $\Phi$ in (12) is the TV-regularization function; the detailed expression for its gradient to $\theta$ can be found in [22]. The global stiffness matrix is the assembly of the elements’ stiffness matrices; and for linear elastic materials, the element stiffness matrix is function of the material elastic properties and the geometric characteristics of the cross-section. Therefore, the derivative of the stiffness matrix to $\theta$, which is either the damage indices, or the stiffness reduction factors as defined in the previous section, and it can be computed analytically.

In conclusion, the adjoint method consists of two consecutive steps: 1) solve equation-(9) for $\lambda^T$, then insert $\lambda^T$ into equation-(11) to calculate the gradient of Lagrangian. With the gradient calculated from adjoint method, classical gradient-based optimization techniques can be used to find the unknown stiffness parameters. In the present paper the gradient-
assisted optimization algorithm available in MATLAB optimization toolbox is used in the simulation illustrated below [23].

**NUMERICAL EXAMPLES**

Application of the Equilibrium gap method

A 10m long simply-supported beam is simulated to demonstrate this method. This beam is assumed to have a constant stiffness along its span; a concentrated load of 5 kN is applied at its center. The stiffness at 2.5 m and 6 m from the left support are reduced to 50% and 70% of the original value (D = 0.5 and D=0.7), respectively. The simply-supported beam is discretized using 100 beam elements; hence, the continuous stiffness distribution function is represented by 100 discrete unknown parameters. A finite element simulation of the deflection profile is used as the measured value. A Gaussian noise is added to the simulated response to emulate real measured signals as expressed by equation (14).

\[
noise(t) = NRND \cdot a \cdot RMS(u)
\]  

(14)

where \(NRND\) is a Gaussian (normal) random number with zero-mean and unit standard deviation; \(a\) is the applied noise level; and \(RMS(u)\) is the root-mean-square of the measured displacement \(u(x)\). In the numerical examples angles are obtained from displacements by minimization of strain energy. The TV-regularized equations are solved using a Primal-dual method detailed in [22].

Figure 1-(a) shows the identified stiffness with no noise added to the deflection measurements; the two damaged zones are perfectly identified; the small variations of damage index are due to the error induced by the recovery of rotation angles from displacements. In Figure 1-(b) and (c), noise levels of 2.5% and 5% are added to the measured deflection data, respectively. The numerical simulations still show the position of the damage. However, it is observed that the noise in the measurement can mask fine-scale damages. If the level of noise is known, the minimum scale of detectable damage corresponding to this level of noise can be obtained by visual inspection. For example, at a 5% noise level, the finest detectable level of damage seems to be 10%. In Figure 1-(d), 10% noise is added, and the damages levels estimated are 30% and 10% for the two regions.
Application of the Adjoint optimization method

In this example, the problem consists at reconstructing the stiffness of a beam from the measurement of the deflection curve. For this validation, the experimental data are emulated from a finite element analysis. A noise level of 5% is introduced to the emulated displacement field. For the first test, we try to recover the stiffness of a beam with constant stiffness. The initial guess takes a uniform stiffness factor of 0.5. The identified and reference distribution of stiffness factors are shown in Figure 2. The results show a good performance of the proposed procedure for regions away from the supports. The displacement of a simply supported beam close to the support are very small (zero at the support), there is not enough information available in this region to improve the initial guess.

For the second case, we attempt to reconstruct a beam’s stiffness with parabolic distribution. The results are shown in Figure 3. Again a constant stiffness factor of 0.5 is used for the initial guess to start the iteration.
CONCLUSION

Two novel inverse computational techniques, equilibrium gap method and adjoint-optimization method, are proposed and implemented for the identification of stiffness distribution and detection of damages in beams. The former is based on the equilibrium gap of the beam; and the latter uses the data discrepancy functional between measured and simulated displacements. Optical measurement of continuous deflection profile under static loading is used as input in these techniques. Two examples are illustrated to validate the accuracy and efficiency of these methods. The equilibrium gap method is efficient in identifying localized damage and the data discrepancy method is a good candidate for the recovery of continuous stiffness distribution. Experimental tests are to be conducted to study the performance of the two techniques in practical engineering applications.

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HIGH TEMPERATURE FLEXIBLE ULTRASONIC TRANSDUCERS FOR STRUCTURAL HEALTH MONITORING AND NDT

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ABSTRACT

Flexible ultrasonic transducers (FUTs) consisting of a 75 μm thick metal membrane, a piezoelectric composite with a thickness larger than 85 μm and a top electrode were developed for structural health monitoring (SHM) and non-destructive testing (NDT) applications. The piezoelectric films were made by a sol-gel spray technique. Its ultrasonic performance in terms of signal strength is at least as good as commercially available broadband ultrasonic transducers at room temperature. Onsite gluing and brazing installation techniques which bond the FUTs onto steel pipes for SHM and NDT purposes up to 100°C and 150°C are developed, respectively. The best thickness measurement accuracy of FUT at 150°C was estimated to be 26 μm.

Keywords: Flexible ultrasonic transducer, High temperature, Thick piezoelectric film, Non-destructive testing, Structural health monitoring, Brazing.
INTRODUCTION

The non-destructive testing (NDT) of pipes in nuclear and fossil fuel power plants [1-3], chemical and petroleum plants [1, 2] and other structures [4] has become an increasingly important element in the improvement of safety and in the extension of a structure’s life span. Structural health monitoring (SHM) [5, 6] is a furtherance of the NDT approach. Ultrasonic techniques are frequently used for these NDT and SHM purposes because of their subsurface inspection capabilities, fast inspection speeds, simplicity and ease of operation. In these applications, ultrasonic transducers (UTs) may need to be made directly in contact with structures that have surfaces with different curvatures and must also be able to operate at elevated temperatures. However, conventional UTs having rigid flat end surfaces may in general show poor performance and therefore may not be readily suitable for the inspection of pipes at elevated temperatures. The poor signal-to-noise ratio (SNR) may in part come from the fact that the contacting area is a narrow line allowing only a small portion of the available ultrasonic energy to be transmitted into the pipe. Different thicknesses in gel couplant that are present in the gap area between the UT flat end surface and the curved surface of the pipe cause the areas other than this narrow contact line of the UT to generate many unwanted noises. Flexible UTs (FUTs) are therefore more suitable under such conditions because they ensure their self-alignment to the object’s surface even in the case of a curved or complex geometry so that the transmitted ultrasonic energy may be maximized and noises may be minimized to improve SNR for NDT or SHM purposes [7 - 9].

In commercially available FUTs, piezoelectric polymers such as polyvinylidene fluoride (PVDF) [8] and piezoelectric ceramic/polymer composites [8, 10] are mainly used as piezoelectric materials. Both materials include polymer, which prevents the use of such flexible UTs at elevated temperatures. Other flexible, high temperature ultrasonic transducers (HTUTs) have also been reported [11]. In order to provide flexibility for the single crystal films used, the thickness of the piezoelectric films were thin, from 0.2 to 10 μm. Thus operating frequencies were normally higher than 30 MHz and these may not be suitable for NDT and SHM of thick and highly attenuating materials.

The objective of this investigation was to develop FUTs that have high flexibility similar to PVDF FUTs [7], but can operate at up to at least 150°C and have a high ultrasonic performance comparable to commercial broadband UTs. In addition to the use of a high temperature couplant for momentary contact NDT, a gluing and a brazing onsite installation technique will be investigated for permanent NDT or SHM over curved surfaces such as on steel pipes and at temperature up to 150°C. The glue and the brazing material between the FUT and the external surface of the pipe serves as a permanent high temperature couplant.

FABRICATION AND PERFORMANCE OF FUT

Fabrication procedures

The fabrication of the FUT consists of a sol-gel based sensor fabrication process [12]. The substrate can be a 75 μm thick titanium (Ti) or stainless steel (SS) membrane. Such an approach consists of six main steps [9, 13]: (1) preparing a high dielectric constant lead-zirconate-titanate (PZT) solution, (2) ball milling the piezoelectric PZT powders in a PZT solution to submicron
sizes. In this case, PZT powders were selected because of their high piezoelectric strength and Curie temperature of 350°C. (3) sensor spraying using slurries from steps (1) and (2) to produce a film with thicknesses of between 5 and 20 μm, (4) heat treating to produce a solid PZT composite (PZT-c) thick film, (5) Corona poling to obtain piezoelectricity, and (6) silver paste painting for depositing electrical connections. Steps (3) and (4) are performed multiple times to produce optimal film thicknesses for specified ultrasonic operating frequencies. In this investigation, steps (4) and (5) are significantly different from the FUT processes presented in [9]. Here a rapid thermal annealing like process [14 - 16] was used for the thermal treatment. The main advantage of rapid thermal annealing is to limit the eventual degradation of the film-substrate interface such as through inter-diffusion of lead and also to improve the crystallization behavior of the PZT film. The short annealing time also avoids the growth of an oxidation layer between the Ti or SS and PZT-c film. An oxidation layer would reduce the electrical conductivity of Ti and SS, thereby decreasing the ultrasonic performance. In this study longitudinal (L) ultrasonic wave transducers are described. The relative dielectric constant was calculated using the capacitance measured by an impedance analyzer.

Performance of FUT

Firstly, an FUT was made with a ~120 μm thick PZT-c film that was deposited onto a 75 μm thick Ti membrane. It is noted that the ultrasonic performance of such FUTs on Ti membranes showed in general a 5 dB stronger signal strength than those reported in [9], whereby FUTs were made onto SS membranes. The improved signal strength comes from the reduced oxidation of the membrane substrates (Ti over SS) during heat treatments and improvement of the sol gel spray technique. The FUT was attached to a 15.1 mm thick aluminum (Al) plate using a gel couplant as shown in Fig. 1a. The pulse-echo measurement was carried out by a handheld EPOCH model LT pulser/receiver (from Olympus-Panametrics, USA) at room temperature. The EPOCH LT is commonly used for NDT in industrial environments. The diameter of the top silver paste electrode of this FUT was 7 mm, which was the size required to match the electrical impedance of the PZT-c film with that of the pulser/receiver and to achieve maximum signal strength in pulse-echo mode at room temperature. The measured ultrasonic data in pulse-echo mode is shown in Fig. 1b, where \( L^n \) is the \( n \)th trip L echo through the plate thickness. The center frequency and the 6 dB bandwidth of \( L^1 \) echo are 8.1 MHz and 5.3 MHz respectively. In Fig. 1b, 0 dB gain out of the available 100 dB receiver gain of EPOCH was used. The SNR of the \( L^1 \) echo is ~38dB. The SNR is defined as the ratio of the amplitude of the first echo (here \( L^1 \)) over that of the surrounding noise. For comparison, Figs 2a and 2b show measured results obtained when commercial broadband UTs with a center frequency of 5 MHz and 10 MHz, respectively, were used at the other side of the Al plate shown in Fig. 1a together with the same gel couplant. The receiver gains used by the EPOCH pulser/receiver were 2 dB and 2 dB, respectively. These results show that while using the EPOCH, the signal strength of the FUT was at least as good as those of the two commercially purchased broadband UTs.
Fig. 1: (a) Measurement setup for a FUT made of PZT-c film attached to an Al plate using an EPOCH LT in pulse-echo mode; (b) Measured ultrasonic signals at room temperature.

Fig. 2: Measured ultrasonic signals from commercial UTs with a center frequency of (a) 5 MHz and (b) 10 MHz operated in pulse/echo mode at the opposite surface of the Al plate.

ULTRASONIC THICKNESS MEASUREMENTS OF PIPES
AT ELEVATED TEMPERATURES

Momentary NDT with FUT

Developed FUTs can be operated in pulse-echo, pitch-catch and transmission configurations. Fig. 3a shows a magnet holder that was used to attach a FUT to a steel pipe with an outer diameter (OD) of 76 mm and an inner diameter (ID) of 50 mm at 150°C. The thickness of the pipe was 13 mm. The temperature was achieved with a hot plate and was controlled by the monitoring of a thermocouple in contact with the pipe. Oil was used as the ultrasonic couplant between the FUT and the external surface of the pipe. A momentary contact method such as the example given here is often used during NDT applications. The same FUT together
with the holder can be used at different locations. The FUT had a 95 µm thick PZT-c film and the diameter of the top silver paste top electrode was 5 mm. Fig. 3b shows the measured ultrasonic signal, where \( L_n \) is the signal traveling through the thickness for the \( n \)th time. The 5 mm top electrode diameter was chosen not to achieve maximum signal strength, but rather to achieve maximum SNR of the \( L_1 \) for thickness measurement. The center frequency and the 6 dB bandwidth of the \( L_1 \) echo were 10.9 MHz and 2.9 MHz, respectively. For this measurement, the holding of the FUT onto the steel pipe was using a magnet as shown in Fig.3a. A gain of 52 dB was used out of the available 100 dB gain provided by the EPOCH LT. The high 52 dB gain may be explained by the presence of the oil couplant, the 150°C high temperature, not optimized top electrode diameter, the insufficient contact force applied to the FUT and the ultrasonic attenuation within the thickness of the pipe wall. The SNR of \( L_1 \) is ~24 dB. The measurement confirms that such a FUT is useful for thickness measurements of a pipe at up to at least 150°C. Certainly if there is a defect existing within the 13 mm thickness of the pipe under the top electrode, NDT of the said defect may be achieved. If the thickness of the pipe is thin enough compared to the wavelength, such a FUT may be able to generate and receive guided waves along the pipe.

![Fig. 3: (a) A FUT is attached onto a steel pipe by a magnet holder and operated at 150°C and (b) shown is its measured ultrasonic signals in pulse-echo mode.](image)

**Permanent NDT with glued FUT**

In certain situations, NDT of pipes must be carried out continuously. The momentary contact approach presented in the previous section may not be appropriate; therefore a onsite gluing technique is performed. Let FUT consisting of a 75 µm thick Ti membrane, a 85 µm thick PZT-c film, a 5 µm thick silver paste was made, glued onto a steel pipe, and heated by a hot plate up to ~100°C as shown in Fig. 4a using a glue which can be cured at room temperature. The pipe was first washed with water, soap and methanol. A thin layer of glue was then deposited on the backside of the FUT. It is then clamped onto the cleaned pipe using a metallic worm clamp and a SS plate that is conformed over the FUT and onto the curvature of the pipe. The OD and the wall thickness of the pipe are 89 mm and 6.5 mm, respectively. In Fig. 4a, there are two top electrodes and each of them serves as one FUT. Using one of the two FUTs the ultrasonic measurement was performed by a handheld EPOCH LT pulser/receiver.
together with two spring electrical contacts of which one connects to the top silver paste electrode and one to the bottom electrode which is the Ti membrane. $L^n$ is the $n^{th}$ round trip ultrasonic echoes within the wall thickness of the pipe. The pulse energy used was the lowest available and the gain was 5 dB out of the available 100 dB at 100°C.

![Image](image.jpg)

**Fig. 4:** (a) A FUT fabricated on a 75 µm thick Ti membrane and glued onto a steel pipe with the ultrasonic measurements displayed using a handheld EPOCH LT pulser/receiver (b) Measurement results obtained at room temperature (upper trace) and 100°C (lower trace).

Fig. 4b shows the pulse/echo measurement results obtained at room temperature (upper trace) and 100°C (lower trace). The center frequency and 6 dB bandwidth of the $L^1$ echo of the lower trace were 11.9 MHz and 4.1 MHz, respectively. The signal strength of the $L^1$ echo obtained at 100°C (lower trace) reduces only about 1 dB comparing to that obtained at room temperature. The onsite installation of FUTs using glue cured at room temperature is therefore suitable for SHM of pipes within this temperature range.

**Permanent NDT with brazing FUT**

Also for permanent NDT and SHM another onsite installation technique using brazing was also performed. Fig. 5 shows a 15 mm wide and 23 mm long 75 µm thick SS membrane which is brazed onto a steel pipe with an OD of 26.6 mm and pipe wall thickness of 2.5 mm. For brazing 75 µm SS rather than Ti membrane was selected for FUT due to its ability to be brazed onto steel pipes with a small compromise to be made with respect to the minor level of oxidation that develops on the SS membrane during heat treatments. The developed onsite installation approach for brazing the FUT onto steel pipes is also simple and easy to follow. Similar to the gluing technique the pipe was first washed with water, soap and ethanol and a thin layer of brazing paste was then deposited on the backside of the FUT as evenly as possible. The FUT is then clamped onto the cleaned pipe using a metallic worm clamp and a SS plate that is conformed over the FUT and onto the curvature of the pipe. The induction heating system was calibrated to maximize the power and frequency of the system and it was controlled by a two-color pyrometer. The induction lasted three minutes and the temperature required was 825 °C for this SS substrate FUT to be brazed onto the steel pipe. After the induction, Corona poling was used to make the film piezoelectric. The developed Corona poling method is a convenient method to pole the FUT that has been brazed onto curved pipes. After poling, a silver paste was used to deposit the top electrode of necessary size.
Table 1 demonstrates the coupling efficiency of the brazing material existing between the SS membrane and the external surface of the pipe under the SS membrane. It was measured with a commercial broad bandwidth UT centered at 10 MHz. Let us assume that 0 dB is the coupling efficiency of a commonly used gel couplant between a UT and the pipe. The amount of dBs appearing in nine measurement points on the pipe proves that the brazing material between the SS membrane and the external surface of the pipe was uniform and that the average loss was about 5 dB. Fig. 6a shows a FUT brazed onto this pipe and the measurement results taken at 150°C are presented in Fig. 6b. The gain used with the EPOCH was 20 dB out of the available 100 dB. This FUT had a 113 µm thick PZT-c film and the diameter of the top silver paste top electrode was 2.5 mm. The 2.5 mm was chosen not to achieve maximum signal strength, but rather to achieve maximum SNR of the L₁ for thickness measurement. Spring-loaded pins, one of which was connected to the top electrode and the other, connected to the grounded steel pipe, served as the electrical contact. The 20 dB gain may be explained by the presence of the brazing material, the 150°C high temperature, not optimized top electrode diameter and the ultrasonic attenuation within the thickness of the pipe wall. Fig. 5b indicates that the brazing technique uniquely used for such a FUT is an excellent approach to bond a FUT to a pipe with an OD of 26.6 mm for NDT or SHM measurements performed at up to at least 150°C.

Table 1: Ultrasonic coupling efficiency of brazing material.

<table>
<thead>
<tr>
<th>Position of brazed SS</th>
<th>Coupling efficiency (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
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<td>4</td>
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<td>6</td>
<td>3</td>
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<tr>
<td>7</td>
<td>6</td>
</tr>
<tr>
<td>8</td>
<td>6</td>
</tr>
<tr>
<td>9</td>
<td>6</td>
</tr>
</tbody>
</table>
Fig. 6: (a) A FUT brazed onto a steel pipe and (b) ultrasonic measurement through thickness at 150°C.

Thickness measurement accuracy

Equation (1) (Equation 19 in [17]) is used here for the estimation of the measurement accuracy for time delay and then that for thickness of the steel pipe shown in Fig. 6a.

\[
\sigma(\Delta t - \Delta t') \geq \frac{3}{2f_0^2 \pi^2 T (B^2 + 12B)} \left( \frac{1}{\rho^2} \left[ \frac{1}{SNR_1} + \frac{1}{SNR_2} \right] - 1 \right)\]

(1)

Table 2: Parameters for Equation 1 and digitization resolution

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values for the brazed FUT on steel pipe</th>
</tr>
</thead>
<tbody>
<tr>
<td>(f_0)</td>
<td>10.8 MHz</td>
</tr>
<tr>
<td>(T)</td>
<td>0.88 µs</td>
</tr>
<tr>
<td>(B)</td>
<td>0.32</td>
</tr>
<tr>
<td>(\rho)</td>
<td>0.91</td>
</tr>
<tr>
<td>(SNR_1)</td>
<td>26 dB</td>
</tr>
<tr>
<td>(SNR_2)</td>
<td>20 dB</td>
</tr>
<tr>
<td>(\sigma(\Delta t - \Delta t'))</td>
<td>2.66 ns</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Digitization resolution (100 MHz including interpolation)</th>
<th>2 ns</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total time delay uncertainty</td>
<td>4.66 ns</td>
</tr>
<tr>
<td>(V_L)</td>
<td>5682 m/s</td>
</tr>
<tr>
<td>Thickness measurement accuracy</td>
<td>26 µm</td>
</tr>
</tbody>
</table>

In this equation, \(f_0\) is the center frequency, \(T\) is the time window length for the selection of \(L^1\) and \(L^2\) in Fig. 6b that is required for the cross correlation measurement, \(B\) is the fractional bandwidth of the signal which is the ratio of the signal bandwidth over \(f_0\), \(\rho\) is the correlation coefficient, \(SNR_1\) and \(SNR_2\) are the SNR of the 1st echo and 2nd echo respectively, and \(\rho(\Delta t - \Delta t')\) is the standard deviation of the measured time delay (\(\Delta t\) being the true time delay and \(\Delta t'\), the estimated time delay). Using Equation 1, the calculated \(\rho(\Delta t - \Delta t')\) was 2.66 ns. Since a sampling rate of 100 MHz was used in the experiment, with the use of the cross correlation
method including interpolation [18], the time measurement error, which may be additionally introduced, was estimated to be 2 ns. The total uncertainty in time delay measurement was therefore 4.66 ns. Since the measured longitudinal velocity $V_L$ in the steel substrate using the pulse-echo technique at 150°C was 5682 m/s, the best possible thickness measurement accuracy achievable using the above parameters given in Table 2 was 26 µm in pulse/echo mode at 150°C. If the sampling rate is increased to more than 100 MS/s, improved thickness measurement accuracy may be obtained.

CONCLUSIONS

Flexible ultrasonic transducers (FUTs) consisting of a 75 µm thick metal membrane, a piezoelectric PZT composite with a thickness larger than 85 µm and a top electrode were developed for NDT and/or SHM applications. The piezoelectric films were made by a sol-gel spray technique together with rapid thermal annealing. The main advantage of rapid thermal annealing is to limit the eventual degradation of the film-substrate interface such as through inter-diffusion of lead and also to improve the crystallization behavior of the PZT film. The short annealing time also prevents oxidation between the Ti or SS and PZT-c film of the FUT. An oxidation layer would reduce the electrical conductivity of Ti and SS, which would decrease the ultrasonic performance. In this study, the ultrasonic performances in terms of signal strength of a FUT were at least as good as commercially available 5 MHz and 10 MHz broadband ultrasonic transducers at room temperature. One FUT developed was used for pipe thickness measurements at 150°C using momentary contact with a high temperature ultrasonic couplant. The onsite installation gluing and brazing techniques were used to glue or braze a FUT to a steel pipe to serve NDT and/or SHM applications such as the monitoring of erosion and corrosion of the inner pipe surfaces. The Corona poling method was conveniently applied to the FUT after the brazing technique. At 150°C the best possible thickness measurement accuracy was estimated to be 26 µm for a steel pipe with a 26.6 mm outer diameter and a thickness of 2.5 mm.

ACKNOWLEDGMENT

Financial support of J.-L. Shih from the Natural Sciences and Engineering Research Council of Canada and the technical assistance of J.-F. Moisan are acknowledged.

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ABSTRACT

The primary objective of Structural Health Monitoring (SHM) is to determine if a structure is performing as expected or it has anomaly in its behaviour as compared to the normal condition. It is also useful in detecting the existence, location and severity of damage. SHM serves to know how the behaviour of a structure changes over time and such information can be used for assessing the rate of degradation in the structure. Vibration Based Damage Identification (VBDI) applied to structural health monitoring can be very useful in interpreting the global vibration response of a structure to identify local changes (e.g. damages) in it. Due to complicated features of real life structures there are some uncertainties involved in its key input parameters (e.g. measured frequencies and mode shape data) where as output is highly sensitive to errors in modal parameters. If Vibration based methods are incorporated with semi-analytical methods such as neural networks and statistical pattern recognition techniques; better accuracy can result in structural health assessment. So far very limited work has been done in SHM using statistical pattern recognition paradigm. This paper contains the study to detect damage of structures by VBDI dynamic analysis method and by statistical pattern recognition approach. The methods have been applied to the Crowchild Bridge in Calgary to detect various damages. The Damage Index and Matrix Update algorithms have been chosen for this analysis. Another in service structure Portage Creek Bridge, British Columbia, has been considered for damage detection using Statistical Pattern Recognition techniques. After filtering and normalizing the data, damage detection features have been extracted by Auto Regressive (AR) Modeling of time series. Both idle and excited bridge conditions are considered in this case.

Keywords: SHM, VBDI, Statistical Pattern Recognition.
INTRODUCTION

Structural Health Monitoring (SHM) originated in aerospace engineering. In the last two decades it has created research interests in other disciplines of engineering including civil engineering. In general, SHM is concerned with performance monitoring of structures to ascertain the strength and performance states of critical members of the structures and determine the presence of any anomaly such as damage, or evaluate its degradation and remaining service life. Most damage detection methods for in-service structural components are Non Destructive Evaluation (NDE) techniques, which are local methods. Vibration Based Damage Detection (VBDI) and Statistical Pattern Recognition provide global techniques which could be practical and cost effective ones for structural condition assessment. VBDI depends on the change of dynamic characteristics of the structures. These characteristics are natural frequencies, mode shapes and damping properties. These characteristics directly depend on material properties, geometry and support condition which contributes to the stiffness and also the distribution of mass. Damage can cause change to any of these dynamic characteristics. Therefore, the VBDI method uses any change to dynamic or modal parameters of structures to identify, locate and detect the severity of the damage. This paper describes the application of VBDI method to an existing structure- Crowchild Bridge, Calgary.

Pattern Recognition Technique is arguably the newest of all global damage detection methods. According to Sohn et al (2000), sensors measuring strains and vibration of a structure produce signals that always respond to the change of environmental and operational conditions. Each group of signals can be considered a pattern (a definable entity) that has some relation to the structural and ambient condition. Pattern recognition is aimed for machine learning process, i.e., ability of a computer to identify and classify (group) them to make a decision. It is this feature that makes it very much attractive to create automated structural health monitoring system. Once a suitable pattern recognition system implemented on a computer that is linked to a database of the sensing system, it will automatically diagnose the structure without human involvement. Recognition of patterns can be divided into two types: (a) Supervised Learning, where input patterns of the vibration of a structure are compared to a pre-defined class; and (b) Unsupervised Learning, where patterns of vibration are categorized to undefined classes. Here the Statistical Pattern Recognition Technique has been applied to an in service structure- Portage Creek Bridge, British Columbia

MODEL-BASED VBDI METHOD

Among several algorithms for VBDI methods, the following two have been used here for their relative merits as explained in Humar et al. (2006): (a) Damage Index Method, and (b) Matrix Update method. These methods are applied on the FEM model of a real structure for simulated damage based on practical input of Dynamic characteristics. The effectiveness of the methods to identify damage of different levels at various elements has been tested. The ability of the algorithms in the presence of simulated random errors has also been examined. The comparison is made between the cases considering measurement errors and without error in the modal properties. The details of these methods are available in Bagchi et al. (2009) and they are omitted here for the lack of space.
The case study used in study of VBDI methods is the Crowchild Bridge (ISIS, 2008) located in Calgary, Alberta, which is a two lane traffic overpass with three continuous spans and reported to be the first continuous steel free deck bridge in the world (Tadros et al., 1998). The bridge was constructed without any internal steel reinforcement to avoid the problem of corrosion in steel. The concrete deck is externally constrained by steel strap to provide anching action to resist transverse loads, and nominal amount of fiber reinforced polymer (FRP) reinforcing bars or fiber reinforced concrete are provided for crack control. The cantilever portion of the deck requires flexural reinforcement which is usually provided using FRP bars. Damage detection algorithms have been implemented in a finite element program M-FEM developed by Bagchi et al. (2007).

**STATISTICAL PATTERN RECOGNITION METHODS IN SHM**

All SHM processes rely on experimental data with inherent uncertainties. Statistical analysis procedures are necessary if one is to identify the dynamic nature of structure including effect of sudden change due to live load and also steady change of temperature over time. The basic components of statistical pattern recognition are: (a) operational evaluation; (b) data acquisition and cleansing; (c) feature extraction; and (d) statistical model development. Feature extraction is the process of the identifying damage-sensitive properties derived from the measured vibration response that allows one to distinguish between the undamaged and damaged structures (Sohn et al. 2000). Typically, systematic differences between time series from the undamaged and damaged structures are nearly impossible to detect by human eyes. Therefore, other features of the measured data must be examined for damage detection. Typically mathematical processes for time series analysis are utilized to extract features.

Statistical model development is a technique that implements algorithms to analyze the distribution of extracted features to determine the damage state of the structure. The appropriate algorithm to use in statistical model development will depend on the availability of data of damaged states. In this study, data for known damaged state of the structures are not available. Therefore outlier analysis, such as the X-bar Control Chart (Fuget et al. 2000) is performed for feature comparison. Control chart analysis is the most commonly used Statistical Process Control technique for Outlier Analysis, and it is also suitable for automated continuous system monitoring. Statistical Pattern recognition technique is tested on Portage Creek Bridge located in Victoria, British Columbia (BC) in Canada. It is disaster Route Bridge and the columns of one of the piers are the bridge has been retrofitted with FRP wraps to improve their seismic performance (Huffman et al., 2006). The data from the strain gauges installed on the columns have used here to evaluate their patterns and condition of the structure.

**DAMAGE SIMULATION IN THE CROWCHILD BRIDGE**

**Description of the finite element model:**

An analytical model of the Crowchild Bridge is constructed here using three dimensional beam elements for the piers, girders, diaphragms and cross frames including the steel straps, and shell elements for the deck and side barriers. Initially the concrete was assumed to be un-cracked. The model based on this assumption was updated and correlated with the data obtained from the vibration test conducted in 1997 by the University of British Columbia.
During the process of model updating, the stiffness coefficients of individual elements were modified to fine tune the resulting modal frequencies of the system. The bridge deck has 15 slab segments. The target area for testing is the segment in the middle along with the longitudinal girders at that location (Fig. 1). All elements at that location are simulated for damage scenario by reducing their stiffness in the FEM by certain fractions which are also considered damage severity factors or simply damage factors. The elements in this area are elements 53 through 58, 174, 177, 180, 183 and 186.

**Fig. 1: Finite Element model of the bridge**

![Damage affected zone](image)

**Fig. 2: Graphs showing damage with element numbers for no error and complete modes with 10% damage severity (a) Damage Index and (b) Pseudo Inverse Method**

**Damage simulation and detection by VBDI methods:**

The following cases for each damage factor were considered: (a) Case- I: No errors in frequency and mode shape measurements and complete mode shapes; (b) Case- II: No errors in frequency and mode shape measurements and incomplete mode shapes; and (c) Case- III: Including error up to 1% in both frequency and mode shape measurements and incomplete mode shapes. All these cases have been analyzed for different extents of damage (i.e., damage factors). Figure 2 shows damage detected using the VBDI methods when the severity of damage is 10% in certain locations (Element 53-58, and 174-178). The damage index can successfully indicate the locations (Fig. 1a), while the matrix update method estimates the
severity well, considering the mean value (Fig. 1b) when no error in frequency and mode shape measurements and complete mode shapes (i.e., Case-I). Similar analysis for Case -II and Case-III has also been performed. It is observed that incomplete mode shapes and some percentage of errors in frequencies did bring down the accuracy of the results comparing to first case. In practice, however, mode shapes are incomplete and the data contains measurement noise. Hence the accuracy of VBDI method may not be good. Other methods such as the Statistical Pattern Recognition methods could work well in such cases.

**STRAIN MONITORING IN THE PORTAGE CREEK BRIDGE**

**Description of the monitored structure:**

The information on the sensing system and also some other research work is available in a research paper by Huffman et al (2006). The Portage Creek Bridge is a 124 m long, three-span structure with a reinforced concrete deck supported on piers, and abutments on H piles. The bridge was designed and built prior to the introduction of current bridge seismic design codes and construction practices. Therefore, it was not designed to resist the earthquake forces as required by today’s standards. Later dynamic analysis done on the two piers showed that strength of the columns of Pier-2 was insufficient. The two columns of the bridge pier were strengthened with GFRP (Glass Fiber Reinforced Polymer) wraps, and eight bi-directional rosette type strain gauges and four long gauge fiber optic sensors attached to the outer layer of the wraps (Fig. 3). In addition, two accelerometers were installed on the pier cap above the columns and a traffic web-cam mounted above the deck.

![Diagram of Portage Creek Bridge Elevation of Pier-2 (short columns) with sensor locations.](Huffman et al. 2006)

**Fig. 3:** Portage Creek Bridge Elevation of Pier-2 (short columns) with sensor locations. (Huffman et al. 2006)
Data Collection And Pre-Processing:

The data is acquired locally by the on-site data acquisition system and transmitted to a central server in order to be accessible through internet using an interactive web page at ISIS Canada’s web site. The earliest and the last time of data available for downloading from the database used here are 2003-04-23 13:54:33 and 2006-08-26 17:58:57 respectively. However there are some periods when data are not available. These off-times have occurred at all the range of the monitoring. The duration of off times varies from a few hours to several months. 30 strains of 16 2D-strain gauges, 6 accelerometer readings of two 3D accelerometers and 1 temperature data available for any starting time falling in the range limited by the time mentioned above. Sampling rates available are 1/32s, 1s, 10s and 1min. Number of data points available in a single download varies from 32 to 30,000. Data can be obtained both graphically and numerically.

![Time Series plot of S_1_1_C1 of 30,000 readings at 1s interval](image-a)
![Time Series plot of S_1_1_C1 of 256 readings at 1s interval](image-b)
![Time Series plot of S_1_1_C1 of 256 readings at 1/32s interval](image-c)
![A time series of strain S_1_1_C1 at Steady State](image-d)

**Fig. 4:** Time series of strain S_1_1_C1: (a) strain variation during Apr 14-15, 2006 (1 Hz data); (b): a typical strain spike due to live loads on Apr 15, 2006 (1 Hz data); (c): strain variation at the spike with higher resolution (32 Hz data); (d): typical strain variation at the steady state.

The sensor data are extracted from the SHM database for the Portage Creek Bridge and viewed graphically for various sample rates and periods. For example, Fig. 4a shows the data for S_1_1_C1 (strain gauge reading 1 along strain direction 1 of column 1) taken at 1 Hz sampling rate (1 data per second) of 30000 points starting at time 17:48:57 on April 14, 2006.
This data time duration is 8 hours 20 minutes. Some random vertical spikes are observed in the data which were perhaps resulted from some sudden impacts. Four of them are long enough to be distinctive. Further analysis revealed that slopes in the strain curve are mostly the effect of temperature changes and the random spikes are the results of moving loads, possibly heavy vehicles. On gradually zooming in by narrowing the data range and picking the starting time very close to one of the vertical spikes, a clearer picture emerges as shown in Fig. 4b. Sampling rate of the signal block is 1 Hz and length is 256 points. If the sampling rate is increased to 32Hz for higher resolution, the strain variation shown in Fig. 4c reveals a hump which is an indication of a moving heavy vehicle on the deck over the bridge pier. The time duration of this block is 8 seconds. It shows clearly that the change of strain over a short period of time resulting from suddenly changing load. In order to study the structural behaviour of the bridge we need to analyze it under two conditions, 1) steady state when only small oscillations are observed (e.g. Fig. 4d) 2) the agitated condition as shown in Fig. 4c in addition to small oscillations.

**Damage Identification by Statistical Model Development:**

The data or signal blocks are collected for 4 types of analysis: (a) steady state strain; (b) live load strain; (c) accelerometer reading under live load; and (d) temperature effect on strain. When the structure undergoes damaged or weakened conditions, the mean and/or variance of the extracted features should change accordingly. There are 132 signal blocks for selected strains and 124 signal blocks for accelerometer and 108 blocks are arranged chronologically for each reading type. For the statistical process control analysis according to Nair and Kiremidjian (2006), the first three AR coefficients give most robust damage indication. In this work the first four coefficients of the AR analysis of each block are considered. The mean and standard deviation of the first quarter of the arranged features are taken as basic mean and standard deviation. As mentioned earlier, an outlier analysis using X-bar control charts (Fuget et al. 2000) is employed here to monitor the changes of the selected feature over time. Subgroup of 4 features is considered here. The subgroup size is taken as 4 according to the suggestion of Montgomery (1997).

**Damage Identification by Pattern Comparison Method:**

For applying this method several sample data blocks for selected strains and accelerometer data were taken. The first block of a particular strain or accelerometer is considered the reference block and rest of the blocks of the set are called test data blocks for comparison. Two methods are utilized for structural health monitoring by statistical pattern recognition: 1) statistical modeling and 2) pattern comparison.

Examining the control charts in Fig 5, only 1 outlier of total 132 (0.75%) subgroups of the first 4 AR coefficients of the selected strain channel is detected. However slight downward tendency of the features is noticeable. Again in the X bar Chart for the first 4 AR coefficients of the accelerometer data (in x and y axes) there is 1 outlier of total 124 of subgroups, and in the first 4 AR coefficients of accelerometer data in z axis there are 3 outliers of total 108 (2.78%) subgroups. In the work done by Sohn et al. (2000) on a concrete column in a laboratory environment, it was observed that at very mild damage state, statistical modeling showed 6.25% and significant damage 29.17% outliers of total subgroups. Comparing their result, the structure considered to be quite healthy (as expected for a bridge of this age).
Fig. 5: Outlier analysis of the first, second, third and fourth AR Coefficients of Strain readings of S_1_1_C1 respectively. Pool size=132, Subgroup size = 4.

Fig. 6: R-values of 27 data blocks of Strain S_1_1_C1 at Steady State Condition

The goodness of fit of AR model of reference data block to others as indicated by the R-value (Residual error index or coefficient of determination) can be also be used for structural health assessment over time. It is logical that R-values over time of a particular strain or accelerometer should be steady and stay close to base line of 1. Upwards trend over time indicates no damage or degradation. In the present study, the R-values for 27 monthly blocks of S_1_1_C1 at steady state show no degradation. Same characteristic is noticed in 55 blocks corresponding to live loaded. Sometimes there are noticeable peak values in these blocks.
There might be two causes of this: (1) the vehicle passing over the particular sensor is such that it produces critically high strain at the location than the rest of strain gauge locations that its pattern deviates from the normal state; and (2) There is another vehicle nearby and the combined effect produces a non-regular pattern.

CONCLUSIONS

The results of the application of Damage Index Method and Matrix Update Method on the Crowchild Bridge indicate that both methods are fairly accurate in identification of the simulated damage when there is no or negligible amount of measurement noise in the frequencies and mode shapes. However, small amount error in frequency and mode shape measurement may affect the accuracy of the damage detection. Structural damage detection by statistical pattern recognition methods has been applied on the Portage Creek Bridge, Canada. From the source database, some measurement types have been selected for the experimentation. The AR process has been applied from derived data blocks to extract the AR coefficients which are then statistically modeled for damage classification by X-bars. From the X-bars of strain and vibration readings, percentages of outliers found are not so high to indicate any damage in the structure or prominent structural degradation, if any. However, a few cases suggest that the structure may be getting slightly degraded towards the end of the period considered, though it is still adequately safe. As an alternative approach to pattern comparison, fitting of the reference models to test blocks has been performed. Computed $R$-values that represent the goodness of fit do not show any trend or consistent discrepancies to indicate any damage in the structure. The sensors are also found to function properly by this method. Considering the age of the bridge there should not be any degradation too. The results of this method and the previous one have confirmed the structural health of the bridge to be at good state.

ACKNOWLEDGEMENT

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COMPACT PIEZOWORM ACTUATOR FOR MR-GUIDED SURGICAL NEEDLE PROCEDURES

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ABSTRACT

A novel design of a piezoworm-type actuator is being developed which can operate safely within the MRI (Magnetic Resonance Imaging) environment. Applications of such an actuator include medical needle procedures such as biopsy and brachytherapy. The device has two degrees of freedom, linear and rotary. This combination of motion allows for accurate steering of the needle during the medical procedure. MR (Magnetic Resonance) compatibility in this paper is considered from the point of view of the forces produced in the material due to the strong magnetic static field. Also, the effects RF (Radio Frequency) and magnetic gradient field are addressed.

Keywords: piezoelectric, piezoworm, MRI compatible actuator.

INTRODUCTION

Various medical procedures require accurate navigation of surgical instruments (e.g., therapy and biopsy needles) to reach its intended target, and this becomes more essential as the target gets smaller. Research by [1] shows that needle rotation about its insertion axis can reduce needle deflection and the target displacement. This rotation method leads to insertions with higher accuracy though increasing the precision of diagnosis [1].

Common imaging methods for guided interventions include X-ray fluoroscopy, computed tomography (CT), ultrasound (US), and magnetic resonance imaging (MRI). MRI stands out...
in its ability to produce high-quality images of the human body and assists in visualization of medical interventions [2]. These advantages make MRI one of the most versatile diagnostic tools in medicine and in areas such as image-guided interventions (IGI).

Developing an actuator that linearly navigates and rotates a needle while is capable of working safely in MRI environment is a challenge as there are severe restrictions on devices used in MR scanners. The MR environment involves extreme magnetic fields that can reach up to 7 Tesla [3], and it is very sensitive to noise emission. Therefore regular electromagnetic actuators and ferromagnetic materials are prohibited near the scanner for safety and operational reasons.

MRI scanners apply radio frequency (RF) pulses. These pulses, in addition to the magnetic field gradients, can heat the actuator constructed from conductive elements [2]. Furthermore, space restriction inside the MR scanner bore limits the size for devices operating within. As a result, the devices should be compact, and MR compatible material has to be employed in the actuator construction.

This paper presents a novel piezoworm actuator design that is capable of linear and rotary motion. Section 2 describes general MR safety and compatibility in different zones. Section 3 describes the concept of developing the actuator. Section 4 addresses the performance requirements and constraints. Section 5 discusses the MR environment effects on the actuator material and Section 6 summarizes the conclusion.

MR SAFETY AND COMPATIBILITY

MR safety can be defined as the safe performance of a device without any added risk to human or equipment when placed in MR environment though it may deteriorate the image quality [4]. A device is said to be “MR-compatible” [5] when it does not impose any safety hazard, maintains the image quality, and performs its specific intended function in the MR environment. When designing MR-compatible devices further classifications of compatibility zones have to be defined [5].

A device operating within the human targeted body part, and stays in contact with it throughout the scanning process is called a Zone 1 compatible device. When operating in the bore but not within targeted part and remains inside during the scanning process, though it could affect the image quality the device is called a Zone 2 compatible device. When operating on the movable bench but not within the bore and can be removed or not used during MR scanning the device is called a Zone 3 compatible device. Finally devices operating outside the MR scanner bore and the movable bench is said to be Zone 4 compatible device.

To realize a safe MR-compatible device operating in zone 1 or 2, interaction between the device and the MR environment has to be investigated. The main concern of this paper is the effect of the strong static magnetic field on the device material and ways to alter this effect through a choice of suitable materials. Also, other factors that affect the safety of the device (i.e. RF pulses and gradient field heating effects) are investigated.

The actuator and the enclosing structure should be a Zone 1 compatible device. The supporting structure connecting to the external robot can be Zone 1 or Zone 2 compatible device. If the operation of the actuator affects the image quality during MR scanning Zone 3 compatibility could be considered.
The control unit and power supply of the actuator will be placed outside the vicinity of the scanner. These parts could be Zone 4 compatible devices, but it is essential to shield and filter all the transmission wires (electrical signals) coming from/to the control unit to the actuator, so at least Zone 2 compatibility should be used.

**ACTUATOR CONCEPT**

Electromagnetic actuators generally are not compatible with the MRI environment as they produce strong magnetic fields that interfere with the operation of the scanner. Also the actuator can malfunction due to the high static magnetic field. Hence, alternative types of actuator have been proposed for MR compatible applications such as hydraulic, pneumatic, piezoelectric and manual actuation methods but drawbacks exist.

Hydraulic actuation is a stiff MR-compatible method of actuation as introduced by [6] but it is difficult to control and requires heavy infrastructure. Pneumatic methods of actuation is low cost, compact in size, and possesses a high power-to-weight ratio as in the case of PneuStep introduced in [7], but limited stiffness and performance degradation with time are some of the disadvantages. One of the commonly utilized technologies for MR-compatible actuators has been the ultrasonic piezoelectric motors. If remote actuation is considered as in [8] this will drastically decrease the stiffness of the system, as remote systems suffer from joint flexibility, backlash, and friction. Manual actuation is a simple method of actuation as in the MRI guided needle placements introduced in [9]. The major drawback in the case is large device size.

Several piezoworms have been developed for non MR-compatible environment as presented in [10]. The novel design presented in this paper is MR-compatible that has two degrees of freedom without using two separate actuators sets. The actuator is capable of performing linear and rotary motion either separately or simultaneously. The linear motion is based on the piezoworm principle. The use of piezoelectric stacks which is MR-compatible offers a high weight to force ratio and makes the device compact in size with high output force. The linear motion is achieved through sequential clamping to step its way to the commanded position [10]. This motion either activated forward or backward for full insertion or retraction of the surgical instrument. The step size and speed is controlled accordingly by means of the changing amplitude and/or frequency of the control signal to achieve the desired performance.

The configuration of the linear section of the developed actuator is shown in Fig.1. This design makes use of special complementary clamp configuration introduced by [11]. The advantage of using such configuration is emphasized in reducing the number of amplifiers used to drive the piezoworm assembly which in turn lower the cost of the system.
The complementary clamps have an opposite clamping action as shown in Fig.2 where one of the clamps is normally open and the other is normally closed. The added value for such configuration is that the normally open clamp will always keep the actuator position despite any power failure.

Fig.1: Configuration of the developed linear inchworm actuator.

Fig.2: Complementary clamps  a) Normally closed Clamp  b) Normally open clamp[6].
The rotary motion is also based on piezoworm configuration as shown in Fig. 3. The configuration consists of two complementary clamps that are activated by one amplifier output signal. The other output of the amplifier activates the extending actuator which moves the rotor away from the rotor clamp when it extends and presses the rotor against a moving arm when it relaxes. By activating the rotor clamp when the extending actuator is relaxed the rotor will incrementally move around its axis with a small arc. By activating the extending actuator the rotor will moving away from the arm. Now by deactivating the first clamp the arm will return back to its original position without affecting the rotor position. By repeating the previous sequence incremental rotary motion is achieved as illustrated in Fig. 4.

Fig. 3: Components of the rotary inchworm actuator.

The novelty of the actuator arises from its capability of achieving linear and rotary motion separately and simultaneously using one set of amplifiers which reduces the cost of the device and increases the precision of needle insertion. In addition the compactness in size allows it to operate deep inside the scanner bore.
PERFORMANCE REQUIREMENTS

Constraints and performance requirements were established by evaluating the device operating conditions. Major factors involved in this process are magnetism, restricted area inside the scanner bore, stiffness and the required accuracy and speed.

Three-dimensional model of actuator was constructed to assist in visualizing the conceptual design and the FEA (finite element analysis) model was built. FEA was employed in developing, simulating, and validating the proposed design. FEA was used to model the stiffness and stress of components and assemblies, especially in the interface regions between the different materials to optimize the coupling between the piezoelectric and structural components.

Size is an important factor in the design of the actuator. The bore diameter of the scanner is around 60 cm and with a patient fully placed inside. The actuator size was set to 30 mm of length, 30 mm width, and a height of 130 mm to generate a max needle stroke of 80 mm. These values were optimized for better the manoeuvrability of the device inside the scanner restricted space.

Force needed to pierce soft skin and the needles’ speeds are related as described in [12]. Fixing the speed to 5mm/s (max speed for the actuator) the related force value can be determined. Accordingly, a PI P-882.10 piezostack was selected to fulfill the speed and force requirements.

Minimizing the use of ferromagnetic material was the main constraint as substantial loss in MR signal occurs due to the effect of these materials on the homogeneity of the magnetic field. MRI scanners apply radio frequency (RF) pulses. These pulses in addition to the magnetic field gradients, can heat the actuator elements constructed from conductive materials, which in turn will transfer this heat to attached tool, and becomes an additional risk to the patient as will be discussed in the next section.
MR-COMPATIBLE MATERIAL CONSIDERATIONS

The MR system consists of a main magnet that produces a strong and constant magnetic field, coils that transmits and receives RF (radiofrequency) pulses, which excites and detects the MR signal and finally magnetic field gradients which spatially localize the MR signals [2].

Static Magnetic Field

The constant magnetic field produces forces in ferromagnetic materials when they are close to the magnet. Non-ferrous metals, plastics, and composite materials do not experience these forces as they have small values of magnetic susceptibility. But due strength of material and structural stiffness needed for this actuator plastics and composite materials were excluded from the material choice. Austenitic stainless steels (300 series) were considered as a suitable MR-compatible material as its magnetic susceptibility range from $10^{-1}$ to $10^{-3}$ [5]. Also, aluminum and beryllium copper were considered as their magnetic susceptibility of $20.7 \times 10^{-6}$ and $4 \times 10^{-6}$ respectively and are well used in MR environment [5].

Aluminum was selected for constructing the structural elements (rails) shown in Fig.5, as aluminum has a low magnetic susceptibility. But as its surface hardness is less than 150 (HB) [5], the contacting surfaces have to be coated with a thin film of ceramic to reduce surface wear with moving parts. YHD50 is a non-standardized stainless steel was considered also for constructing the rails as it possess magnetic susceptibility $1900 \times 10^{-6}$ and hardness number of 420 (HB) [5]. The final choice of the rail material will be made based on the least image distortion and the least heat generated after experiments are performed with representative sample. For moving parts i.e. clamps and the extender beryllium copper was selected as it can withstands high fatigue stress due to repeated cyclic load in these components.

![Diagram of the actuator and frame parameters](image)

**Fig. 5**: a) Overview of the actuator. b) Frame parameters
GRADIENT AND RF FIELDS

RF radiation can produce heating effect in electrical conductive materials. According to [13] there are three mechanisms by which heating can be caused by RF radiation: heating from eddy currents, heating from induction loops and heating by resonating RF waves along the conductor. In the first two mechanisms heat is produced instantaneously and it rarely exceeds a few degrees Celsius (no energy storage occurs) [13]. The final mechanism can produces higher temperatures if there is a sufficient conductor length (energy storage exists).

The change in magnetic flux in the case of direct gradient switching induces eddy currents in the actuator. The resistance of the conducting material converts the electric energy into thermal energy as in [15]. The dissipated power can be estimated analytically to roughly estimate the induced temperature as follows:

Considering the frame in Fig. 5b the frame side length is l, A is the cross section of the conductor and Ax is the area perpendicular to the frame. Electric resistance of the loop is given by (1).

\[ R = \frac{\rho L}{A} \]  

(1)

where \( \rho \) is the specific ohmic resistance  
\( L \) is the frame perimeter \( L = 4l \)  
\( A \) = cross sectional area of the conductor

The voltage V induced by changing magnetic flux is:

\[ V = N \frac{d\phi}{dt} \]  

(2)

where \( N \) is the number of turns (N=1)  
\( d\phi \) = magnetic flux

Since \( \phi = B \cdot A_x \), the induced voltage can be written as:

\[ V = N A_x \frac{dB}{dt} \]  

(3)

The dissipated power in the loop:

\[ P = \frac{V^2}{R} \]  

(4)

Dissipated power is in the form of heat, the P can be written as follows:

\[ P = h a (T_i - T_r) \]  

(5)

where \( h \) is the coefficient of heat transfer due to convection
a is the surface area of heat transfer
Ti is the induced temperature and Tr is the room temperature

Using these formulas with an average slew rate \((dB/dt)\) of 100 T/s, an average coefficient of convective heat transfer \(h = 55 \, \text{W/m}^2\cdot\text{K}\) and the physical parameters listed in Table 1, the estimated temperature rise in the frame was 20 °C. This is to be validated experimentally in future prototype tests.

<table>
<thead>
<tr>
<th>(dB/dt)</th>
<th>100 T/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>(l)</td>
<td>0.030 m</td>
</tr>
<tr>
<td>(A)</td>
<td>0.000009 m²</td>
</tr>
<tr>
<td>(\rho)</td>
<td>2.73x10⁻⁸ Ω·m</td>
</tr>
<tr>
<td>(N)</td>
<td>1</td>
</tr>
<tr>
<td>(h)</td>
<td>55 W/m²·K</td>
</tr>
<tr>
<td>(a)</td>
<td>0.0195 m²</td>
</tr>
<tr>
<td>(A_x)</td>
<td>0.0009 m²</td>
</tr>
</tbody>
</table>

**Table 1:** Parameters used to calculate the frames’ induced temperature.

**CONCLUSION**

A design of a compact actuator that is capable linear and rotary motion was achieved. The actuator utilizes one amplifier for controlling both types of motions separately and/or simultaneously which makes it practical and less expensive than other devices. The combination of linear and rotary motion makes it ideal for precise navigation of needle insertion procedures. It has a high force to size ratio and packs up to 10N of push force and speeds up to 5mm/s which can easily pierce through soft tissues. The choice of nonmagnetic material for the actuator component makes a good candidate for MRI compatibility but further study on the effect of MR environment will be conducted with the actuator to ensure it will perform properly and safely in the MRI bore.

**REFERENCES**

MECHANICAL CHARACTERIZATION OF POROUS MEMBRANE CORE MORPHOLOGIES FOR CONDUCTIVE POLYMER TRILAYER ACTUATORS

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ABSTRACT

Multilayer electroactive polymer actuators consisting of polypyrrole films electopolymerized on a passive polymer membrane core have been harnessed as a source of simple actuation. As an integral component of the actuator, the membrane plays a vital role in the transport of ionic species and largely dictates the stiffness of the layered configuration, yet in past studies the specification of the membrane has remained largely arbitrary. In this investigation we review the pertinent conductive polymer models to identify the properties of the membrane that play a vital role in the actuation response of polypyrrole-based trilayer bending actuators. Candidate materials with distinctly varied microcellular morphologies are identified and include polyvinylidene difluoride, nylon, and nitrocellulose. The quasi-static stress-strain response and the frequency dependent viscoelastic nature of the candidates is then evaluated. On the basis of mechanical properties these results indicate that polyvinylidene difluoride membranes are superior to the other candidates for application as trilayer actuator cores.

Keywords: Conductive polymer, polypyrrole, trilayer actuator, porous membrane.

INTRODUCTION

Conductive polymer (CP) actuators possess many favourable advantages over conventional mechanical actuator technologies [1]. The range of motion of these actuators has been amplified by fabricating layered configurations which bend upon the application of an external electric field due to the net motion of ions. Trilayer actuators (as illustrated in Figure 1) are typically employed when the CP actuator is required to operate in the absence of an electrolytic environment. In this arrangement encapsulation of the device is achieved
through the electropolymereization of a conductive polymer such as polypyrrole onto opposing sides of a porous membrane core. The core thus serves as an electrolyte reservoir.

The membrane material and its corresponding pore structure influence the transport properties of the ionic species within the electrolyte and also heavily dictate the overall stiffness of the composite; however existing studies involving CP trilayers and their application have ignored these contributions. Thus the specification of the membrane material and structure remains a subjective process. In this study we review the pertinent diffusion and electrochemomechanical models to identify the characteristics of the membrane that play a vital role in the actuation response of polypyrrole-based trilayer bending actuators. Candidate membrane materials with distinctly varied microcellular morphologies are then identified and characterized. Finally, the quasi-static stress-strain response and the frequency dependant viscoelastic nature of the candidates are evaluated to quantitatively determine the most suitable core membrane candidate.

![Fig. 1](image)

**Fig. 1** The trilayer bending actuator consists of a porous membrane core with a conductive polymer such as polypyrrole deposited on opposing faces. The actuator is shown (a) at rest and (b) in the deformed state.

**THEORETICAL BACKGROUND**

**Underlying Actuation Mechanism**

The mechanism responsible for the actuation behaviour of conductive polymers is complex and is a result of several interrelated phenomena [2]. When an electric potential is applied to the polymer the primary source of actuation is attributed to dilation of the polymer upon the incorporation of ions from the neighbouring electrolyte (or conversely the polymer may contract upon expulsion of ions). On a molecular level, the ions interact with the polymer backbone in order to maintain charge neutrality upon modification in the latter’s oxidation state. In a trilayer actuator one CP layer swells while the opposite CP layer
contracts. This induced differential strain results in a macroscopic bending effect which may be harnessed to perform mechanical work.

Prospective Applications

Conductive polymer actuators have been successfully demonstrated in several prototypes. More recent innovations include active catheters for biomedical use [3, 4] and fins for aquatic robots [5]. As response times for conductive polymer actuators are diffusion dependant, miniaturization is expected to greatly improve actuation bandwidth beyond currently achievable levels [6]. Nanoscale lithography techniques are currently being explored to develop high-speed CP actuator technology that will be beneficial for a broad range of applications.

Conductive Polymer Synthesis and Trilayer Fabrication

Polymerization of the pyrrole monomer is achieved through a multi-step electrochemical oxidation reaction [7, 8]. Several alternative growth mechanisms have been proposed, and in all cases the reaction product is highly dependant on parameters such as solvent choice, temperature, and electrolyte dopant [8]. The trilayer arrangement is fabricated using a three electrode electrochemical cell as shown in Figure 2 that is connected to a potentiostat. This apparatus permits the simultaneous coating of both sides of a centrally fixed membrane core under identical environmental conditions. The membranes are sputter coated with a thin layer of platinum prior to installation in the cell to facilitate conduction.

![Fig. 2](image.png)

**Fig. 2** A polymerization vessel with opposing counter electrodes is utilized to simultaneously electropolymerize both faces of a porous membrane core.

Modeling

The interdisciplinary nature of these actuators consisting of electrical, mechanical, and chemical phenomena has resulted in the suggestion of analytical models with varying foundations. Two such approaches include diffusion based models such as those proposed by
Wang, Shapiro, and Smela [9, 10] and electrochemomechanical models such as those proposed by Alici and Fang [11-13].

![Diagram of polymer-electrolyte interface geometry](image)

**Fig. 3** (a) The idealized polymer-electrolyte interface geometry for the one dimensional charge transport model proposed by Wang et al. can be extended to (b) the trilayer case.

As the actuation mechanism ultimately depends on ion motion, models based on diffusion and migration of charge carriers have been established for a single layer of conductive polymer in electrolyte as shown in Figure 3(a) [10]. Extending this scenario to the trilayer configuration results in the modified geometry indicated in Figure 3(b). The basis for this modeling approach is to apply the Nernst-Planck equation to each mobile charge carrier in the electrolyte and conductive polymer to form a system of partial differential equations. These equations are then coupled together using Poisson’s equation which determines the electric potential for the instantaneous charge distribution. A summary of the governing equations is provided in Table 1 and the model parameters are defined in Table 3(a).

Solutions of the system have been determined using finite element analysis when the boundary (Dirichlet) and flux (Neumann) conditions outlined in Table 2 are enforced. In the trilayer case, a modification to the diffusion coefficient for the membrane core is required to account for the open area and tortuosity inherent to the porous structure [7]. The model parameters indicate the relative importance of the diffusivity and mobility of the mobile species in each layer. A systematic investigation of these parameters will be conducted in the future to assess the most favourable combination of membrane material, electrolyte, and solvent.

The electrochemomechanical models are generally extensions of the Diffuse Elastic Metal (DEM) model proposed by J. Madden [6, 14] in which an abstraction of the ionic response is achieved through an equivalent electrical admittance circuit. This abstraction is then coupled with the principle that the polymer dilation is a function of the ionic charge transferred to it, and thus an empirically determined strain-to-charge ratio is necessary. Figure 4 indicates that the functional elements of the model proposed by Fang et al. include an admittance module, an electromechanical coupling module, and a mechanical output module. The corresponding parameter definitions are summarized in Table 3(b). This model
outputs the free tip displacement of a trilayer actuator when an input potential is prescribed. The $C_m$ coefficient captures material effects such as viscoelasticity. The viscoelastic nature of the actuator is not negligible, and a linear viscoelastic model with empirically tuned parameters has been demonstrated with limited success [11]. Thus we are motivated to quantify the viscoelastic behaviour of membrane materials in order to improve modeling accuracy and objectively determine the most suitable membrane material. This formulation also indicates that the actuator tip deflection is inversely proportional to the membrane stiffness.

Table 1 Charge diffusion and migration as modeled by the Wang diffusion model.

<table>
<thead>
<tr>
<th>Region</th>
<th>Phenomenon</th>
<th>Governing Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polymer:</td>
<td>Cation diffusion</td>
<td>$\frac{\partial C}{\partial t} = -\nabla \cdot (-D_C \nabla C - \mu_C C \nabla \phi)$</td>
</tr>
<tr>
<td></td>
<td>Hole diffusion</td>
<td>$\frac{\partial H}{\partial t} = -\nabla \cdot (-D_H \nabla H - \mu_H H \nabla \phi)$</td>
</tr>
<tr>
<td></td>
<td>Poisson’s equation</td>
<td>$\varepsilon_0 \nabla \cdot (\varepsilon \nabla \phi) = Q = C + H - N$</td>
</tr>
<tr>
<td>Electrolyte:</td>
<td>Cation diffusion</td>
<td>$\frac{\partial C_e}{\partial t} = -\nabla \cdot (-D_{C_e} \nabla C - \mu_{C_e} C \nabla \phi)$</td>
</tr>
<tr>
<td></td>
<td>Anion diffusion</td>
<td>$\frac{\partial A}{\partial t} = -\nabla \cdot (-D_A \nabla A - \mu_A A \nabla \phi)$</td>
</tr>
<tr>
<td></td>
<td>Poisson’s equation</td>
<td>$\varepsilon_0 \nabla \cdot (\varepsilon \nabla \phi) = Q = C_e - A$</td>
</tr>
</tbody>
</table>

Table 2 Boundary conditions corresponding to the 1D geometry illustrated in Figure 3(a).

<table>
<thead>
<tr>
<th>Location</th>
<th>Phenomenon</th>
<th>Boundary Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference electrode:</td>
<td>Initial concentration</td>
<td>$C_e = A = 0.03C_{\text{max}}$</td>
</tr>
<tr>
<td></td>
<td>Reference voltage</td>
<td>$\phi = 0$</td>
</tr>
<tr>
<td>Electrolyte-polymer interface:</td>
<td>Zero anion flux</td>
<td>$\vec{J}_A \cdot \hat{n} = 0$ (electrolyte side)</td>
</tr>
<tr>
<td></td>
<td>Zero hole flux</td>
<td>$\vec{J}_H \cdot \hat{n} = 0$ (polymer side)</td>
</tr>
<tr>
<td>Working electrode:</td>
<td>Zero cation flux</td>
<td>$\vec{J}_C \cdot \hat{n} = 0$</td>
</tr>
<tr>
<td></td>
<td>Hole flux</td>
<td>$\vec{J}_H = \mu CE$</td>
</tr>
<tr>
<td></td>
<td>Applied voltage</td>
<td>$\phi = -V$</td>
</tr>
</tbody>
</table>
Table 3 Definition of nomenclature employed for each of the primary modeling approaches.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Definition</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C$</td>
<td>cation concentration</td>
<td>mol/cm$^3$</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
<td>s</td>
</tr>
<tr>
<td>$D$</td>
<td>diffusion coefficient</td>
<td>cm$^2$/s</td>
</tr>
<tr>
<td>$\mu$</td>
<td>mobility</td>
<td>cm$^2$/V s</td>
</tr>
<tr>
<td>$\phi$</td>
<td>electric potential</td>
<td>V</td>
</tr>
<tr>
<td>$H$</td>
<td>hole concentration</td>
<td>mol/cm$^3$</td>
</tr>
<tr>
<td>$\varepsilon_0$</td>
<td>vacuum permittivity</td>
<td>$8.85 \times 10^{-12}$ F/m</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>dielectric constant</td>
<td>–</td>
</tr>
<tr>
<td>$Q$</td>
<td>net charge density</td>
<td>C/cm$^3$</td>
</tr>
<tr>
<td>$N$</td>
<td>electron concentration</td>
<td>mol/cm$^3$</td>
</tr>
<tr>
<td>$J$</td>
<td>species flux</td>
<td>mol/(cm$^2$ s)</td>
</tr>
<tr>
<td>$L$</td>
<td>layer thickness</td>
<td>m</td>
</tr>
<tr>
<td>$A$</td>
<td>anion concentration</td>
<td>mol/cm$^3$</td>
</tr>
</tbody>
</table>

Subscripts:

- $C$: cation
- $H$: hole
- $e$: electron
- $p$: polymer
- $m$: membrane

(b) Electrochemomechanical model

<table>
<thead>
<tr>
<th>Designation</th>
<th>Definition</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$s$</td>
<td>complex variable</td>
<td>–</td>
</tr>
<tr>
<td>$Y$</td>
<td>admittance</td>
<td>$\Omega^{-1}$</td>
</tr>
<tr>
<td>$I$</td>
<td>electric current</td>
<td>A</td>
</tr>
<tr>
<td>$V$</td>
<td>electric potential</td>
<td>V</td>
</tr>
<tr>
<td>$D$</td>
<td>diffusion coefficient of ion in polymer</td>
<td>cm$^2$/s</td>
</tr>
<tr>
<td>$R$</td>
<td>electrolyte/contact resistance</td>
<td>$\Omega$</td>
</tr>
<tr>
<td>$C$</td>
<td>double layer capacitance</td>
<td>F</td>
</tr>
<tr>
<td>$\delta$</td>
<td>double layer thickness</td>
<td>m</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>induced stress</td>
<td>Pa</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>strain-to-charge ratio</td>
<td>m$^3$/C</td>
</tr>
<tr>
<td>$E$</td>
<td>elastic modulus</td>
<td>Pa</td>
</tr>
<tr>
<td>$W$</td>
<td>conductive polymer width</td>
<td>m</td>
</tr>
<tr>
<td>$L$</td>
<td>conductive polymer length</td>
<td>m</td>
</tr>
<tr>
<td>$h$</td>
<td>conductive polymer layer thickness (each)</td>
<td>m</td>
</tr>
<tr>
<td>$C_m$</td>
<td>material and geometry coefficient</td>
<td>(m C)$^{-1}$</td>
</tr>
<tr>
<td>$y$</td>
<td>actuator tip displacement</td>
<td>m</td>
</tr>
</tbody>
</table>
The complete electromechanical model proposed by Fang et al. is comprised of three functional components.

**EXPERIMENTAL METHODOLOGY AND RESULTS**

**Morphology of Candidate Membrane Materials**

Several candidate membrane materials have been selected based on the diversity of their properties and microstructure. Characterization of the mechanical properties of membranes will objectively indicate which membrane is best suited as a trilayer core based on their static and viscoelastic response (the identification of diffusion characteristics is ongoing). The candidates are commercially available and are all rated for a retention size of 0.45 µm to facilitate comparison. SEM images of the membranes are shown in Figure 5 as follows: (a) nitrocellulose (Hybond ECL, RPN203D, GE Healthcare), (b) polyvinylidene difluoride (PVDF) (Westran CS, #10485288, Whatman), and (c) nylon (RPN303B, GE Healthcare). These images depict the variety of microstructures available despite their equivalent retention size. The nitrocellulose and nylon membranes are fibrous in nature and their pore morphologies consist of a wide distribution of pore sizes. In contrast, the PVDF morphology is relatively homogeneous exhibiting an average cell density of $7 \times 10^{11}$ cells per cubic
centimeter. A separate study to investigate how the nature of these morphologies affects the ion mobility must be undertaken; however it is apparent that both the constituent material itself as well as its associated pore structure will impact the actuator performance.

(a) Nitrocellulose

(b) PVDF

(c) Nylon

Fig. 5 SEM images reveal the distinct porous morphologies of the (a) nitrocellulose, (b) PVDF, and (c) nylon membranes.

Fig. 6 The stress-strain response of the membranes illustrates their diverse mechanical behaviour. The curves depict a single representative sample of each material.

Quasi-static response of candidate membrane materials

The stress-strain behaviour was determined using a TA Instruments Q800 dynamic mechanical analyzer (DMA) in tensile stress-strain mode at ambient temperature. A strain rate of 1000 µm/min was prescribed to obtain the elastic response. As indicated in Figure 6, the nylon membrane exhibits the highest strength and stiffness ($E_{\text{nylon}} = 1.90$ GPa), followed by the PVDF membrane ($E_{\text{PVDF}} = 1.69$ GPa), and finally although nitrocellulose was nearly as stiff as the PVDF ($E_{\text{nitrocellulose}} = 1.58$ GPa), it was found to be extremely brittle. Hence
these results eliminate nitrocellulose as a practical candidate material since the limited failure strain may result in localized failure of the membrane upon bending.

**Viscoelastic response of candidate membrane materials**

Determination of the viscoelastic properties of the PVDF and nylon membranes was performed using the TA Instruments Q800 dynamic mechanical analyzer (DMA) in tensile mode at ambient temperature. An oscillating tensile strain of 1% was prescribed over a frequency range of 0.1 Hz to 10 Hz under a 1 mN preload. The results indicated in Figure 7 confirm the increased stiffness of the nylon membrane with respect to the PVDF. Furthermore, the tan δ measurements indicate that the PVDF has a larger degree of damping (especially in the low frequency range). These results indicate that PVDF is the best candidate if tip-deflection is to be maximized, but the increased tip deflection comes at the expense of mechanical efficiency since the PVDF dissipates more energy than the nylon.

![Graphs](image_url)

**Fig. 7** Plots of the storage modulus (left axis), loss modulus (outer right axis), and tan δ (inner right axis) convey the viscoelastic nature of the (a) PVDF and (b) nylon polymer membranes.

**CONCLUSIONS**

This paper reviewed models pertaining to the actuation behaviour of conductive polymers, and it was shown that ion diffusion rates, membrane core stiffness, and membrane core viscoelasticity were particularly vital parameters in the optimization of the response. This review elucidated the importance of a deliberate membrane selection process. A modification to an existing diffusion based model was also suggested to extend the model to the trilayer actuator configuration. The mechanical properties of several alternative core materials were subsequently characterized based on their porous morphology, quasi-static stress strain response, and viscoelastic response. From these results it was determined that PVDF is the most favorable candidate material if tip displacement is to be maximized.

**ACKNOWLEDGEMENTS**

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REFERENCES

DEVELOPMENT OF A SMART MATERIAL BASED MICROVASCULAR CLAMP

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ABSTRACT

There is a need for controllable microvascular clamps in the surgical industry. Currently used microvascular clamps are designed in a range of sizes to make them suitable for use with various size vessels and clamping force requirements. In some procedures, a clamp is required to remain inside the body after a surgical procedure. However, the clamp typically needs to be released at a later time. A controllable microvascular clamp has the potential to accomplish these goals while minimizing the invasiveness of the post-operative releasing process. A Ni-Ti alloy, commercially referred to as Nitinol, is being evaluated to ascertain whether it can be incorporated into a design which can exert a clamping force in the range of 50 to 150 g on a vessel. This paper will relate the features and testing of the first design of such a microvascular clamp.

Keywords: Microvascular clamp, Shape memory alloy, Nitinol clip
INTRODUCTION

Microvascular clamps are used in thoracic and laparoscopic surgery. In certain circumstances, the clamp is required to stay inside the body so that it can be released at a later time. A procedure that may require this is endoscopic thoracic sympathetic clamping, or ETC [1]. Currently, the method to disengage microvascular clamps entails the use of a string connected to the clamp. To avoid this invasive procedure, or the need for a second, albeit simpler, follow-up procedure, this research proposes the utilization of a smart material based clamp which can be released more conveniently.

In certain surgical procedures, stopping blood flow in a vessel is critical. However, it is imperative that the clamping device not damage the vessel through the application of an excessively large clamping force. This is particularly important in laparoscopic or thoracic procedures in which the vessel must be returned to functionality without incurring damage from the clamp. The controllable clamps based on smart materials will be beneficial because they can easily stop and then release blood flow in vessels and/or the function of nerves during and after a surgical procedure. In the event that the clamp needs to be released after the wound has been sutured, a minimally invasive procedure could be used to trigger a shape memory alloy based latch on the clamp. This would be more reliable than existing methods using strings.

In order to impart the desired functionality in the clamps, this project intends to use smart materials, specifically a shape memory alloy (SMA). Smart, or active, materials exhibit useful coupling between multiple physical domains, and their mechanical properties can be changed using external stimulus, such as electric fields and magnetic fields. In biomedical devices, particularly for microvascular clamps, an innovative material/component is desirable because conventional mechanical components, such as valves and gears make the device too big and complex. Large devices take up space in the body cavity, and the number of components should be limited to reduce failure rate and reduce the number of foreign implements inside the body. Smart materials bridge the gap between mechanisms and biocompatibility; they provide the mechanical means to stop the blood flow with minimally intrusive components.

Shape memory alloys are biocompatible, making them an excellent choice for medical devices. The biocompatibility of Nitinol, the selected shape memory alloy, can be enhanced through specific treatment methods, such as acidic passivation, to preclude tissue contamination. The biocompatibility of Nitinol wire is demonstrated by Nitinol stents which are implanted within the body [2]. Different activation temperatures are available.

This paper describes the design constraints and fabrication of the first prototype. It is intended to serve as a proof of concept and test bed for optimizing the design of the SMA based latch/release mechanism.
FABRICATION

Rather than preparing a design that was entirely based on the deformation and recovery characteristics of a SMA, it was decided that a hinged pin design utilizing a SMA only for a latching and release mechanism would afford the greatest flexibility in tuning for clamping forces and activation temperatures. While the prototype was machined from Al 7075-T6 for design evaluation, biocompatible versions could be prepared using cobalt steel or titanium. Fig. 1 demonstrates how a clamp is used during a surgical procedure. The leftmost box represents the clip as it was mechanically opened and placed around a vessel. The middle box represents the clip at body temperature. The box on the right demonstrates the removal of the clip by applying a cool temperature, roughly 20°C.

Fig. 1: Demonstration of vascular clip functionality [3].

Clamps are applied using forceps or another implementation device. In addition to the goal of designing a clip to occlude the vessel, some design considerations that should be taken into account are the insertion and removal of the clip and the interference of the clip with the surgical procedure. The clip must be removable from the vessel without an open surgery.

There are two key points to the design. There is a torsion spring supplying the opening force, and a Nitinol wire keeping the clamp closed. The prototype is pictured in Fig. 2 in the closed position. The Nitinol wire and top arm wire are secured in the device by tightening set screws inserted into the end face of the clamp. The SMA latches onto a rigid stainless steel pin similarly secured into the second leg of the clamp.

Fig. 2: First prototype in the closed position.
The prototype clamp is largely rectangular for ease of manufacturing. Other features were incorporated into the design that will maximize the effectiveness of the clamp. Instead of being uniformly rectangular, the width of the top arm steps out after the hinge. This gives the top and bottom arms the same width where the vessel is gripped. A few other geometries were machined into the prototype around the spring so that the top and bottom arms would not have a rough interface.

Shape memory alloys could be further incorporated into the design. This is demonstrated by the development of a pair of atraumatic haemostatic forceps [4]. In these forceps, the amount of force applied to the blood vessel can be controlled. This feature could potentially be incorporated into the next design to control the amount of force applied to the vessel, in addition to controlling the release of the clamp.

**EXPERIMENTATION**

The features of the clamp can be further optimized for mass fabrication and enhanced functionality. There are a few design changes that will be implemented into the next design iteration. First, the hole for the SMA wire or the wire it hooks on to would be positioned differently. Currently, they are aligned vertically, but this does not take into account the diameters of the SMA wire. The wire should be able to be positioned vertically and neatly hooked onto the wire on the top leg. It proved to be difficult to hook the wire on to the pin to close the clamp. One improvement would be to make the length of the Nitinol wire longer to make it easier to manipulate within the small network of nerves which are present during surgery. The SMA wire chosen had a diameter of 0.002 in. The strength of the wire is sufficient, but a thinner wire would be able to hook on to the wire in the top arm with less manipulation.

The preliminary setup for a procedure testing the release process is shown in Fig. 3. Since the Nitinol can be custom ordered with different activation temperatures, the response of the clamp can be tuned. However, it is then necessary to verify the actual activation temperature of the wire. This was accomplished by using a dynamic mechanical analyzer (DMA) in a temperature ramp mode while monitoring the force on the plate positioned just above the clamp. In addition to determining the release temperature, this test also verified the performance of the SMA latch. The test showed that the wire caught slightly on the pin. It released in a horizontal direction normal to the pin in addition to releasing vertically. The next design iteration will consider this behavior in the design. The pin will also be shortened to minimize binding. During testing, the prototype is not centered on the fixture as pictured in Fig. 3 so that the wire has clearance to release.
The testing was done on a dynamic mechanical analyzer (TA Instruments, model RSA3). This equipment is capable of measuring low forces in an environmentally controlled chamber. A 15 mm compression flattened fixture was used during the test. The temperature was ramped to 100°C at 10°C per minute. The release of the SMA latch is evidenced by a sharp increase in force at approximately 77°C. The transition temperature for the 0.002 in diameter Nitinol wire is specified to be 70°C.

**CONCLUSION**

The design of the microvascular clamp includes a torsion spring and latch mechanism incorporating a shape memory alloy. The results from testing have proven the concept that shape memory alloys can function in microvascular clamps. A design configuration utilizing
a spring and a shape memory alloy wire latch would be able to be used during surgery. The results have shown that the concept of using a shape memory alloy in a surgical clamp is possible. At this time the existing design will be improved for ease of use. The clamp will be smaller and the hook mechanism will be easier to manipulate using forceps during a surgery. A shape memory alloy can be ordered with a lower activation temperature for use in the body.

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POLYPYRROLE DRIVEN CATHETER

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ABSTRACT

An optical fiber 2D scanner and positioner intended for in-vivo imaging application is presented. The design is a catheter coated with conducting polymer actuators and equipped with an optical fiber. Conducting polymer actuators (here polypyrrole) enables both positioning of the catheter to the desired region to be visualized and 2D scanning of the optical fiber. The optical fiber is connected to an optical coherence tomography (OCT) imaging system which performs high resolution depth resolved imaging. OCT depth imaging combined with the 2D scanning of the fiber will result in a complete high resolution 3D image. This technique enables detection and monitoring of changes in tissues, which is useful for assessment of vascular disease, cancer tissue progression, and surgical guidance.

Initial prototypes of the active catheter have been constructed and tested. In these devices a commercial catheter is coated with polypyrrole, which is laser micromachined into electrodes. These polypyrrole electrodes are electrochemically activated, leading to bending of the catheter. The catheter is able to achieve a radius of curvature of 10 mm, which is similar to the curvature achieved in wire guided catheters used at present. The catheter scans at frequencies of up to 10 Hz, but lifetime is short at this frequency. The primary challenge is to demonstrate high speed with reasonable lifet ime. An electromechanical model of the active catheter suggests means of achieving high speed actuation and sufficient cycle life.

Keywords: Polypyrrole actuator, Catheter, Optical coherence tomography, In-vivo imaging.
INTRODUCTION

Catheterization is a common medical procedure, in which a hollow tube (i.e. the catheter) is inserted into body cavities to provide a channel for fluid passage or an entry for a medical device. Catheters as fluid channels may be used to drain urine from the urinary bladder and administrate the intravenous fluids and medication directly into the body. Catheters are also used to direct a medical tool to a particular part of the body for minimally invasive surgical procedures. In angioplasty, for instance, catheters are employed to guide a therapeutic device to open a blockage inside a blood vessel [1].

The conventional method of handling a catheter involves inserting it into the body passively by employing push/pull control mechanism that is outside the body, where a wire configured to be pushed or pulled along a longitudinal axis to bend the catheter tip [2]. Limitations of the current catheter and guidewire designs include long procedural times, and risk of lumen or vessel wall damage, both as a result of slow and inexact guidance. These issues become more critical when dealing with narrow and complex passages such as blood vessels of the brain and tertiary bronchi of the lung [1]. Therefore, advanced active catheter designs with controllable features are required in order to enhance the performance of these devices during minimally invasive medical procedures. Various active catheters driven by different methods have been suggested, however no active catheters are in wide spread use. Micromotors mounted on catheters had been reported for ultrasonography [3,4], but they feature a relatively large size (diameter of 1.9 mm [4]) and expensive fabrication processes. Shape memory alloy (SMA) and Shape Memory Polymers (SMP) actuators have been used for steerable endoscopes [5-11]. Although these types of actuators are potentially able to provide a large degree of bending, their slow response, high operating temperature and the possibility of electrical current leakage limit their applications [9]. Controllable catheters utilizing hydraulic mechanisms have also been developed, where the catheter position is controlled by varying the size of inflatable balloons mounted on its tip using electro-thermally controlled microvalves [2]. This mechanism is cumbersome and controlling the microvalves is slow; hence not suitable for many applications [1]. Ionic Polymer Composites (IPMC) actuator has been also suggested to design active catheters [12-14]. These actuators can generate large displacements at relatively low voltages (<10 V) and moderate speed; however, their manufacturing process is often relatively expensive and additional energy is usually consumed for holding the actuator in a position. Conducting polymer actuators have shown attractive properties, which make them promising to be employed extensively in active catheter applications. Some of the characteristics include low actuation voltage, ease of fabrication, relatively high strain, and biocompatibility.

In this paper a conducting polymer actuated catheter for minimally invasive intervention inside arteries is designed and demonstrated. The design is composed of two active catheters optimized for two distinct applications (shown in Figure 1). One active catheter, “scanner”, is responsible for scanning an optical fiber over a distance of 1 mm, with a relatively high speed to provide forward viewing images from an artery lesion. The second catheter, the “positioner”, is then actively bent to reach the target lesion and guide a wire to navigate through the lesion, while the first fiber optic mounted catheter is monitoring the procedure in real time. Real time forward viewing is done by a technique called optical coherence
tomography (OCT) which performs depth resolved imaging by sending wide band near infrared light into tissue and observing the backscattered light interferometrically [15]. OCT depth imaging combined with the conducting polymer based scanning catheter shown in Figure 1 (which provides planar 2D scanning of the optical fiber), will result in a complete high resolution 3D image. This technique can serve as an important diagnostic adjunct, enabling the detection and the monitoring of changes in tissues, therefore is useful for assessment of vascular disease and cancer tissue progression [15], and surgical guidance. Each active catheter is encapsulated within a biomaterial structure containing an ionic electrolyte. Polypyrrole electrodes coated on the catheters are used as the conducting polymer based active elements and electrochemically actuated inside the ionic electrolyte using voltage changes of <1 V per polymer electrode. A lens is embedded at the end of the encapsulation tube containing the imaging catheter, which can amplify the angular scanning range (see Figure 1).

**Fig.1**: Schematic of the final encapsulated device

**EXPERIMENTS**

**Mechanism of actuation**

The conducting polymer, in this work polypyrrole, is electronically conductive. It is also porous, enabling ion insertion. Application of a voltage to the polymer in an electrolyte alters the charging of the polymer. The electronic charging occurs throughout the volume of the polymer, and is balanced by the insertion of ions from the electrolyte. This charging process is associated with changes in volume, where increase in volume generally occurs when ions are inserted into the polymer matrix. Figure 2 illustrates the mechanism of the dimensional change as a result of the electrochemical activation. As shown in the figure a substrate (here a catheter) coated with conducting polymer electrodes on both sides is in an electrolyte solution containing mobile negative ions and large immobile positive ions. An alternating voltage is applied across the two polymer electrodes resulting in charging and discharging of the polymer electrodes. During charging the mobile negative ions enter the polymer from the electrolyte, therefore expand the structure. During discharging they exit the polymer to the electrolyte and contract it. Expansion on one side and contraction on the other side induces a stress gradient on the polymer/catheter interfaces and causes the whole structure to bend in one direction. Alternating expansion and contraction will result in the movement of the catheter in 2 directions inside the electrolyte solution. Coating the catheter with four polymer electrodes enables actuation in 2 dimensions.
The amount of dimensional change “strain” \( \varepsilon \), is proportional to the amount of ion insertion and equivalently the charge per unit volume, \( \rho \), via the relationship:

\[
\varepsilon = \alpha \rho
\]  

(1)

\( \alpha \) is an empirically determined strain to volumetric charge ratio [24-26]. The rate of actuation of polypyrrole is proportional to the rate of ion insertion, and hence to the current. The current can be limited by the both ionic and electronic resistivities, diffusion coefficients, and the capacitance of the electrodes. For a long device the total polymer resistance along the length of the catheter, \( R_{ppy} \), can significantly limit the rate [26], with the predicted time constant being \( \tau = R_{ppy} C = \frac{C_{v} l^2}{\sigma_{e}} \) [26], where \( \sigma_{e} \) is the electronic conductivity of the polymer, and \( C_{v} \) is the polymer volumetric capacitance. For a thick device the ionic resistance through the thickness, \( t_{p} \), is the rate limiting factor, with the time constant of \( \tau = \frac{t_{p}^2}{D} \) [26], where \( D \) is the effective diffusion coefficient. An electrochemical model was developed which uses the above factors to design optimum devices for both the positioner and the scanner catheters [21, 22, 23].

**Fabrication**

The catheter used in this experiment was a Micro Therapeutics Inc. (Irvine, CA) UltraflowTM HPC, with inner and outer diameters of 0.28 mm and 0.5 mm respectively. Polyppyrole deposition on the catheter was done in two steps. First a thin layer of polymer was deposited using electroless deposition by chemical polymerization of pyrrole monomer in the presence of oxidizing agent [16]. Then polyppyrole films are electrochemically grown from a solution of 0.06 M pyrrole monomer [www.aldrich.com] and 0.05 M tetraethylammonium hexafluorophosphate [www.aldrich.com] and 1 % vol distilled water in propylene carbonate, following the procedure of Yamaura [17]. Polyppyrole is deposited galvanostatically on to the catheter at the current density of 1.25 A/m² and at temperatures between -30 °C and -45 °C with a rate of ~1.3 μm/hr. The resulting polymer is in the doped or oxidized state with the doping level in as-grown polyppyrole of approximately one charge per three monomers [18]. Polymer coating was then divided into 4 electrodes using laser ablation (Figure 3) [19].

**Fig. 2.** Compressive and tensile stress upon insertion and removal of ions from the polypyrrole layer.
Experimental Catheter Actuation

According to the developed model a thickness of > 40 μm is required to achieve the large degree of bending (i.e. the requirement for the positioner catheter). The model, however, suggests that a polymer thicknesses of ~ 10 μm results in maximum tip displacement at high actuation rate (i.e. the requirement for scanner catheter). Therefore the positioner catheter was coated with 40 μm thick polypyrrole and the scanner catheter was coated with 10 μm thick polymer, and their performances were tested separately. Both catheters are actuated inside an aqueous solution of sodium hexafluorophosphate (NaPF₆) which has been previously shown to produce relatively large strains [27].

Positioner Catheter

The polymer coated positioner catheter was actuated inside an aqueous solution of NaPF₆ by applying a step potential of -0.8 to +0.8 V versus Ag/AgCl reference electrode and a bending radius of 9.76 mm was achieved in 30 seconds. This is similar to the curvature achieved in wire guided catheters used at present [22]. Figure 4 illustrates the positioner catheter bending.

The scanner catheter with 4 polymer electrodes was also tested in the same electrolyte (AQ-NaPF₆) by applying potential across the two facing polymer electrodes. Note that two potential sources are used (see Figure 5); one applies an alternating voltage across the two facing electrodes with relatively high frequency which causes the catheter to move back and forth in y direction relatively quickly; the other source simultaneously applies a potential with a slower frequency to the second polymer electrode pairs resulting in x direction actuation.
The primary demonstration of the scanner was performed with a speed of 0.1 Hz in x direction and 1 Hz in y direction. Figure 6 shows a view of the scanner tip at initial and final positions with the arrows showing the route of the scanning.

In order to achieve real time OCT imaging the scanner catheter requires scanning an optical fibre in 2 dimensions with high speed (10-30 Hz). The scanning speed of 10 Hz was achieved in one dimension by applying a high actuation potential (10 V), however the number of scanning cycles was limited due to short polymer lifetime at high actuation voltages. An electromechanical model was used to study the feasibility of achieving high speed actuation (10 Hz – 30 Hz) appropriate for OCT imaging. The detailed description of the model can be found in our previous work in [21]. According to our model, electrochemical characteristics of the conducting polymer such as electronic conductivity, ionic conductivity and electrochemical strain need to be improved to achieve the desired catheter scanning speed (i.e. 30 Hz) for real time imaging [21,23]. Table 1 illustrated the current values of these parameters and the model suggested values for obtaining 30 Hz actuation over a distance of 1mm. The catheter length of 35 mm was considered in this calculation (defined by our collaborators in medicine). Table 1 also contains recommendations for achieving the required electrochemical characteristics.
<table>
<thead>
<tr>
<th>Parameters</th>
<th>Current value</th>
<th>Required value</th>
<th>Recommendations [21]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electronic conductivity</td>
<td>$0.4 \times 10^4$ S/m</td>
<td>$5 \times 10^4$ S/m</td>
<td>- Add a metal coating</td>
</tr>
<tr>
<td>Ionic conductivity</td>
<td>$1.8 \times 10^{-3}$ S/m</td>
<td>$9 \times 10^{-3}$ S/m</td>
<td>- Increase polymer porosity</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>- Use fast mobile ions</td>
</tr>
<tr>
<td>Electrochemical strain</td>
<td>2%</td>
<td>4%</td>
<td>- Use larger mobile ions</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>- Mechanical amplification</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>- Actuate in resonance</td>
</tr>
</tbody>
</table>

**Table 1.** Electrochemical parameters of the polypyrrole driven catheter; current values, model suggested value for achieving 30 Hz actuation and recommendations for improving the mentioned parameters.

**CONCLUSION**

Two polypyrrole based active catheters were designed and fabricated for imaging and maneuvering applications inside artery. One active catheter, “scanner”, is responsible for scanning an optical fiber within a distance of 1 mm in two dimensions, with a relatively high speed to provide forward viewing images from an artery lesion. The second catheter, the “positioner”, is then actively bent to reach the target lesion and guide a wire to navigate through the lesion, while the first fiber optic mounted catheter is monitoring the procedure in real time using optical coherence tomography (OCT). The bending of the positioner catheter using polypyrrole actuators was demonstrated in an aqueous solution of NaPF$_6$ by applying a square potential of ±0.8 V across the two polymer electrodes and a bending radius of 9.76 mm was achieved in 30 seconds. The scanner catheter was also tested and a 2-D motion with a speed of 1 Hz in y direction and 0.1 Hz in x direction was achieved. A fast actuation of 10 Hz was also tested and a total tip displacement of 1 mm was achieved using over-potential actuation, which degraded the polymer within few cycles. The primary challenge to achieving an effective polypyrrole driven scanner catheter for real time OCT imaging is to demonstrate high speed with reasonable lifet ime. According to a developed model mechanical amplification along with the polymer electrochemical properties’ improvement will result in a high speed scanner catheter suitable for real time imaging applications.

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THE INFLUENCE OF REACTIVE SINTERING PROCESS PARAMETERS ON THE MARTENSITE TO AUSTENITE PHASE TRANSFORMATION OF NITI

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ABSTRACT

NiTi is an intermetallic compound which exhibits a shape memory effect (SME) through a martensitic-austenitic phase change in the temperature range of -200 to 100 °C. The fabrication of NiTi through the reactive sintering of pure Ni and Ti powders offers the advantage of net shape processing and lower processing temperatures compared to more traditional casting and wrought routes of fabrication. Despite over 20 years of research on the sintering of NiTi using pure powders, the influence of processing parameters, including powder size, bulk composition, peak temperature and heating rate, on the shape memory transformation behaviour (i.e. martensite and austenite start and finish temperatures and enthalpies of transformation) is unclear.

In this investigation, the influence of Ni powder size, sintering temperature/time profile and compact bulk composition on the SMA transformation behaviour of NiTi was determined. The sintering investigations were performed in a high temperature DSC, while the phase transformations of post sintered samples were investigated in a low temperature DSC. The overall result included the identification of the optimum conditions to achieve the strongest shape memory transformation.

Keywords: SMA, NiTi, reactive sintering, phase transformations.
INTRODUCTION

NiTi is an intermetallic compound which exhibits a shape memory effect (SME) through a martensitic-austenitic phase change in the temperature range of -200 to 150 °C. A recent analysis has been completed on the influence of Ni-Ti composition on the temperature at which austenite transforms to martensite during cooling (i.e. the martensite start temperature $M_s$) [1]. This work indicates that NiTi with compositions ranging from 48 to 50 at% Ni (i.e. Ti-rich NiTi alloys) have $M_s$ temperatures in the relatively narrow range of 60 to 77 °C. Conversely, Ni-rich NiTi alloys (i.e. 50 to 52 at% Ni) exhibit a rapidly decreasing $M_s$ with increasing Ni content, reaching room temperature at approximately 50.5 at%, and below -175 °C at compositions > 51 at% Ni. Applications of NiTi in industrial couplings and actuators [2,3] utilize the alloy’s ability to undergo a shape memory effect (SME) upon heating and cooling. Generally, the component is martensite at room temperature and transforms to austenite during heating up to 65 to 100°C. This necessitates a high $M_s$ temperature and a Ti-rich or equi-atomic NiTi composition. In these applications, the need for high strength and fatigue resistance requires high density NiTi. From the literature, four general powder metallurgy processes for the production of high density SME NiTi can be outlined including; 1) “pressureless” Ar atmosphere [4-6] or a high vacuum [7-9] sintering of elemental Ni and Ti powders; 2) Hot Isostatic Pressing (HIPing) of elemental Ni and Ti powders [8-10]; 3) HIPing of pre-alloyed NiTi powders [11] and 4) Ca vapor phase (VPCR) sintering of Ni and TiH₂ powders [12, 13]. Despite the reported advantages of HIPing, the use of pre-alloyed powders, or the VPCR approach, pressureless sintering of elemental Ni and Ti powders still represents the least complex, most flexible and economic sintering route available. Consequently, further advancements in the performance of this sintering route are of great technological importance.

While previous research has been extremely useful in advancing our understanding of the sintering behaviour of Ni+Ti powders to form NiTi, a more systematic and encompassing study examining the influence of powder processing parameters on the SME behaviour is now needed to further advance the performance of this pressureless sintering processing route. The objective of this investigation was to determine the influence of 1) Ni and Ti powder size, 2) powder compact composition and 3) sintering profile on the as-sintered NiTi SME transformation temperatures and enthalpies. In order to limit the number of experiments required, a series of optimization steps were executed. First an optimum powder size combination was determined using a limited series of sintering profiles. Second, an optimum compact composition was determined using the optimum powder size from the 1st step and a broader series of sintering profiles.

MATERIALS AND METHODOLOGY

Four different Ni powders (very fine Ni, fine Ni, medium Ni and coarse Ni) and one Ti powder (medium Ti) were used in this investigation. The very fine Ni and fine Ni powder had a mean particle size of 1.2 and 10.7 μm, respectively, as determined by a Malvern Mastersizer 2000. The remaining two Ni powder sizes and the Ti powder were sieved to the following sizes: 32-38 μm (medium Ni), 75-90 μm (coarse Ni) and 25-32 μm (medium Ti). Four powder compacts were prepared including: very fine Ni/medium Ti, fine Ni/medium Ti, medium Ni/medium Ti, and coarse Ni/medium Ti. These compacts represent a wide range of
Ni to Ti powder volume ratios (i.e. approximately 1:30, 1:3, 1:1, 3:1). The 1.2 and 10.7 μm Ni powders were donated by Vale-Inco Ltd., while all other powders were purchased from Alfa Aesar.

All powder mixtures were prepared to the desired composition and subsequently dry milled for 2 hours in a glass jar containing argon gas. The powders were then uniaxially die compacted using 3 successive stages at pressures of 1000 MPa, 750 MPa and 500 MPa. Samples were disc shaped with a pressed diameter of 4.87 mm and thicknesses in the range of 0.6 to 1.0 mm. This allowed them to be sintered in a Netzsch 404C high temperature differential scanning calorimeter (DSC) followed by placement into a TA Instruments Q2000 low temperature DSC for SME measurements without the need for additional sample preparation. The weight and dimensions of each sample before and after sintering were recorded. Pressed green densities varied from 5.15 to 5.7 g/cm³ which correspond to theoretical densities (assuming a fully dense mixture of unreacted Ti-49.6 at% Ni has a density of 6.17 g/cm³) of 83 to 92%.

Sintering was completed directly in the high temperature DSC in a high purity Ar atmosphere using a variety of heating profiles which will be described in more detail in the subsections below. Following sintering, samples were cycled between -197 °C (in liquid N₂) and +150 °C, 10 times in order to stabilize the SME transformation. Following this cyclic treatment the samples were placed in the low temperature DSC, equilibrated at -75 °C, then heated to +150 °C and cooled back to -75 °C using a heating and cooling rate of 5 °C/min. Using the TA software, martensite start, peak and finish temperatures (i.e. $M_s$, $M_p$, $M_f$) were measured during cooling. The enthalpies of transformation from A to M (i.e. $\Delta H_M$) were also measured during cooling by measuring the area under the transformation peak.

**RESULTS AND DISCUSSION**

Influence of Ni powder size

Fig. 1 illustrates the low-temperature DSC heating and cooling traces for all four powder size combinations mixed to a bulk composition of 49.6 at%, after sintering up to 900 °C with no hold. The transformations from martensite to austenite during heating, and austenite to martensite during cooling, are indicated by the endothermic and exothermic peaks respectively. The area under the cooling peaks (i.e. the enthalpy of transformation), and martensite start, peak, finish temperatures (i.e. $M_s$, $M_p$, $M_f$) during cooling measured from this peak are given in Table 1 for the two sintering conditions completed during the powder size investigation.

The magnitude of the peak is the highest (and nearly identical) for the very fine Ni/Ti and fine Ni/Ti powder mixtures. The coarse Ni has only a small, broad peak while the medium Ni/Ti mixture has an intermediate magnitude. Table 1 also indicates that the $M_s$ temperature is very consistent, ranging from 65 to 68 °C. With the exception of the coarse Ni/Ti mixtures, the transformation peaks have a duplex characteristic. A double transformation peak has been reported by a number of researchers to be due to a two stage transformation involving a metastable R phase [12]. However, in those cases the two transform peaks are more separated than observed in Fig. 1. Since these samples are still very inhomogeneous, due to the short
sintering cycle, the doublet peak in this case is more likely due to the transformation of austenite to martensite at slightly different temperatures due to small variations in the NiTi bulk compositions.

![Graph](image)

**Fig. 1:** SME transformation behaviour of all four powder mixtures, with a bulk composition of 49.6% Ni, after sintering at 900 °C.

**Table 1:** Shape memory transformation measurements for 49.6 at% Ni-Ti mixtures as a function of Ni powder size and peak sintering temperature. (*duplex peak)

<table>
<thead>
<tr>
<th>Powder mixture</th>
<th>Sintering Temperature (°C)</th>
<th>Ms (°C)</th>
<th>Mp (°C)</th>
<th>Mt (°C)</th>
<th>ΔHM (J/g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse Ni/Ti</td>
<td>900</td>
<td>65</td>
<td>47</td>
<td>6</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>1020</td>
<td>67</td>
<td>60</td>
<td>53</td>
<td>5</td>
</tr>
<tr>
<td>Medium Ni/Ti</td>
<td>900</td>
<td>66</td>
<td>55-61*</td>
<td>42</td>
<td>11</td>
</tr>
<tr>
<td></td>
<td>1020</td>
<td>67</td>
<td>62</td>
<td>55</td>
<td>14</td>
</tr>
<tr>
<td>Fine Ni/Ti</td>
<td>900</td>
<td>65</td>
<td>57-62*</td>
<td>49</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>1020</td>
<td>68</td>
<td>62</td>
<td>57</td>
<td>18.5</td>
</tr>
<tr>
<td>Very fine Ni/Ti</td>
<td>900</td>
<td>65</td>
<td>57-62*</td>
<td>48</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>1020</td>
<td>68</td>
<td>61</td>
<td>54</td>
<td>20</td>
</tr>
</tbody>
</table>

Fig. 2 illustrates the shape memory transformation behaviour for all four 49.6 at% Ni powder mixtures after sintering up to 1020 °C with no hold. The data of Table 1 and
comparison of Fig. 2 with Fig. 1 indicates that, in all cases, heating to 1020 °C increases the magnitude of the shape memory transformation. The trend of larger enthalpies of transformation with a decrease in Ni powder size observed at 900 °C is also followed for the 1020 °C sintering condition. In general, materials sintered to 1020 °C exhibit a sharper transformation primarily due to an increased $M_f$ temperature (i.e. since $M_s$ changes very little). Also, the doublet nature of the transformation peak disappears indicating that a more homogeneous NiTi phase is produced when heating to 1020 °C.

![SME transformation behaviour of all 49.6 at % powder mixtures after sintering at 1020 °C.](image)

The transformation behaviours of the very fine and fine Ni/Ti samples are similar, with the very fine Ni mixture having a slightly higher $\Delta H_m$. While the very fine Ni mixtures indicate the highest $\Delta H_m$ values, the fineness of the powder made it subject to some agglomeration under the powder mixing conditions used in this investigation. For this reason the fine Ni/medium Ti mixture was chosen as the optimum powder size combination for further study. The fine Ni/Ti powder mixture exhibited a $\Delta H_m$ enthalpy of 18.5 J/g which approaches that measured by the authors for commercial NiTi alloys (24 J/g).

**Influence of powder compact composition**

The short sintering cycles of 900 and 1020 °C with no hold time, produce an inhomogeneous microstructure. This inhomogeneity is illustrated in Fig. 3, which presents an
XRD pattern for a 49.6 at% fine Ni/Ti mixture sintered with no hold at 1020 °C. Both the austenitic and martensitic forms of NiTi are present in the sample as well as Ni$_3$Ti and Ti$_2$Ni intermetallics and residual Ni solid solution phase.

**Fig. 3:** XRD pattern for a 49.6 at% fine Ni/Ti mixture sintered with no hold at 1020 °C

In an effort to produce a more homogeneous microstructure, the use of a two-stage sintering process, similar to that used by other researchers [4], was performed on a 49.6 at% fine Ni/Ti powder mixture similar. The first stage involved heating the sample at 20 °C/min. to 950 °C, followed by an isothermal hold at that temperature for 10 to 30 minutes. The second stage involved heating at 20 °C/min. to 1000°C, followed by an isothermal hold for either 2, 4 or 6 hours. The purpose of this step was to increase the volume fraction of NiTi through homogenization of the microstructure.

Fig 4 illustrates the low temperature cooling DSC trace for the three, two-stage sintering cycles as well as a first-stage-only cycle for comparison purposes. Only the cooling trace is shown in the figure since the same trends were observed during heating. Measurements made from the cooling traces are given in Table 2.

Sintering the mixture under only the first-stage hold segment at 950 °C produces an SME transformation behaviour that is very similar to that developed in the fine Ni/Ti mixture heated directly to 1020 °C. Therefore, increasing the temperature from 900 °C to 950 °C and adding a hold segment further homogenizes the solid-state sintered microstructure and produces a stronger SME transformation. However, adding a further homogenization stage at 1000 °C causes a significant broadening of the transformation over a wider temperature range and also reduces its enthalpy from a maximum of 20 J/g, after 2 hours at 1000 °C, down to 16 J/g after 6 hours. This indicates that a lower fraction of martensitic NiTi is present at room temperature, in the more homogenized samples.
Table 2: Shape memory transformation characteristics as a function of sintered powder composition and a single and two step sintering profile for a fine Ni/medium Ti powder mixture.

<table>
<thead>
<tr>
<th>Compact composition</th>
<th>Sintering Time (min.)</th>
<th>( M_s ) (°C)</th>
<th>( M_p ) (°C)</th>
<th>( M_f ) (°C)</th>
<th>( \Delta H_M ) (J/g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>49.6 at%Ni/Ti</td>
<td>950 °C</td>
<td>70</td>
<td>69.2</td>
<td>60</td>
<td>18</td>
</tr>
<tr>
<td></td>
<td>950 + 1000 °C (2 hrs.)</td>
<td>70</td>
<td>63</td>
<td>-20</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>950 + 1000 °C (4 hrs.)</td>
<td>67</td>
<td>43</td>
<td>-21</td>
<td>16</td>
</tr>
<tr>
<td></td>
<td>950 + 1000 °C (6 hrs.)</td>
<td>67</td>
<td>34</td>
<td>-34</td>
<td>16</td>
</tr>
<tr>
<td>49 at%Ni/Ti</td>
<td>950 °C</td>
<td>70</td>
<td>64</td>
<td>60</td>
<td>18</td>
</tr>
<tr>
<td></td>
<td>950 + 1000 °C (6 hrs.)</td>
<td>72</td>
<td>64</td>
<td>40</td>
<td>20</td>
</tr>
<tr>
<td>50 at%Ni/Ti</td>
<td>950 °C</td>
<td>68</td>
<td>63</td>
<td>59</td>
<td>16</td>
</tr>
<tr>
<td></td>
<td>950 + 1000 °C (6 hrs.)</td>
<td>49</td>
<td>-17</td>
<td>-48</td>
<td>7</td>
</tr>
</tbody>
</table>

The above conclusion is supported by the XRD results of Figure 5. Firstly, comparing the XRD spectrum of the 1000°C (4hrs.) sample to the 1020 °C (no hold) sample of Fig. 3, indicates that the longer hold time does cause some homogenization, eliminating the \( \text{Ni}_3\text{Ti} \) and residual Ni phase from the sample. Secondly, the 4 hr. specimen consists of a mixture of...
austenitic and martensitic NiTi with a small fraction of residual Ti2Ni (or Ti4Ni2O which has a similar XRD spectrum) present. In addition, observations of the relative peak heights of the spectrum arising from austenite and martensite in Figs. 3 and 4 give a semi-quantitative indication that relatively more austenite compared to martensite develops with the 1000 °C (4 hrs.) heat treatment. Finally, increasing the hold time at 1000 °C to 6 hrs. develops a predominantly austenitic NiTi microstructure, with martensitic NiTi as a minor phase.

![XRD pattern](image)

*Fig. 5: XRD pattern of 49.6 at% fine Ni/Ti powder mixtures following a second-stage homogenization sintering cycle for 4 and 6 hrs, showing the phases present in the microstructure.*

This trend of transformation behaviour with sintering profile is hypothesized to be due to progressive homogenization coupled with oxide contamination. At short sintering times, the microstructure is inhomogeneous which produces a variable NiTi composition at the sintering temperature. According to the Ni-Ti binary phase diagram, the NiTi composition can range from 48 to 55 at% Ni at 1000 °C. Following cooling, Ti rich NiTi will contain martensite at room temperature, while Ni rich regions will contain austenite, because of the variation in Ms with composition. The presence of a dual phase microstructure is confirmed by the XRD results of Fig. 3. The martensitic NiTi will give rise to the high temperature transformation behaviour exhibited in the DSC trace of Fig. 2, while the Ni-rich NiTi will have a transformation behaviour at temperatures below -75 °C (and therefore not visible on the DSC trace).

As sintering time increases, the NiTi composition becomes more uniform and will approach the bulk composition of the mixture. The 1000°C (6 hrs.) sample XRD pattern (cf. Fig. 5) confirms that the homogenized NiTi is in the form of austenite, indicating a Ni-rich
NiTi composition. This is inconsistent with the prepared bulk composition of 49.6 at% Ni. This discrepancy can be understood by considering oxide contamination. As pointed out in [12, 13], the preferred oxide in the Ni+Ti system is Ti$_4$Ni$_2$O. Since this oxide combines more Ti than Ni, its formation can have the effect of shifting the NiTi phase to a higher Ni content than that indicated by the bulk composition.

To test this hypothesis, two additional powder mixtures made from the fine Ni/medium Ti powders were prepared at compositions above and below 49.6 at% Ni (i.e., at 49 at% and 50 at%). These samples were then sintered using: 1) a first-stage-only, at 950 °C with no hold cycle to produce an inhomogeneous mixture and 2) a two-stage process with the second at 1000 °C for 6 hours to further homogenize the NiTi phase. These sintered samples were then placed in the low temperature DSC and their transformation behaviour measured. The key results and measurements are given in Fig. 6 and Table 2 above.

![Graph showing heat flow vs. temperature for Ni/Ti mixtures](image)

**Fig. 6:** Austenite to martensite transformation behaviour of fine Ni/Ti mixtures with 3 compact compositions sintered using the two-stage sintering procedure: i.e. 950 + 1000°C (6 hrs.)

While not included in Fig. 6, all three powder compositions produced nearly identical, sharp, high temperature transformations following only the short sintering cycle at 950 °C. The data of Table 2 illustrates this. This supports the above hypothesis, where inhomogeneous mixtures will produce NiTi with a compositional gradient from 48 to 55 at%, regardless of the prepared bulk composition. After the homogenizing second sintering stage, the transformation response in the mixtures is very different as illustrated in Fig. 6. The 49 at% Ni sample has maintained a strong, although slightly broader, high temperature transformation with a $M_s$ that actually increases from 70 to 72 °C and an enthalpy of
transformation that increases from 18 to 20 J/g. In contrast, the 50 at% Ni sample exhibits a significant reduction and broadening of its transformation behaviour to a lower temperature range and a reduction in the M_s from 68 to 49 °C. Again these results support the hypothesis that, following the removal of intermediate phases, the NiTi is free to homogenize to a uniform composition. This NiTi composition is slightly Ni-rich compared to the bulk composition, due to minor oxide contamination. Preparing a more Ti-rich bulk composition is capable of counteracting the oxidation effects, thus stabilizing a martensitic NiTi composition at room temperature. Thus the 49 at% fine Ni/medium Ti alloy produces a strong high temperature transformation response, even after homogenization.

SUMMARY AND CONCLUSIONS

During the sintering of pure Ni and Ti powders, the use of finer Ni powders produces more rapid NiTi phase formation. This results in a strong shape memory transformation with a high transformation temperature (i.e. M_s = 68 °C) even at short sintering times. For 49.6 at% Ni powder mixtures, longer sintering times produces further homogenization but a diminishing SME in the high temperature range of interest. This is due to oxide contamination which produces a Ni-rich homogenized NiTi phase. Adjusting the bulk composition to a more Ti-rich value, counteracts the oxide contamination effects. A Ti-49 at% Ni mixture of 10 μm Ni/32 μm Ti powders sintered at 1000°C for 6 hours produces a homogeneous NiTi phase which maintains a strong SME exhibiting an M_s = 72 °C.

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MODELLING OF PIEZOELECTRIC FIBER COMPOSITE MATERIAL FOR ULTRASONIC WAVE GENERATION

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ABSTRACT

This paper presents the assessment of an analytical model for the simulation of ultrasonic wave generation by piezoelectric fiber composites (PFC) with inter-digital electrodes in plate-like structures. PFC are attractive for applications where directional actuation or measurement is sought. While most of previous modelling work on PFC has been reported at low frequency or for out-of plane poling, this work extends the investigation to higher frequencies with in-plane poling for potential application in structural health monitoring of aerospace structures. The piezoelectric constitutive equations are adapted to this configuration of PFC and planar wave generation is assumed. Simulation results are presented to illustrate the directivity pattern associated to the use of such PFC actuators. These results are then assessed with experimental measurements conducted with a laser vibrometer at different frequencies, and thus at different wavelengths with respect to the inter-digital spacing. Results indicate a strong directivity at specific frequencies which might allow selective excitation of waves in specific areas of inspected structures. This work has been supported by the Ministère du Développement économique, de l’innovation et de l’exportation (MDEIE) in Québec, Canada.

Keywords: Piezoelectric fiber composites, inter-digital electrodes, structural wave propagation, Structural Health Monitoring, aerospace structures.
INTRODUCTION

Bulk or monolithic piezoelectric sensors and actuators have largely been used in active control of noise and vibration, and structural health monitoring applications [1-2]. Piezoceramics (PZT) have been used mostly and, while they offer a large sensitivity to strain, they suffer from brittleness, omnidirectional excitation or measurement and difficulties associated with electrical connections. Polyvinylidene fluoride (PVDF) piezoelectric films offer an interesting alternative to PZT, with increased material flexibility. A number of electrode patterns have been proposed for PVDF, including interdigital transducers [3]. However, PVDF suffer from low sensitivity and connection problems.

Structural Health Monitoring (SHM) requires the sensors to have large sensitivity and since a spatial localization of a defect is sought, directional sensing is desirable. A number of approaches have been proposed to address these requirements, from arrays to inter-digital transducers [4]. By using proper signal processing, these approaches have allowed directive measurements or generation to be made. However, these approaches intrinsically involve spatial filtering and thus, tend to restrict the frequency band of the sensor or actuator. Moreover, with arrays, the directivity will vary whether the measurements are conducted in the near-field or in the far-field.

Macro Fiber composites (MFC) [5], also called Active Fiber Composites (AFC) [6] or Piezoelectric Fiber Composites (PFC) have been more recently developed to offer directional sensing and actuation with electrode connection through screen-printing tools. Greater sensitivity is naturally obtained along the axis of the piezoelectric fibers used in the fabrication of the transducer which are embedded in an epoxy matrix and sandwiched between two sets of inter-digital electrodes (IDT). Modelling of these transducers have been conducted either using finite element techniques [7], but without assessing directivity for pulse generation, analytically for source localization [8], or experimentally [9], but restricted to the through-the-thickness polarization of the piezoelectric material.

It is the purpose of this paper to assess an analytical model for the simulation of the directional measurement and generation of Lamb waves using a PFC polarized along the length of the fibers.

THE PIEZOELECTRIC FIBER COMPOSITE TRANSUDER

The PFC is made from piezoelectric fibers aligned along axis 1 and embedded with an epoxy matrix on both sides of which electrode fingers (along axis 2) are deposited, in such a way that positive and negative electrodes alternate with each other, as shown in Figure 1a).

Polarization of the piezoelectric material can be applied using two distinct configurations. The piezoelectric fibers can be polarized along axis 3, i.e electrodes 1 and 2 are connected together on each side to create one electrode on top and one electrode on the bottom. Fibers can also be polarized along axis 1 by connecting together electrodes 1 (top and bottom) and by connecting together electrodes 2 (top and bottom), as shown in Figure 1b).
Fig. 1: a) Configuration of the Piezoelectric Fiber Composite (PFC) and b) resulting polarization field along the length of the fibers.

ANALYTICAL MODELLING

Formulation for piezoelectric electro-mechanical coupling

The PFC is modelled as an orthotropic laminate subjected to plane strain. With these assumptions, the governing equations are:

\[
\begin{align*}
\{\sigma\} &= \begin{bmatrix} \sigma_x \\ \sigma_y \\ \sigma_{xy} \end{bmatrix} = [Q] \{\varepsilon\} - E_i \{d\} \\
\{\tau\} &= \begin{bmatrix} \tau_{xz} \\ \tau_{xz} \end{bmatrix} = \begin{bmatrix} Q_{44} & Q_{45} \\ Q_{45} & Q_{55} \end{bmatrix} \begin{bmatrix} \gamma_{12} \\ \gamma_{22} \end{bmatrix}
\end{align*}
\]  

(1)  

(2)
\[ D_i = \{d\}^T [\mathcal{Q}](\{\varepsilon\} - Ec_i \{d\}) + \xi_{ii} Ec_i \]  

(3)

where \(\{\sigma\}\) and \(\{\tau\}\) represent normal stress and shear vectors, \(D_i\) is the electrical displacement along the \(i\) direction, \([\mathcal{Q}]\) is the equivalent anisotropic elasticity matrix, \(\{\varepsilon\} = \begin{bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \end{bmatrix}\) and \(\{\gamma\} = \begin{bmatrix} \gamma_{yy} \\ \gamma_{xx} \end{bmatrix}\) are the strain vectors, \(Ec_i\) is the electric field along the \(i\) direction, \(\xi_{ii}\) is the electric permittivity along the \(i\) direction, and \(\{d\}\) is the vector of piezoelectric coupling coefficients. This formulation is valid for any polarization direction. For the case studied in this paper, \(i.e.\) for polarization along axis 1, \(i = 1\) and

\[ \{d\} = \begin{bmatrix} d_{11} \\ d_{12} \\ 0 \end{bmatrix} \]  

(4)

Sensor application of the PFC

When used as a sensor, and considering polarization along axis 1, Eq. (3) becomes

\[ D_1 = \begin{bmatrix} d_{11} & d_{12} & 0 \end{bmatrix} \begin{bmatrix} E_1 & \nu_{12}E_1 & 0 \\ \nu_{12}E_1 & E_2 & 0 \\ 0 & 0 & G_{12} \end{bmatrix} \begin{bmatrix} \varepsilon_{11} \\ \varepsilon_{22} \\ \gamma_{12} \end{bmatrix} - Ec_1 \begin{bmatrix} d_{11} \\ d_{12} \\ 0 \end{bmatrix} + \xi_{11} Ec_1 \]  

(5)

where \(E_1, E_2, \nu_{12}\) and \(\nu_{21}\) are the equivalent mechanical properties of the anisotropic composite allowing to calculate the coefficients of the matrix \([\mathcal{Q}]\). This equation is valid within each section of the fiber between two lines of electrodes. Depending on the poling direction, the sign of \(\{d\}\) will change. Since the charge is preserved on the electrodes, the following can be written over the surface of the electrodes:

\[ \int_{S_e} D_i ds = 0 \]  

(6)

Using Eqs. (5) and (6), the following can be written:

\[ Ec_1(x,y) = \frac{\varepsilon_{11}(x,y)(d_{11}Q_{11} + d_{12}Q_{12}) + \varepsilon_{22}(x,y)(d_{11}Q_{12} + d_{12}Q_{12})}{d_{11}^2Q_{11} + 2d_{11}d_{12}Q_{12} + d_{12}^2Q_{22} - \xi_{11}} \]  

\[ = A_1\varepsilon_{11}(x,y) + A_2\varepsilon_{22}(x,y) \]  

(7)
As illustrated in Figure 2, the electrical potential $V$ can be estimated by integrating the electrical field over an inter-digital domain:

$$V = \iiint \frac{Ec_1}{S} \, dx \, dy \, dz$$  \hspace{1cm} (8)

![Figure 2: a) Electrical field in an inter-digital domain and b) simplified representation.](image)

As the electrodes constitute an inter-digital pattern of length $nd$ with even $n$ ($2d$ being the pitch of the pattern), and as the width of the sensor is $b$ and the thickness of the fibers is $h$, Eq. (8) can be rewritten as:

$$V_{\text{total}} = \sum_{i=1}^{n} \int_{(i-1)d-h/2}^{id+h/2} \frac{Ec_i(x,y)}{b} \, dx \, dy = \int_{-(nd/2-h/2)}^{nd/2-h/2} \frac{Ec_1(x,y)}{b} \, dx \, dy$$  \hspace{1cm} (9)

where $Ec_i(x,y)$ is the electric field induced within a section $i$ of the sensor located between $(i-1)d$ and $id$ along axis 1.

**Directivity**

Figure 3 presents a plane wave propagating in a structure along a direction $x'$ at an angle $\theta$ with respect to axis 1 (fiber axis). Along this axis, the displacement of the wave is expressed as $u_{x'} = Ae^{-jkx'}$ and the strain along the same axis can be expressed as:

$$\varepsilon_{x'x'} = \frac{\partial u_{x'}}{\partial x'} = -jkAe^{-jkx'}$$  \hspace{1cm} (10)

In the sensor’s system of axis, the strains are given by:

$$\varepsilon_{11} = \varepsilon_{x'x'} \cos^2 \theta = \left(jkA \cos^2 \theta \right) e^{-jkx'}$$

$$\varepsilon_{22} = \varepsilon_{x'x'} \sin^2 \theta = \left(jkA \sin^2 \theta \right) e^{-jkx'}$$  \hspace{1cm} (11)

and from Eq. (7), the electrical field can be expressed as:

$$Ec_1(x') = \left(A_1 \cos^2 \theta + A_2 \sin^2 \theta \right) \varepsilon_{x'x'}$$  \hspace{1cm} (12)
Fig. 3: Plane wave propagating in a structure.

In the sensor’s system of axis, the strains are given by:

\[ \varepsilon_{11} = \varepsilon_{xx} \cos^2 \theta = -(jKA \cos^2 \theta) e^{-jke^x} \]
\[ \varepsilon_{22} = \varepsilon_{xx} \sin^2 \theta = -(jKA \sin^2 \theta) e^{-jke^x} \]  

(13)

and from Eq. (7), the electrical field can be expressed as:

\[ E_{e_1}(x') = (A_1 \cos^2 \theta + A_2 \sin^2 \theta) e_{xx} \]  

(14)

Using Eq. (9), the total electrical potential is obtained:

\[ V_{total} = \int_{-nd/2}^{nd/2} \int_{-b/2}^{b/2} \frac{A_1 \cos^2 \theta + A_2 \sin^2 \theta}{b} e_{xx} \ dx dy = \int_{-nd/2}^{nd/2} \int_{-b/2}^{b/2} C(\theta) e_{xx} \ dx dy \]  

(15)

which, using Eq. (10), leads to:

\[ V_{total} = -jKC(\theta) \int_{-nd/2}^{nd/2} \int_{-b/2}^{b/2} e^{-jke^x \cos \theta - jky \sin \theta} \ dx dy \]
\[ = -\frac{4jKC(\theta)}{k \cos \theta \sin \theta} \sin \left( \frac{knd \cos \theta}{2} \right) \sin \left( \frac{kb \sin \theta}{2} \right) \]  

(16)

The directivity is expressed as:

\[ D_{total} = \left| \frac{V_{total}(\theta)}{\bar{V}_{total}} \right| \]  

(17)

where \( \bar{V}_{total} \) is the mean value of \( V_{total}(\theta) \) over all the angles.
EXPERIMENTAL ASSESSMENT

Parameters used for the assessment

In the following, experimental measurements are conducted to assess the directivity simulation results at three wavelengths, corresponding to (i) length of the sensor, (ii) 2/3 of the length of the sensor and (iii) half the length of the sensor. Table I presents the properties used in the modelling of the directivity of the PFC. The dispersion curves for aluminium shown in Figure 4 are used to determine the frequencies corresponding to these wavelengths, as presented in Table II. Simulations results for the directivity obtained using Eq. (17) are presented in Figures 6c), 7c) and 8c).

<p>| Table I: Properties of the PFC. |</p>
<table>
<thead>
<tr>
<th>Properties</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1$</td>
<td>30.34 GPa</td>
</tr>
<tr>
<td>$E_2$</td>
<td>15.86 GPa</td>
</tr>
<tr>
<td>$G_{12}$</td>
<td>5.52 GPa</td>
</tr>
<tr>
<td>$\eta_{12}$</td>
<td>0.31</td>
</tr>
<tr>
<td>$\eta_{21}$</td>
<td>0.16</td>
</tr>
<tr>
<td>$d_{11}$</td>
<td>$460 \times 10^{-12}$ C/N</td>
</tr>
<tr>
<td>$d_{12}$</td>
<td>$-210 \times 10^{-12}$ C/N</td>
</tr>
<tr>
<td>$\zeta_{11}$</td>
<td>$16.37 \times 10^{-9}$ F/m</td>
</tr>
<tr>
<td>$b$</td>
<td>14 mm</td>
</tr>
<tr>
<td>$n$</td>
<td>28</td>
</tr>
<tr>
<td>$d$</td>
<td>0.5 mm</td>
</tr>
</tbody>
</table>

Fig. 4: Dispersion curve for flexural wave propagation in aluminium.

<table>
<thead>
<tr>
<th>Table II: Wavelengths and frequencies.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wavelength</td>
</tr>
<tr>
<td>----------------------------------</td>
</tr>
<tr>
<td>28 mm (length of the sensor)</td>
</tr>
<tr>
<td>18.6 mm (2/3 of the length of the sensor)</td>
</tr>
<tr>
<td>14 mm (1/2 of the length of the sensor)</td>
</tr>
</tbody>
</table>

Experimental setup

The PFC was bonded at the center of a 1.08 mm thick aluminum plate. Only the first wavefront is considered. Figures 5a) and 5b) present the experimental setup used, with the 0.75 m x 1.05 m clamped plate.

In order to measure the directivity of the transducer two experiments were conducted. The first experiment is the measurement of the time response of the PFC used as sensor. The second experiment is the measurement of the velocity field resulting from the radiation of the PFC used as an actuator. Each experiment is done for a surface representing angles from 0° to 90° centered at the transducer therefore reducing measurement time and exploiting the symmetry of the PFC. The plate is scanned using a Polytec laser vibrometer with a bandpass filter to provide a velocity reference in the calculation of the sensitivity.
Assessment for sensing PFC

This experiment is conducted using ten bulk PZT actuators (8 mm x 8 mm x 0.5 mm) bonded on the back of the aluminum plate. Bursts are generated at the ten PZTs (0 to 90° using 10° steps) and the response of the plate is measured by the PFC. The actuators are located far enough to ensure that a plane wave reaches the PFC before any reflections from the plate boundaries. Ten time sequences (sampling frequency of 10 MHz) are averaged at each frequency and the results are presented as polar plots. The radius of the graph corresponds to the time vector and the amplitude is the envelope of the sensor voltage response. It results in a pseudo-polar surface where the maximum amplitude is correlated with the directivity of the sensor.

Figure 6a) presents is the time response at 13.37 kHz. The maximum sensitivity, obtained from a laser measurement at the center of the PFC, is 40.75 V/(m/s) for a direction of 45° which is confirmed by the analytical directivity plot. Figure 7a) presents the time response at 30.29 kHz, where the maximum sensitivity (84.5 V/(m/s)) is obtained for a direction of 0°. Figure 8a) presents the time response at 53.46 kHz. The maximum sensitivity is 116 V/(m/s) at 30°. In all three cases, very good agreement between simulation and experimental results is obtained. One can note that S0 mode is clearly visible in Figure 8 and appears before the A0 mode.

Assessment for actuating PFC

Bursts are generated at the PFC using a high voltage amplifier at 80 V to 96 V, depending on the frequency. Bursts are centered around the frequency of interest with a sampling frequency of 4 MHz. Figures 6b), 7b) and 8b) present the RMS vibration level over the area scanned by the laser vibrometer. Again, the agreement between the simulation results and the measurement conducted with the laser vibrometer is very good.
CONCLUSIONS

Analytical directivity patterns of a PFC were experimentally assessed as a sensor. Directivity patterns were shown to depend on the wavelength with respect to the length of the sensor. Then the PFC was used as an actuator. It was shown that it is possible to measure the directivities of this emitter. For the plate used, sensitivity was measured in (m/s)/V. To confirm the analytical model, the second experiment consisted in using a set of secondary
actuators to send a plane wave to the IDT transducer acting now as a sensor. The angular resolution was chosen as 10°. Once again, the IDT sensor acts as predicted. It was also possible to determine the sensibility of this sensor in the principal direction of sensing $V/(m/s)$. Future work includes using those actuators and sensors in SHM strategies where a particular direction is to be used for sensing. As the principal lobe of directivity is dependent on the frequency, the use of a PFC as an actuator can also be used to send vibration waves in particular directions considering that the actuator is fixed. This has to be compared with using several actuators in a beam forming strategy.

ACKNOWLEDGEMENTS

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POWDER METALLURGY FABRICATION OF HYBRID MONOLITHIC SMA ACTUATORS

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ABSTRACT

Shape memory alloys (SMA) provide exciting opportunities for scalable actuation systems. As SMA actuators are scaled down in size, cooling increases and bandwidth, one of the traditional drawbacks of larger-scale SMA actuators, improves. However, the inclusion of a bias element with which to cycle the SMA actuator becomes difficult at very small scales. One technique that has been applied to avoid the necessity of having to include a separate bias element is the use of local annealing to fabricate a monolithic device out of a single piece of non-annealed NiTi. The annealed portion exhibits the shape memory effect while the remainder acts as structural support and provides the bias force required for cycling. This approach suffers several limitations in both fabrication and design.

Here, we present the evolution of this idea: the hybrid monolithic actuator. Using powder metallurgy (PM) techniques, a hybrid monolithic actuator is fabricated with areas of shape memory effect (SME) NiTi as well as areas of superelastic NiTi which act as restoring members. Bulk training is possible by constraining the monolithic component in an appropriate jig and subjecting the entire device to a heat treatment cycle, avoiding the difficulties of local annealing. Recoverable strains in the superelastic NiTi are comparable to those in the SME material, increasing capability for cyclic motion. A prototype PM-fabricated bending actuator is demonstrated which achieves cyclic tip deflection of approximately 10\% of its length during a heating/cooling cycle.

Keywords: Shape Memory Alloys, Monolithic Actuator, Powder Metallurgy, NiTi.
INTRODUCTION

The shape memory effect (SME) is associated with a thermally induced phase transformation in the shape memory alloy (SMA), NiTi. As a result of the difference in stress-strain characteristics between the two SMA phases, the SME can be used to design thermo-mechanical actuators capable of repeatedly generating significant stress (~170MPa) over displacements of up to 6% strain. SMA actuators are generally fabricated using bulk drawn NiTi, in either wire or spring forms, with a regular spring providing the required actuator reset force. The inherently linear motion, high power/weight ratio, and scalability of SMA actuators make them appealing for motion generation at meso- and micro-scales\(^1\). As a side benefit of scaling, the increased ratio of surface area to volume results in better cooling and higher actuator bandwidths.

Much attention has been given to micro-electromechanical systems (MEMS) applications of thin-film SMA actuators, where traditional MEMS fabrication techniques, such as vapour deposition or sputtering, can be used to form NiTi [1]. However, the design of meso-scale actuators remains a challenge, notably due to difficulties in establishing mechanical connections with reset force mechanisms. At these scales, existing NiTi forms (wire, helical springs) and fabrication techniques become increasingly difficult or impossible to use. At the same time, increasingly miniature devices require actuators with high energy density, and in some cases SMA is one of the only viable options. For example, the next generation of ingestible medical devices for capsule endoscopy should have the ability to stop progression through the digestive tract, and eventually the capacity for independent, controlled motion. Because of the volume constraints, SMA actuators have been proposed. For example, [2] proposes a clamping device for immobilizing a capsule in the gastrointestinal tract using a compliant clamp actuated by discrete SMA wires. In [3], SMA-actuated “legs” are proposed which are again actuated by discrete SMA wires routed over several pulleys. The assembly of actuation and restoring-force components for micro- and meso-scale actuators requires a high level of precision in order to achieve the required accuracy and actuator pre-strain. In addition, the small size of mechanical components complicates their manipulation and joining.

The eventual goal of the research described in this paper is to develop design and fabrication techniques for monolithic SMA actuators using a powder metallurgical approach. Monolithic actuators are those in which the actuation and reset mechanisms are fabricated from a single mechanical component, eliminating the need for micro-assembly. In previous work, we have described the advantages of the powder metallurgy (PM) approach over existing techniques, and presented FEM simulation results for a proof-of-concept actuator [4]. Here, we briefly review the concept and drawbacks of past approaches to monolithic actuator design and the benefits of using a PM approach. We then present experimental results from a novel PM-fabricated prototype hybrid monolithic bending actuator.

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\(^1\) Meso, meaning “in the middle,” denotes here dimensions between “macro-scale” and “micro-scale” (MEMS) devices, typically on the order of a few millimeters.
MONOLITHIC ACTUATORS

Researchers at the Ecole Polytechnique in Lausanne (EPFL) developed the concept of the monolithic SMA actuator [5], in which the actuation and reset mechanisms are co-fabricated from a single piece of material. Conceptually, the fabrication of a monolithic SMA actuator is a two-stage process. First, the appropriate geometry is created from a single piece of material. Subsequently, the SME is imparted only to the actuation portion of the device through a local annealing process. The ability to localize the annealing is critical to the process, since the non-annealed portions of the material are required to act as a restoring bias force and must not demonstrate the SME. Researchers at EPFL have produced a number of functional micro-grippers using laser processing. As an example, in [6] the geometry is cut in a first stage from NiTiCu sheet using an Nd-YAG laser, and pulsed laser processing is later used to perform local annealing and create the active part of the device.

In [7], a prototype monolithic actuator was fabricated from a single strip of NiTi ribbon using water-jet cutting and local annealing by resistive contact heating. Figure 1 shows the drawings and final part, fabricated from a sample strip of SM495 NiTi ribbon from Nitinol Devices & Components (NDC), approximately 7 mm wide and 0.25 mm thick. The actuator comprises two functional parts: the annealed beam that displays the SME (shown hatched) and the non-annealed portal which surrounds the beam. The large area and through-holes on the left are non-functional and are for clamping and fixturing during annealing.

Figure 2 shows the conceptual operation of the actuator. During annealing of the center beam, the actuator is constrained in a curved fixture, in position a). This sets the memorized shape of the center beam while simultaneously storing energy in the non-annealed portal. After annealing, the part is removed from the fixture. As the beam cools and martensite forms, the energy stored in the portal causes tip deflection to decrease (position c). When the center beam is subsequently heated, the transformation to austenite causes it to want to return to its annealed shape, increasing the tip deflection and energy stored in the restoring portal (position b). Thus, the actuator generates an out-of-plane change in tip deflection during thermal cycling.

In [7], a lumped-element equivalent spring-mass model of the actuator was developed and coupled with heating and SMA phase kinetics models in order to simulate the expected actuator behaviour. For this geometry, cyclic tip deflections of 0.34 mm were predicted. In [4], COMSOL Multiphysics FEM software was used in conjunction with the solid mechanics package to obtain predictions of actuator deflection during operation (see Figure 3), with results which closely matched the lumped model predictions from [7].
Despite the success of this work, several drawbacks were identified with the monolithic approach. First, fabrication of the final shape from bulk material can be difficult and expensive due to the limitations of traditional machining. Secondly, it is difficult to impart local annealing only within the regions of the actuator in which the SME is desired. This thermal control becomes increasingly difficult as the overall size of the device is reduced. Finally, due to the limited yield strain of the non-annealed restoring portal, the range of motion is minimal in comparison with the recoverable strains achievable by most SMAs. In other words, the portal material limits performance.

An alternative approach to material removal for part fabrication is to use a powder metallurgy (PM) approach to create a near-net shape part through sintering. The process is potentially less expensive than the standard high-temperature vacuum arc re-melting process used for bulk forms, and allows the direct fabrication of planar parts of complicated geometry, eliminating the machining step. The standard powder approach involves mixing a combination of elemental Ni and Ti powders in the appropriate ratio (typically about 50 at% Ni) and milling to obtain a uniform mixture. This dry powder mixture is then compressed in a die corresponding to the desired geometry, and subsequently sintered. To facilitate manipulation and cutting the desired geometry, tape casting may be used. This process involves suspending the powder mixture in a polymer matrix, creating slurry which can be used to produce large flat sheets of polymer-suspended powder. These sheets can then be cut into various geometries, stacked, pressed, and sintered. The powder sintering approach eliminates the need for difficult machining of NiTi, and the ability to combine and sinter tapes of different mixtures introduces the possibility of creating so-called “hybrid” monolithic actuators in which other materials combine with NiTi in the same structure.

**HYBRID MONOLITHIC ACTUATOR**

In [4], we presented FEM simulations of a hybrid monolithic actuator design capable of overcoming the three major disadvantages mentioned in the previous section. Utilizing powder metallurgy tape casting techniques, we are able to avoid the difficult and expensive shaping methodologies used to create the monolithic actuator already described. The chief advantage of the tape casting process is that it is capable of forming large-area, flat sheets that can be cut, stacked, and pressed to form complex planar geometries. Introducing sheets of varying compositions and materials provides a controllable method of manufacturing...
monolithic actuators with non-uniform, or hybrid, compositions. Once sintered, we are left with a planar near net shape hybrid monolithic actuator.

To overcome the limitation imposed by the yield strain of the restoring material, one with strains similar to those recoverable through the SME is required. By varying the elemental composition of NiTi only slightly, it is possible to obtain the so-called “superelastic” response, with strains of up to 8% recoverable upon unloading [8]. In addition to high yield strains, the similarity in composition reduces the possibility of mechanical problems at the material interface.

To alleviate the need for local annealing, a partial separation of the actuator is used, allowing deformation of the SMA beam independent of the superelastic portal as seen in Figure 4b. Annealing in this state sets the desired shape of both materials. The left regions of the actuator are then clamped together forming the final state of the actuator as seen in Figure 4c. Using the same physical dimensions as the monolithic actuator described in the previous section, FEM simulations predict tip deflections for the hybrid monolithic actuator in the range of 2.5mm without the need for a complex manufacturing and annealing process. This represents an increase of over seven times the deflection predicted for the monolithic actuator.

![Fig. 4. FEM Model of Hybrid Monolithic Actuator](image)

**HYBRID MONOTLITHIC ACTUATOR PROTOTYPE FABRICATION**

A prototype of the hybrid monolithic actuator shown in Figure 4 was fabricated using conventional reactive sintering of tapes made from elemental Ni and Ti powders. Reactive sintering is done at relatively low temperatures (below Ni and Ti melting points) and relies primarily on atomic diffusion. The major challenge of reactive sintering is establishing sintering parameters which favour the formation of dense, homogenous NiTi while avoiding the formation of, and residual deposits of the intermetallic compounds NiTi$_2$ and Ni$_3$Ti.

We have been investigating the effect of powder size and sintering profiles on the homogeneity and density of the resulting parts for several years. Based on the results of our most recent work [9], a composition of 49 at% Ni was chosen for the SMA beam and a composition of 50 at% Ni for the super-elastic beams. The Ni powder utilized is Inco 123 with a mean particle diameter of 10.7 μm and a Ti powder (Alfa Aesar) with a range 25-38 μm (range expanded somewhat from [9]).
To produce a tape the Ti-49 at% Ni (or 50 at% Ni) powders were mixed together in a dispersant and solvent and ball milled for 7 hours. The polymer was added and the whole mixture milled for a further 18 hours. Once mixing was complete the metal powder/polymer mix was cast into a tape to a thickness of approximately 0.5 mm. After the solvent had evaporated a solid but flexible polymer matrix held the powders in place, and desired shapes could then be cut out and pressed (see Figure 5). The shapes are typically pressed to approximately 210 MPa.

Before the pressed shapes were sintered, the polymer was burned off by slowly heating the parts in a vacuum furnace up to 350 °C. At such low temperatures, little to no sintering occurred and voids/pores remained in the areas previously occupied by the polymer. The shapes were pressed a second time, to approximately 500 MPa, to increase the density prior to sintering. Sintering was performed in a vacuum furnace using a slow heating rate of 2 °C/min from 500 °C to 1000 °C followed by a hold for 4 hours at 1000 °C.

Prior to designing a prototype, preliminary mechanical testing was performed on the tapes. A die with a tensile “dog bone” shape was designed and used for pressing the tapes to obtain tensile specimens with consistent dimensions (Figure 6). Testing was performed on an Instron 5548 with LVDT displacement measurements accurate to ±0.0001 mm. Specimens were cycled multiple times with the goal of stabilizing the shape memory effect and the super-elastic behaviour. Figure 7 shows the stress-strain results for each composition of tape on the first and fourth cycle. The shape memory effect was captured by pulling at a strain rate of 0.5 mm/min to a predetermined displacement of 0.25 mm. The specimen was then unloaded at the same strain rate until zero load was achieved. Upon unloading, at approximately 30 N a heat gun pointed at the specimen was turned on. This caused austenite to form and the specimen to recover during unloading. The upswing in load halfway through unloading in Figure 7 was where the heat gun was turned on. Specimens displaying super-elastic behaviour were cycled in the same way only no heat gun was used as recovery was not temperature dependant. No extensometer was available and machine stiffness was not known, therefore a conservative gauge length was used. This led to smaller reported strains than were achieved. Currently steel is being used to test machine stiffness. The results from the steel experiments will allow more accurate strain calculations.
Fig. 7: Stress-strain curves of the first and fourth cycle for Ti-49 at% Ni and Ti-50 at% Ni tapes. The lines indicate values used in the design of the prototype actuator.

The prototype actuators were fabricated using two tape layers with the outside material being super-elastic (50 at% Ni) and the central member being SMA (49 at% Ni). To be able to easily match the forces the total area of the SMA member was made equal to the total area of the super-elastic members (i.e. match the stress and the forces match). Figure 8 shows the make-up of each layer. The design allowed for overlap of the two compositions at the top of the actuator to better amalgamate the two materials and provide good structural support. The shapes were cut from their respective tapes with a scalpel and pressed in a 12 x 20 mm rectangular die. With no support between the members during pressing, the gaps were eliminated as the members were pressed together. This required carefully cutting between the members with a scalpel to ensure the members did not sinter together. The actuators were then sintered according to the previously described steps. The post-sintered dimensions vary from those presented in Figure 8 but the pre-sintered dimensions were used as an approximation. The post-sintered actuators were on average 350 μm thick.

Fig. 8: Pre-sintered dimensions and lay up of monolithic actuator. Light Grey represents superelastic, dark grey represents SMA.

Fig. 9: a) Stainless Steel jig used for annealing, b) Prototype actuator prior to and c) after annealing.
A stainless steel jig was designed to bend the outer members and the central member separately and constrain the material during shape setting. The goal was to have an actuator with a resting position, that when clamped was straight. The stress-strain data of the fourth cycle were used in the design. The lines in Figure 7 display how the radii on the jig were determined. The SMA member was selected to have a strain 1.8%. This middle value was chosen with the goal of visually displaying tip deflection with the naked eye while minimizing fatigue on the members. The strain in the super-elastic member was then determined to be 0.86% by matching the stress value. The equation $\rho = t/2\varepsilon_{\text{max}}$ was used to convert these strains to radii of curvature, where $\rho$ is the radius of curvature, $\varepsilon_{\text{max}}$ is the maximum strain and $t$ is the thickness of the actuator. This results in a $\rho$ of approximately 10 and 20 mm for the SMA member and the super-elastic member respectively. The prototype was annealed at 550 °C for 15 minutes. The jig and a prototype actuator before and after annealing can be seen in Figure 9. It is noted that due to the preliminary tensile data and the approximate actuator dimensions, the actuator is viewed as a proof of concept only. As such, the values presented are nominal and real values may deviate.

The prototype actuator was clamped to an aluminum base and a flame was used to observe the actuation. Figure 10 shows time lapse images of the actuation sequence. It was observed that the actuator was self-resetting, with a tip deflection of approximately 2 mm, or 10% of its 20 mm length.

![Fig. 10: Time lapse images of the actuation sequence showing 2 mm of deflection (the scale is in mm).](image)
CONCLUSION & FUTURE WORK

This paper demonstrates a new approach to the fabrication of monolithic SMA actuators, which takes advantage of powder metallurgy techniques to overcome some of the limitations of traditional machining and local annealing. The actuator was fabricated without machining, using well-established and inexpensive techniques of tape casting followed by reactive sintering. The actuator is a functionally-graded hybrid monolithic structure, with areas of superelastic NiTi alloyed directly to regions of shape-memory NiTi. The proof-of-concept device successfully demonstrated cyclic self-resetting bending motion, with tip deflections of approximately 10% of the device length.

Future work in actuator design will include the development of a more formal design process, following the collection of more precise mechanical test data. Longer-term, we will investigate other geometries, with a goal of developing a library of “active compliant structures” capable of integration into meso-scale devices.

We continue also to develop and refine the PM and sintering processes. The use of polymer in the tape casting process introduces carbon contamination which led to the loss of some Ti to TiC, thus altering the Ti-49 at% Ni (or 50 at% Ni) mixed ratios. DSC traces indicated a shift of transformation temperatures to lower temperatures. The traces indicate that the 50 at% Ni parts would be fully austenitic at room temperature while the 49 at% parts were mixed austenite/martensite. This suggests that with more careful choice of mixtures (e.g., increasing the Ti content in the original mixture), we may be able to achieve improved cyclic deflection by ensuring the shape-memory regions were fully martensitic at room temperature.

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REFERENCES


PRELIMINARY ASSESSMENT OF MANUFACTURING IMPACTS ON DIELECTRIC ELASTOMER ACTUATORS RELIABILITY

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ABSTRACT

Dielectric Elastomer Actuators (DEA) have demonstrated their potential as promising low-cost actuators. However, their commercial venue has, essentially, been limited by their reliability, which is affected by their manufacturing process. This paper investigates how a typical actuator fabrication process influences the life cycle reliability of cone-shaped actuators made of acrylic VHB 4905™. The proposed manufacturing process consists of: (1) film biaxial pre-stretching, using a manual 2-axis pre-stretching apparatus; (2) electrode deposition of conductive ink mixtures by airbrushing, and (3) film-defect detection and covering with Neoprene™ glue.

The uniformity and precision of the film’s biaxial pre-stretch levels are studied through finite element analysis. The effects of electrode deposition techniques on reliability are optimized by measuring the breakdown voltage. The evolution of the film defects is observed through mechanical and electrical cycling tests. These tests are repeated with Neoprene™ glue covering the defects, as it is expected to limit defect propagation and thus increase the film’s life cycle.

Results show that a properly designed multiple grip point pre-stretch jig can offer a uniform and accurate biaxial stretch for maximum actuator reliability and minimal film loss. The initial breakdown voltage is 30% higher when a gap is present between the edge of the electrode and the side of the DEA than when a gap is not present. Mechanical cycling generates defects around the center frame that participate in long-term electrical cycling failure. Neoprene™ is an effective covering agent in high voltage applications but its high stiffness affects the long-term reliability under low electrical cycling.

This paper intends to take a first glance at potential improvements of DEA reliability through its manufacturing techniques and sets the grounds for future in-depth work on the subject.

Keywords: DEA, Manufacturing, Reliability, Cycling, Electrode deposition, Failure
INTRODUCTION

Dielectric Elastomer Actuators (DEAs) are an interesting alternative to conventional electromagnetic technologies in many robotic and mechatronic applications, particularly where high force-to-weight ratios and high strains are required at low cost.

DEAs are governed by electrostatic attraction [1][2]. Oppositely-charged electrodes, created on each side of a soft polymer film, induce an equivalent Maxwell pressure that squeezes the polymer, which in turn, expands in its planar direction (Figure 1a). The two major polymer families used in DEAs are acrylics and silicone [3][4]. This research concentrates on the manufacturing process of DEAs made from two layer laminates of acrylic VHB 4905TM from 3M, since it is the most currently used material for its high energy density [5].

![Fig. 1 a) DEA’s working principle b) Cone-shaped actuator used in the experimental setup](image)

A typical cone-shaped DEA is shown in Figure 1b, where the polymer film is held between two rigid circular frames while the center frame is pushed outwards by a conservative force provided here by a suspended mass [6]. The mass moves down when the film expands in area under an equivalent Maxwell pressure produced by the electrostatic field (1) [2][7][8]:

\[
P_{\text{MAX}} = \varepsilon_{\text{rel}} \varepsilon_{\text{die}} \left( \frac{V}{t} \right)^2
\]

\( \varepsilon_{\text{rel}} \): vacuum permittivity \\
\( \varepsilon_{\text{die}} \): polymer dielectric constant \\
\( V \): voltage \\
\( t \): thickness of the film

DEA assembly has three major steps. First, the film is pre-stretched in its planar direction in order to avoid film buckling during electrostatic compression as well as to increase mechanical efficiency [9][10] and breakdown strength of the film [11][12]. Second, the film is fastened to rigid frames to maintain its initial pre-stretch state. Third, compliant electrodes are applied on each side of the polymer. A common low-cost electrode material used is carbon black powder, which is generally mixed with a solvent and applied by paintbrush or airbrushing technique. This type of electrode is one of the most compliant and low-cost amongst electrodes used in DEA [13].

Three large-scale failure modes of DEAs have been proposed: pull-in instability (aka buckling), dielectric strength failure and material strength failure [12]. Pull-in instability is observable as wrinkles on the film surface [16][17]. Dielectric strength failure occurs when the applied electric voltage exceeds the dielectric strength of the material. Finally, the
material strength failure happens when the material is stretched beyond the unfolded material’s polymer chain length.

Past experience has shown DEA reliability to be greatly affected by actuator manufacturing methods [14][15]. This is an important drawback, since fabricating reliable DEA layers is key to develop high force actuators using 10’s to 100’s of layers.

Since VHB is a commercial adhesive and its quality is not controlled for DEA applications, defects are generally present in the film. One approach to improve reliability through manufacturing is the use of fault-tolerant electrodes [18][19]. Neoprene™ glue can also be used to improve dielectric breakdown strength by covering film imperfections [20].

This paper takes a first look at a typical DEA fabrication process and investigates its impact on the short and long term reliability of acrylic VHB 4905™ actuators. Cone-shaped DEAs are built according to a typical manufacturing process consisting of: (1) film biaxial pre-stretching using a manual 2-axis pre-stretching apparatus, (2) electrode deposition techniques using different air-brushed conductive-ink mixtures and (3) film-defect detection and covering with Neoprene™ glue. A 12-point pre-stretching jig design is presented and validated with Finite Element Analysis. Microscope observations are performed to monitor film evolution during mechanical cycling, while breakdown tests are performed on different electrode configurations. Electrical cycling tests are carried out on DEAs to gather statistical data of actuator failure in normal operating conditions, both with and without Neoprene™ defect covering.

Results show that pre-stretching equipment must be carefully designed to maximize film uniformity and usable film area as demonstrated with a 12-point jig example. Breakdown voltage tests revealed that leaving a gap between the electrode and frame improves breakdown voltages by as much as 30%. Mechanical and electrical cycling at low voltages revealed that most failures occur in the film region where the stretches are the highest and where film buckling is observed. Moreover, these highly-loaded regions show defect-like grooves, nucleating from the film-frame interface, that propagate with cycling. Finally, covering the film defects with Neoprene™ glue potentially improves the average film life by up to a factor of 4 at high electric fields but its high stiffness has negative impacts in long term cyclic tests.

**OBJECTIVES AND METHODOLOGY**

This section describes the objectives and methodology of this research, targeting the impact of the proposed actuator fabrication process on reliability:

**Film pre-stretch**

Actuator reliability in regard to pre-stretch has already been investigated in past research and is deemed critical on actuator performance and efficiency [10][16]. Therefore, a good pre-stretching apparatus must be able to uniformly pre-stretch polymer films over a large area, regardless of the actuator’s configuration; whether the actuators require a symmetrical stretch ($\lambda_x\lambda_y = 3\times3$ for cone actuators [6]) or a non-symmetrical stretch ($\lambda_x\lambda_y = 4\times2.5$ for diamond shaped DEA [20]).

Finite element analysis is used to design and compare two concepts: a 4-grip point machine versus a 12-grip point machine. Note that all films used in this paper where pre-stretched with a 4-grip point machine using only the optimal area of the pre-stretched film.
Mechanical cycling

A MTS 322 Test Frame Fatigue device is used to force DEAs into typical displacement cycles. The defects’ generation and evolution are observable throughout the actuator’s life cycle as no opaque electrode is present. Two imposed displacements are used to match typical displacements under low- and high-voltage operation as used later in electrical cyclic tests. Setup 1 is used to witness the effects of various manufacturing parameters on the reliability of a DEA used in a high-voltage application in a short life period. Setup 2 mimics the displacement of a DEA used in a low-voltage application, which is expected to withstand a long lifetime. Figure 2 illustrates the test parameters of these two setups.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Setup 1</th>
<th>Setup 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage</td>
<td>8.4 kV</td>
<td>6.4 kV</td>
</tr>
<tr>
<td>% of breakdown*</td>
<td>86 %</td>
<td>66 %</td>
</tr>
<tr>
<td>Pre-stretch</td>
<td>3x3</td>
<td>3x3</td>
</tr>
<tr>
<td>Mass</td>
<td>150 g</td>
<td>150 g</td>
</tr>
<tr>
<td>Equivalent pre-load</td>
<td>13 mm</td>
<td>13 mm</td>
</tr>
<tr>
<td>Generated displacement</td>
<td>11 mm</td>
<td>6 mm</td>
</tr>
<tr>
<td>Speed</td>
<td>1 Hz</td>
<td>1 Hz</td>
</tr>
</tbody>
</table>

*based on breakdown results (figure 11)

Tests are paused during cycling for defect visualization and measurement with a 40X microscope equipped with a digital camera in the dust-free environment of a clean bench. Mechanical fatigue tests will oversee the effects of the use of different electrodes’ solvents (ethanol, toluene) and of the Neoprene™ glue.

Dust particles, or contaminants, constitute the most common type of defects on DEA films. They are noticeable as dark and twisted lines, and are often visible to the naked eye under appropriate lighting (Figure 3a). The second known group of defects present in the film are bubble-like defects and scratches, either included in the film or induced by the laminating process [21]. The bubble-like defects are difficult to locate as they result in a faint color variation on the microscope images (Figure 3b).

This work focuses on bubbles-like defects, as they are supposed to have the most damaging effects on the materials’ reliability as they cause a local reduction of the breakdown strength [21]. As well, only defects visible to the naked eye (60μm) will be observed, as they are the easiest to locate in a low-cost manufacturing process.
Electrode deposition

This section compares the influence of two basic electrode mixtures as well as electrode geometry on breakdown voltage.

Paint-brushing is not considered in this study because of its inability to generate smooth edge electrodes in a controllable and precise fashion. Research has shown that electric charges gather at the electrode edges and create unwanted electric field concentration that can cause DEA breakdown at lower voltages [22]. Also, paint-brushing requires a contact with the film, which can be detrimental and even destructive to the film.

Air-brushing is the main technique studied in this work, as it is a fast, low-cost, contact-free method that can be implemented for high-volume manufacturing. Mixtures of carbon-black and ethanol are the main components of this study due to their high compliance and low cost. Toluene effect on breakdown strength is also investigated as it is a necessary chemical of an upcoming inkjet impression system currently in development. Masking of particular film regions will be proposed as a simple method to improve overall breakdown strength. Indeed, this avoids superimposing the maximal electrical field to the maximal stress field near the edges of the frame, where the material is thinner.

Electrical cycling

As an ultimate reliability test, electrical cycling tests are performed on cone-shaped DEAs, using the low- and high-voltage setups of Figure 2. Electrical cycling allows the collection of small statistical data of actuators, built with the simple low-cost production techniques proposed here. A USB-powered DAQ device, controlled by a Labview program, sends software-timed analog voltage updates to a Matsusada Power Supply, while monitoring the current directly from the current monitor output. When a DEA fails, the increase in current is detected and the voltage is rapidly shut, preserving the DEA for analysis of failure location.

The parameters of Setup 1 (see Figure 2) were chosen to maximize the strain output, regardless of the occurrence of buckling. When the film buckles, it undergoes very large deformations to reach a stable state characterized by its wrinkled shape. The tests are done at ~85% of the experimental breakdown voltage. The purpose of Setup 1 is to witness the average improved life-cycle of the actuator used in high-voltage applications when the visible defects are covered with Neoprene™.

Setup 2 is used to obtain the actuator’s maximum displacement without buckling for a given mass and initial pre-stretch. This setup intends to evaluate the long-term life expectancy of the DEAs with and without Neoprene™. Similar to Setup 1, the applied voltage represents ~65% of the breakdown voltage.

Note that in all mechanical and electrical cycling tests, reference DEAs were manufactured from VHB film with no visible defects, while DEAs with defects were manufactured where defects were visible but not overabundant.

RESULTS AND DISCUSSION

Film pre-stretch

The pre-stretch sequence of a 4-point jig (see Figure 4a) was analyzed with Abaqus software, using a non-linear high-strain model for a nominal film pre-stretch of $\lambda_x \times \lambda_y = 4 \times 4$. The material parameters of the incompressible second order Ogden model were obtained from past researches [20]. As shown in Figure 5, the 4-point jig does not generate a large...
uniform film region where the nominal pre-stretch values are precisely met. Only ~13% of the total stretched film can be used. This study also shows that the 4-point jig has a high-tear risk near the corners of the grip, as stretches of up to 6.9 are encountered near the grips (see Figure 5b). Note that the stretch ($\lambda$) is defined as the ratio of the final length over the initial length.

Fig. 4 Film stretching using a) a 4 points jig and b) a 12 points jig

Fig. 5 FEA result of the stretches a) in the x direction b) in the y direction

A uniform stretch can be obtained with a greater area of usable film with a well-designed pre-stretch jig equipped as the multiple mobile grips allows unconstrained displacements in the x direction. For example, a 12-point jig was designed and evaluated by FEA (see Figure 5b). In fact, for a desired stretch of $\lambda_x \lambda_y = 4 \times 4$, the jig can uniformly stretch the film in both directions on 75% of the film’s area (Figure 6). Maximum stretches are also reduced ($\lambda = 5.6$) which significantly reduces tear risks during actuator manufacturing.

Fig. 6 FEA result of the stretches a) in the x direction b) in the y direction
Mechanical cycling

Microscopic observations revealed that the size and shape of film defects (in this case, air bubbles, see Figure 3b) that are initially present remain constant throughout the mechanical life cycle of the DEA. On the other hand, groove-like defects rapidly appear on the film edges around the center piece (Figure 7a); these grooves were also observed in previous work [20]. Experiments show that grooves appear near 1,000 cycles, and expand gradually as cycling continues. The evolution of the cracks is described in Figure 8 for both experimental setups. The groove length doubles when the mechanical displacements doubles, suggesting a linear behaviour between groove propagation and mechanical displacements.

**Fig. 7** a) Defects around the center frame, b) defects around the center frame, with Neoprene™ applied around it and c) defects around a glue drop, all after 10,000 cycles

**Fig. 8** Length of the appearing cracks during mechanical cycling

Mechanical DEA failure was always caused by tearing around the center frame (Figure 9). Note that mechanical cycling showed that the maximal life-cycle of a cone-shaped actuator is over 65,000 cycles, using the parameters of Setup 1 and 22,000 with the parameters of Setup 2.

**Fig. 9** Failure mode of mechanically cycled DEA

Observations tend to demonstrate that the grooves are caused by a film-slippage effect around the center piece of the actuator, induced by a high-stress concentration. As seen in Figure 8, to improve life cycle, a smaller preload displacement could be adapted for smaller strains, implying the usual trade-off between actuator performance and reliability. Note that the geometric ratio between the outer and inner frame can also be adapted to limit the strain of the polymer near the center of the cone-shaped actuator. When Neoprene™ is applied around the center frame of the DEA, grooves propagation is significantly reduced, and after
24,000 cycles, their length reached only ~ 0.09 mm (see Figure 8). However, an adequate glue deposition technique still needs to be developed for precise and repeatable results and no further tests were made with this technique.

Just as the non-covered defects, the Neoprene™ covered defects did not move or grow when submitted to mechanical cycling. However, after ~ 10,000 cycles, small grooves began to appear around the Neoprene™ spots (Figure 7c). Those grooves are similar to the ones observed near the center frame, but are much smaller. They are most likely caused by the higher rigidity of Neoprene™ as opposed to VHB. The Neoprene™ induced grooves limit the long term reliability improvement as it will be shown later in electrical cycling tests of Setup 2.

No obvious difference was recorded through the mechanical cycling of chemically-stressed DEAs, as no defects generation was visible and no actuator failed during the first 25,000 test cycles, for both ethanol and toluene.

**Electrode deposition**

Breakdown tests (Figure 10) have shown a 30% increase in the maximum voltage when a ~1 mm gap is left between the electrode and frame. The gap separates the mechanical stress concentrations at the film edges from the higher electric fields at electrode edges [21]. Note that all breakdowns occurred at the edge of the electrode.

![Fig. 10](image)

**Fig. 10** a) Breakdown test parameters b) Top view of the locations of the breakdowns and average values of voltage recorded

In order to create this 1 mm gap, the mask must overlap the frame by at least ~3 mm, mainly because the mask cannot be set closer to the film without risk of damaging it. This distance can be problematic in the production of miniature actuators. Also, as dust is generated by this deposition technique, it is not optimal for precise manufacturing.

As a potential alternative to airbrush, an inkjet printer is currently being studied. The main problem in the development of this technology is the mixture’s constituent’s polarity difference that causes the carbon power to agglomerate and clog the nozzles of the cartridge, making off-the-shelf inkjet printers unsuitable for printing this kind of ink. Toluene is a more compatible solvent that improves the clogging issues. Breakdown tests (Figure 11) and the mechanical cycling both indicate that the change in solvent does not significantly affect the breakdown and mechanical strength of the DEA.
Fig. 11 a) Breakdown test parameters b) Top view of the locations of the breakdowns and average values of voltage recorded

Electrical cycling

Setup 1 (short term – high voltage): With the parameters of this setup, buckling was severe, as shown by the large central spikes of Figure 12. At such high voltage, the average life cycle is as low as ~1,000 cycles for both reference and Neoprene™-covered samples. The Neoprene™-covered samples lasted slightly longer, while their variance was twice that of the reference sample. This is partially explained by inadequate glue deposition technique, as 2 of the 10 samples failed directly on the covered zone upon initial activation.

In these tests, it is also noted that the majority of the failures occurred on the edge of the buckling region, indicating that buckling is a major failure phenomenon that is yet to be fully understood.

When compared to samples where defects were not covered, Neoprene™ proved to have a significant impact on average durability. In fact, the average life-cycle of DEA with non-covered defects was 309, which is 4 times lower than the DEA with Neoprene™ covered defects, with a standard deviation of 387. These results should be used with care, since failure locations were not identified with absolute certainty and failure may have been unrelated to film defects.

Setup 2 (long term – low voltage): As seen in Figure 13, the average life cycle is higher when no Neoprene™ glue is applied. This suggests that the grooves induced by the Neoprene™ drops, which appeared during mechanical cycling, are a cause of long-term failure. This is not fully corroborated by Figure 13b, since only one failed where Neoprene™ was applied.
Nonetheless, 6 out of the 13 Neoprene™-covered DEAs suffered severe film rupture making impossible the location of the initial failure. These particular DEAs were computed in the calculation of the average life-cycle, but are not shown in the following figure.

![Fig. 13 Position of failure for electric cycling test using Setup 2 for the a) reference test, b) the Neoprene™ covered defects](image)

For the reference tests of Setup 2, the majority of the failures occurred near the center frame (less than 2 mm thereof) as showed in Figure 13a. This corroborates the hypothesis that the defects induced by the mechanical stress (tearing near the center) are linked to the failure of the DEA. In addition, while buckling was thought to be avoided in this test, the center region showed slight buckling occurrences after about ~1,000 cycles (see the central spikes in Figure 13). The large deformations and stresses due to buckling certainly increase failure probability. The time delay in buckling occurrences is explained by the viscoelastic relaxation of the VHB film. Finally, from Figure 8, it was found that the average failure occurs when mechanically-induced center cracks reach a minimal projected width of ~700 µm.

The mechanical cyclic test confirmed that DEA can withstand the cyclic stresses of 22,000 cycles and 67,000 cycles for setup 1 and 2 respectively. The fact that the actuators in the electrical cyclic tests failed before these values indicates that failure is likely due to coupled mechanical-electrical effects. For example, the polymer thins down near the grooves at the center frame, which increases the electrical field, causing an electrical breakdown. As well, Neoprene™ is an effective covering agent in high voltage applications but its high stiffness affects the long term reliability under low electrical cycling as it nucleates groove-like defects.

**CONCLUSION**

The main objective of this research was to take a first glance at the impact of three low-cost manufacturing steps (pre-stretch, electrode deposition, defect covering with Neoprene™) on the reliability of DEAs.

Different pre-stretch strategies were evaluated through FEA film analysis. Results show that a well designed pre-stretch device with mobile attach-points can improve both the stretch uniformity and usable area of the polymer film.

Dielectric breakdown tests have shown that leaving a gap between the electrodes and the frames improves the breakdown voltage of actuators by 30 %, since this action decouples the mechanical stress concentration at the film edge from the high electric field at the electrode edge.
Mechanical cycling tests show that the observable defects in the DEA film (air bubbles and scratches) do not grow throughout cycling, indicating that they have no significant impact on long-term reliability. Mechanical cycling also revealed that hazardous grooves growing with number of cycles appeared near the center frame. These grooves are likely caused by film slippage under the frame due to high local stresses.

Electrical cycling tests at high voltages have shown that covering the defects with Neoprene™ glue can potentially increase the DEA’s life expectancy by a factor of ~4.

Electrical and mechanical failures were found to be closely linked. The presence and growth of the grooves limits film life, as actuators tested in long-term electrical cycling failed at low voltage, almost exclusively near the center. Similar grooves appeared around Neoprene™ glue, which also shorten the life expectancy of the DEAs, suggesting that Neoprene™ is an effective covering agent in high voltage applications but that its high stiffness affects the long term reliability under low voltage electrical cycling. Particular caution should be taken with the electrical cycling results as the failure location technique needs to be further improved, and as more tests should be conducted to confirm the averages and standard deviations presented here.

FUTURE WORK

Film quality is the critical issue that has to be addressed for the expansion of DEA technology. Since the casting process of commercially available films such as VHB cannot be controlled, silicones are presently being studied as a more reliable material. Nonetheless, this material has a small strain, suggesting that new polymers need to be developed for DEAs.

New electrode-deposition techniques using inkjet technology are presently being developed for an accurate deposition of the toluene–carbon-black compliant electrodes. A new defect-covering agent with lower stiffness will also be tested.

REFERENCES

STATIONARY HADAMARD SHUTTER BASED ON SEMICONDUCTOR-TO-METALLIC PHASE TRANSITION OF THERMOCHROMIC W-DOPED VO$_2$ ARRAYS

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ABSTRACT

We have fabricated stationary Hadamard shutter arrays based on the well-known semiconductor-to-metallic phase transition of W-doped VO$_2$ smart coatings. This shutter consists of 16 planar micro-optical slits for which either transmittance or reflectance switching of the individual slit can be controlled by an external voltage to perform any desirable on-off switching combinations. The transmittance switching was as high as 25 dB, while the reflectance switching of the individual slit was about 6 dB at $\lambda = 1.55$ $\mu$m. In addition, the electrotransmittance switching modulation was demonstrated at 1.55 by switching the individual micro-optical slit by an ac voltage. This electrically controllable W-doped VO$_2$ shutter can be used as multi-entrance slits (i.e. a stationary Hadamard shutter) of dispersive IR spectrometers to enhance the signal-to-noise ratio and then improvement of both the sensitivity and the resolutions of these spectrometers.

Keywords: Vanadium dioxide (VO$_2$), Semiconductor-to-metallic phase transition, Smart coatings, Thermochromic, infrared spectrometer, Stationary Hadamard shutter, Micro-optical switches.
INTRODUCTION

Thermochromic vanadium dioxide (VO\textsubscript{2}) smart coatings exhibit a reversible semiconductor-to-metallic phase transition (SMT) from a low-temperature monoclinic phase to a high-temperature tetragonal phase at relatively low transition temperature ($T_t \approx 68$ °C) [1]. This phase transition is accompanied by a strong modification of electrical and optical properties in the infrared region. The $T_t$ can be controlled by metal doping such as Al, W, Mo, Ti, etc. As an example, in W-doped VO\textsubscript{2} thin films, the transition temperature decreases by about 22.85 °C per 1 at. % of W doping [2]. Also, the hysteresis loops which appear in both optical and electrical switching can be completely suppressed by judiciously codoping VO\textsubscript{2} films with W and Ti [3]. In addition, the phase transition of VO\textsubscript{2} can be controlled by external parameters such as temperature, pressure, photo-carriers injection into a VO\textsubscript{2} heterostructure, photo-excitation and an electric-field. VO\textsubscript{2} smart coatings are thus excellent materials for various technological applications such as IR uncooled bolometer, photonic band gap, optical fiber switching devices, ultrafast switching, smart windows, smart radiator devices for spacecraft, and sunshields for spacecraft.

Here, we exploit the transmitting semiconductor (on) to the reflecting metallic (off) phase transition of W(1.4 at. %)-doped VO\textsubscript{2} smart coatings in the fabrication of stationary Hadamard shutter arrays. This shutter consists of 16 planar micro-optical switches for which both the infrared reflectance and transmittance switching of the individual slit can be controlled at room temperature by an external switching voltage (either a dc or ac voltage). This allows performing any desirable on-off switching combinations. Both the electrotransmittance and electroreflectance switching of the individual slit were investigated at $\lambda = 1.55$ μm. In addition, the electrotransmittance switching modulation (on/off) was demonstrated at 1.55 μm by switching the micro-optical slits by an external ac signal. This shutter can be used as individually programmable 16 multi-entrance slits (i.e., stationary Hadamard shutter) instead of the traditional single entrance slit of dispersive IR spectrometers.

EXPERIMENTS

The W-doped VO\textsubscript{2} smart coatings were fabricated onto sapphire substrates by means of reactive pulsed laser deposition (RPLD) using a XeCl excimer laser (308 nm wavelength and 12 ns pulse duration). A highly pure (99.9%) and highly dense W(1.6 at.%)-doped vanadium oxide target was used for this RPLD process. The laser fluence on the target was about 5 J/cm\textsuperscript{2}. The deposition was performed under background gas mixture of reactive oxygen and argon. The details of the fabrication process were reported elsewhere [2]. The optimized deposition parameters for formation of single phase of W-doped VO\textsubscript{2} layers were 100 mtorr for the total pressure and 5% of O\textsubscript{2}:Ar ratio. The deposition temperature was fixed to 500 °C for which the optical contrast switching was much higher than that of the films synthesized at low temperature (300 °C). The thickness of the fabricated W-doped VO\textsubscript{2} layers was about 150 nm. The concentration of W dopant in the fabricated layers was found to be about 1.4 at. % as evaluated from x-ray photoelectron spectroscopy analysis [3]. The infrared transmittance switching of W-doped VO\textsubscript{2} coatings was investigated as a function of temperature by means of Fourier transform infrared spectrometer (FTIR) equipped with controllable heater cell.
The stationary Hadamard shutter consisting of 16 planar micro-optical slits was patterned by the standard photolithography followed by plasma etching, while the Au/NiCr electrodes were achieved on the top of the individual micro-optical slits by means of the lift-off process. Each micro-optical slit is about 56 μm wide by 1000 μm long, while the distance between the micro-slits is about 15 μm. In order to block the light between the VO$_2$-micro-optical slits, a chromium arrays were patterned by a lift-off process onto backside of the device. The fabricated device was packaged and Au wires were sonically bonded to the Au electrode of the shutter.

Fig. 1 shows a microscope photography of the fabricated stationary Hadamard shutter arrays. The W-doped VO$_2$ active micro-optical slits can be switched individually between their transmitting semiconducting (on) and reflecting metallic (off) states by applying an external voltage through the Au/NiCr electrodes.

![Fig. 1. A microscope photography of the stationary Hadamard shutter device consisting of 16 programmable micro-optical slits and based on transmitting semiconducting (on) to the reflecting metallic (off) phase transition of W-doped VO$_2$ active layer. The optical switching (on/off) of each micro-optical slit can be controlled by an external voltage applied through the Au/NiCr electrodes.](image)

The optical response (either transmittance or reflectance switching) of the individual micro-optical slit was investigated at a wavelength of 1.55 μm. Fig. 2 shows the schematic configuration of the experimental optical switching setup. The variable external load resistance ($R_L$) was used in series with the device in order to protect it from the jump of the current induced by the applied voltage.

![Fig. 2. Schematic configuration of the experimental electrotransmittance switching setup.](image)
In this experiment, the laser beam provided by tunable laser, was coupled to an input optical single mode fiber and excited the individual W-doped VO$_2$ micro-optical slit at normal incidence. The transmitted light was collected by an output single mode optical fiber, which was aligned to the input fiber. Then, the transmitted switching light was recorded, as a function of applied voltage, around 1.55 μm by an optical spectrometer analyser (OSA).

Fig. 3 shows the schematic configuration of the experimental electroreflectance switching setup using an optical Y coupler. In this experiment, the probe laser beam at 1.55 μm was coupled in the input optical fiber port of the coupler and excited the single W-doped VO$_2$ micro-optical slit at normal incidence. The reflected light travelled through the central coupling region and collected by the output port of the coupler and then recorded as a function of applied voltage by the OSA.

![Fig. 3. Schematic configuration of the experimental electroreflectance switching setup.](image)

**EXPERIMENTAL RESULTS AND DISCUSSION**

Fig. 4 shows the temperature dependence of the infrared transmittance of as-deposited W-doped VO$_2$ onto sapphire. These spectra were recorded at room temperature and at 50 °C, relatively to the sapphire substrate, in the spectral range between 1500 and 4000 cm$^{-1}$. The transmittance decreases as the temperature increases. Thus results of the phase transition from the transmitting semiconducting (on) to the reflecting metallic (off) state of W-doped VO$_2$ active layer. At room temperature (i.e., in the semiconducting state) the active layer exhibits relatively good IR transmittance ($T_{IR}$) in the large spectral range investigated here ($T_{IR} \approx 43\%$ at 1500 cm$^{-1}$ and $T_{IR} \approx 49\%$ at 4000 cm$^{-1}$). At 50 °C which is much higher than the $T_1 \approx 36$ °C of W(1.4at. %)-doped VO$_2$ (i.e., in the metallic state), the active layer becomes completely opaque and its transmittance drops to 0%. The contrast switching between the transmitting (on) and the completely opaque (off) states is about 45%. Note that this opacity of the active layer in the metallic state (i.e., $T_{IR} = 0\%$) is extremely required for the realization of the infrared stationary hadamard shutter arrays operating in transmission mode.
In order to determine the switching voltage that required to switch on/off the W-doped VO$_2$ arrays, we measured the I-V characteristic of the individual micro-slit by applying a variable dc voltage through the Au/NiCr electrodes and measuring the current flowing in protective external load resistance ($R_L = 4$ k$\Omega$) placed in series with the device (see Fig. 2). Fig. 5 shows the typical I-V characteristic measured in air at room temperature of the individual W-doped VO$_2$ micro-slit. The flowing current increases lightly as the applied dc voltage increases and jumps abruptly (up to about 25 mA) when the threshold voltage is reached ($V_{th} \approx 114$ V).

Fig. 4. Infrared transmittance of the transmitting semiconducting (on) state at room temperature and the reflecting metallic (off) state at 50 °C of W-doped VO$_2$-coated sapphire.

Fig. 5. Typical current-voltage characteristic of individual W-doped VO$_2$ micro-slit.
This result shows clearly that the applied voltage induces the phase transition from the high resistance semiconducting state to the low resistance metallic state. When the phase transition occurs the current uniformly flows in the micro-slit. Note, that $V_{th}$ can be significantly reduced by optimizing the size of the micro-optical slits.

The electrical switching mechanism in VO$_2$ has been subject of intensive experimental and theoretical investigations and still under debate in the literature. Mansingh and Sing [4] observed that the surface temperature of undoped and doped (Ti, Al, and Cr) VO$_2$ single crystals increased by only 3–6 °C when an external electric field induced the switching to the metallic state at temperatures lower than their transition temperatures. These results suggested that the switching is not due to an increase in the temperature of the entire bulk of the VO$_2$ crystals and is due to the formation of a current filament “small channel” at the surface with temperature approaching the transition temperature of the crystals. Recently, to explain the electrical switching in VO$_2$, Boriskov et al. [5] developed a model taking into account both the electron-electron and the electron-phonon interactions and based on the dependence of the carrier switching density on both applied electric field and switching channel temperature. This model described very well their experimental temperature dependence of the current-voltage characteristics of VO$_2$. Kim et al. [6] investigated the SMT by using I-V and micro-Raman measurements as a function of temperature and under applying an external voltage to their planar VO$_2$/Al$_2$O$_3$ device (50×20 μm$^2$). This study demonstrated that the SMT is not accompanied with the structural phase transition from the monoclinic to the tetragonal structure and the conducting filament or current channel was not formed. Thus, suggested that Joule heating is not responsible for the SMT. Okimura and Sakai [7] investigated the time-dependent characteristics of electric-field induced phase transition of planar VO$_2$-micro switch devices. They observed that the VO$_2$ switched from its high resistance state (HRS) to its low resistance state (LRS) under application of an external electric field. The resistance of the LRS was time-independent and current dependent. Also, they observed that the transition state is stable even for a high current. They suggested that the electric field–induced the phase transition of a VO$_2$ can realize a stable resistive switching based on the Mott transition eliminating the Joule heating effect. However, more recently, Lee et al. [8] performed experimentally the time-resolved visualization of the heat flow in VO$_2$/Al$_2$O$_3$. In that experiment, the phase transition was induced by applying a voltage pulse (amplitude of 14 V, 100 μs pulse width, and 10 Hz repetition rate) to the VO$_2$/Al$_2$O$_3$ planar micro-optical switch (15×50 μm$^2$) through the Au/Cr electrodes at temperature of 56 °C. By using synchrotron-based infrared microscopy, they observed that the voltage pulse generated Joule heat around the electrodes and they also recorded the local temperature variation between 56 and 75 °C for which the VO$_2$ channel switched to its metallic state. For our device, since $T_t = 36$ °C of W-doped VO$_2$ is relatively close to room temperature, and according to this recent study [8] we suspect that the switching mechanism is due to the Joule heating.

Using a switching voltage of 125 V, which is relatively high than $V_{th}$ determined before, we performed both electrotransmittance and electroreflectance switching measurements of our device at $\lambda = 1.55$ μm. Fig. 6 shows the typical electrotransmittance switching of the individual W-doped VO$_2$ micro-optical slit. The transmittance decreases as the applied voltage increases. The switching contrast between the on and off states is as high as 25 dB. This electrotransmittance switching behavior was reproducible and completely reversible.
Fig. 6. Typical electrotransmittance switching at $\lambda = 1.55$ $\mu$m of the individual W-doped VO$_2$ micro-slit.

Fig. 7 shows the measured electroreflectance switching on/off of the individual micro-slit. It is observed that the reflectance increases as the applied voltage increases: the W-doped VO$_2$ micro-optical slit switches from its transmitting semiconductor (on) state to its reflective metallic (off) state. The contrast switching between the two states is about 6 dB.

Fig. 7. Electroreflectance switching of single W-doped VO$_2$ micro-optical slit as measured at $\lambda = 1.55$ $\mu$m by means of optical Y coupler.
The electrotransmittance switching modulation (on/off) of the stationary Hadamard shutter was demonstrated at $\lambda = 1.55 \, \mu m$. In this experiment, the individual micro-optical slit was switched on/off by ac switching signal. In this case also a protective load resistance was used in series with the device (see Fig. 2).

The transmitted signal was recorded by digital oscilloscope via optical power sensor (HP-81532A). The applied ac signal with 40 ms of period modulated the charge density around the critical density $N_C$ for which the individual W-doped VO$_2$ micro-optical slit switches from its semiconducting (on) to its metallic (off) states. The W-doped VO$_2$ micro-optical slits were switched on/off about $10^5$ without any deterioration of their performance. Fig. 8 shows an example of the transmittance switching modulation (on-off) of 5 cycles. The rise time is about 14 ms, while the fall time is about 7 ms. Note, however, that these times are limited only by the response time of our optical power sensor.

![Transmitting (on) state](image1)  
![Opaque (off) state](image2)

Fig. 8. Electrotransmittance switching modulation at $\lambda = 1.55 \, \mu m$ of the individual micro-optical slit.

Since W-doped VO$_2$ smart coatings exhibit good optical switching in large infrared spectral range, which is limited only by the characteristics of the substrates (see Fig. 4), the fabricated W-doped VO$_2$-arrays device can be used as stationary Hadamard shutter [10] for dispersive IR spectrometers to increase the signal-to-noise ratio (SNR) and then improve both their sensitivities and their resolutions. For that, this stationary Hadamard shutter (i.e., individually programmable $N$-multi-entrance slits) can replace the single entrance slit of the traditional dispersive IR spectrometers. The optical switching (on/off) of these multi-slits can be controlled individually by an external voltage. Then, any desirable on-off switching combinations can be realized. Thus, the role of this Hadamard shutter will consist of multiplexing the incoming information onto the output detector-element arrays using binary coding. This technique based on Hadamard principle employs binary coding of the multiplexed input signals using $N \times N$ binary Hadamard matrix code.

We will integrate this Hadamard shutter with their 16 electrically controllable micro-optical slits to the miniaturized infrared optical spectrometer (IOSPEC) that has been developed by MPB communications [11]. The experimental results will be reported
elsewhere. For this Hadamard transform-IOSP EC, we anticipate enhancement of SNR by a factor of 2 (i.e., \(\frac{\sqrt{N}}{2}\), where \(N = 16\) slits) according to the Hadamard principle. Thus, enable to use this advanced miniaturized spectrometer in various terrestrial and aerospace applications.

**CONCLUSION**

The well-known semiconductor-to-metallic phase transition characteristic of thermochromic W-doped VO\(_2\) smart coating was exploited in the fabrication of stationary Hadamard shutter arrays operating at room temperature and driven by an external voltage. This shutter consists of 16 planar micro-optical switches for which both the transmittance and reflectance switching can be controlled individually to perform any desirable combinations of switching on-off.

The starting thermochromic W-doped VO\(_2\) active layer was synthesized by reactive pulsed laser deposition onto Al\(_2\)O\(_3\) substrate. The stationary Hadamard shutter was patterned by photolithography followed by plasma etching, while the lift-off process achieved the Au/NiCr electrodes onto the top of the individual planar micro-optical slits. Transmittance switching as high as 25 dB and reflectance switching about 6 dB were achieved for the individual micro-optical slits at \(\lambda = 1.55\) \(\mu\)m. Finally, this device can be used as multi-entrance slits (i.e., stationary Hadamard shutter) to enhance both the sensitivity and the resolution of dispersive infrared spectrometers.

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