EXPERIMENTAL STUDY AND CHARACTERIZATION OF MAGNETOELASTIC RIBBONS AS VIBRATION SENSORS AND THEIR APPLICATION FOR THE IDENTIFICATION OF CRACKS IN CANTILEVER BEAMS THROUGH THE DYNAMIC BEHAVIOR OF THE BEAM

by

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A Dissertation Submitted in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

Department of Chemical Engineering of University of Patras

June 2020
Διδακτορική διατριβή
Υποβληθείσα στο
Τμήμα Χημικών Μηχανικών του Πανεπιστημίου Πατρών
Υπό
Γεώργιο Σαμουργκανίδη

Η ΕΠΙΤΑΜΕΛΗΣ ΕΞΕΤΑΣΤΙΚΗ ΕΠΙΤΡΟΠΗ

Επιβλέπονες
Δημήτριος Κουζόκος. Αναπληρωτής Καθηγητής
Σοφούν Μπουγιώτης. Καθηγητής
Χριστάκης Παπακωστάκης. Καθηγητής
Γεώργιος Παπαράκης. Αναπληρωτής Καθηγητής
Γεώργιος Κυριακού. Αναπληρωτής Καθηγητής
Μανόλης Τοπογλίδης. Επίκουρος Καθηγητής
Γεώργιος Παπανικολάου. Ομότιμος Καθηγητής
...to my parents Rouslan and Liana, and my sister Elli.
...and we’ve reached the end. At the end of a journey that had everything and was full of emotions, colors and thrills. Joy, expectation, hope, frustration, fatigue, satisfaction are just some of them. However, all the bad and gray look so distant and small in front of the sense of accomplishment that I feel as I browse through the thesis that follows.

I would like to start by thanking all the committee members Dimitris Kouzoudis, Soghomon Boghosian, Christakis Paraskeva, Georgios Pasparakis, Georgios Kyriakou, Georgios Papanicolaou and Emmanuel Topoglidis who accepted to supervise my thesis and honored me for attending the presentation, reading and signing my PhD thesis. I think I have to speak separately to committee chairman Dimitri Kouzoudi, an associate professor of the department of Chemical Engineering, who may not have had direct and frequent contact with my course all these years, but his presence in the laboratory and the key questions from time to time in our discussions acted as a point of departure for new questions and answers.

I continue by thanking my very good friend in the lab Konstantino Spiliotopoulos, undergraduate student of the department of Chemical Engineering, who, during the course of his diplomatic project in our laboratory, spent many hours with me in the lab watching and learning with great mood. His company and happy mood helped me a lot in getting my own piece of work, as I spent most of my time working alone in the laboratory. Konstantine, thank you for all the great times we had in the laboratory, with the endless laughter, our successes and failures in the experiments, and the constant urge to not give up.

I would like to thank a very important person of mine, my girlfriend Kallia, for all her advice and support she gave me. She is the only one who has been able
to stand up to my strangeness in these years, and I’m so grateful to her for her patience. She is a very important person for me and I owe her a lot.

I want to thank the engineer of the University of Patras, Gerasimo Diamanti, who besides the many useful constructions for which he devoted time and care, he is to me a very good friend with whom I share a lot in common. I would also like to thank Mr Georgio Papanicolaou, professor of the department of Mechanical Engineering and Aeronautics, for his valuable conversations and comments on this thesis. Also, I couldn’t help but thank the "Andreas Mentzelopoulos scholarships for the University of Patras" that have supported me financially for the last two years, and gave me the opportunity to continue and develop my work.

Finally, I would like to thank deeply my family, my parents and my sister, who in their unique way always supported me in whatever I did.
Abstract

In the current thesis thin magnetoelastic ribbons of metallic glass alloy known as Metglas 2826MB were investigated, characterized and applied as vibration-based structural health monitoring sensors. Such materials have the property of changing their magnetic state (magnetization) when they are stressed mechanically (Villary effect), and vice versa they are stressed mechanically when they are magnetized by an external magnetic field (magnetostriction effect). These materials were used in the form of thin ribbons in contact with a mechanical structure, such as a cantilever beam, as a vibration sensor, in order to monitor the structure’s mechanically health state. The monitoring was established through the detection of the natural frequencies of the mechanical structure.

The study of the thesis is divided into three main parts which are, the ”proof of concept” of the work, the characterization procedure and the application process. As far as the first part is concerned, the ability of the ribbons in sensing and transmitting the vibrational state of a cantilever beam was investigated, as well as the accuracy of the recorded data in detecting the change of the vibrational state of the structure due to damages. To carry out this task, a number of different beam specimens, undamaged and damaged, of aluminum alloy 6063 material were used and the results were compared to computational ones using ANSYS modal analysis. The second part was the characterization of the ribbons as structural vibration sensors and the process involved seven different sensor parameters such as the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. The experiment was accomplished using two different experimental setups, one to examine the frequency response parameter and one to examine the rest of the parameters. The last part included the application of the under consideration vibration sensors to detect and identify cracks in cantilever beams, through a proposed crack identification methodology. The methodology involved the use of a pattern matching process, through a minimization procedure, in order to identify the crack location and depth. Each one of the three parts was examined in detail and thoroughly, with the results of the experiments being properly presented and described.
Acknowledgements

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1.1 Objectives of the thesis

The concept of the current thesis is to introduce a new class of materials in the field of structural vibration sensors. These materials belong to the category of magnetic materials, and more specifically in the field of ferromagnetic materials. They are thin ribbons of amorphous metal alloys, mainly made of iron and nickel, and exhibit magnetoelastic behavior due to their ferromagnetic nature. Their trade name is Metglas and in the current thesis the material used is the Metglas 2826MB series.

Thus, the overall objective of the thesis is to introduce a new non-invasive method for detecting mechanical vibrations based on the magnetoelastic property of these materials, while the specific objective of the thesis is to investigate the aforementioned amorphous ferromagnetic alloy materials for the study and development of a new type of structural vibration sensors, and to use these sensors as a special mean on detecting dynamically and identifying the health state of mechanical structures, such as cantilever beams.
1.2 Thesis structure

The thesis is made up of eight main chapters in total, plus the appendices. The chapter 1 is the thesis introduction and is divided into two sections. The first section discusses the goals of the study, with the basic direction stated in a short and concise manner, while the second section describes the structure of the thesis and the chapters outline.

Chapter 2 is about the theory of magnetism in magnetoelastic materials and consists of four in total sections. In the first section an introduction is made on the fundamentals of magnetism in order to familiarize the reader with the concepts of magnetism. Then, in the second and third section there is an extensive reference to the types of magnetic materials and their interaction with magnetic fields, with a greater emphasis on ferromagnetic materials. Finally, in the fourth section the theory about the properties and behavior of magnetoelastic materials is discussed, as well as the applications of those found in the literature.

Chapter 3 covers the background literature of vibrations sensors and the methods in which they are used in the structural health monitoring (SHM) applications. Totally, four different sections are included in this chapter, with the first being the introduction and the last the bibliography used. The second section describes the current technology in the field of vibration sensors and presents in detail the principle of operation of different types of vibration sensing devices, along with their advantages and disadvantages. The third section relates to the application of vibration sensors in the field of SHM.

Chapter 4 is the beginning of the experimental part of the thesis and includes descriptions of all the experimental configurations and methods used to measure and collect the data. It is divided into three sections. The first section is the introduction of the chapter. The second section discusses the experimental setups and methods related to the proof of concept of the thesis, while the third section presents all the experimental setup and procedures that were followed for the characterization of the sensors.

Chapter 5 is the first chapter of the thesis with experimental results and dis-
cusses the proof of concept of the thesis, which is the usage of magnetoelastic ribbons as structural vibration sensors in detecting the natural frequencies of undamaged and damaged cantilever beams. It is divided into eight different sections, with the first two being the abstract and the introduction of the chapter. The next four sections are related to the procedure that was followed in order to gather and analyze all the experimental and computational data needed, and to compare them with each other in order to extract the results. The final two sections are the conclusions of the chapter along with the bibliography used.

Chapter 6 discusses the results of the characterization experiment mentioned in chapter 4 and it consists of five in total sections. The first two sections are the abstract and the introduction of the chapter, while the last two sections are the conclusions and the bibliography of the chapter, just as in chapter 5. The third section includes all the characterization results and it is divided into seven subsections, one for each sensor parameter.

Chapter 7 contains the results related to the use of the proposed vibration sensor in a vibration-based structural health monitoring application. Specifically, this chapter presents a methodology that takes as an input the vibrational state of a mechanical structure, such as a cantilever beam, and outputs the health state of the structure. It is divided into seven sections, with the first two being the abstract and the introduction, and the last two the conclusions and the bibliography, as before. The in-between three sections consist of the process in which the methodology is structured and verified, with the third section being the modeling of the structure, the fourth section being the simulations of the model and the fifth section being the verification results.

Chapter 8 is the final chapter of thesis and summarizes the most important conclusions of the thesis. Also, thoughts and suggestions for future research are listed as a continuation of the thesis work. The appendices are listed at the end of the chapters and are three in total. Appendix A includes all the MATLAB scripts and functions used to carry out the experiments and simulations of the thesis. In appendix B, all the normalized frequency ratio graphs between the bending modes of the simulated cantilever beam, from the chapter 7.4.2, are presented. Finally,
appendix C includes all the published work in international scientific journals, as part of the preparation of this thesis.
2.1 Fundamentals of magnetism

Although the natural phenomenon of magnetism had been observed thousands of years before our time, the principle of understanding it began only in the mid-18th century, when Michell (1750, England) and Coulomb (1785, France) discovered the law of interaction between the magnetic poles of long bar magnets. They found empirically that the force between two magnetic poles is proportional to the product of their pole strengths, $p$, and inversely proportional to the square of the distance $r$ between them (Eq. 2.1).

\[ F \propto \frac{p_1 p_2}{r^2} \]  

To comprehend the origin of the force, we can think of the first pole generating a magnetic field, $\mathbf{H}$, which in turn exerts a force on the second pole. By convection, the north pole of a bar magnet is the source of the magnetic field where the field lines originate, and the south pole is the sink where the field lines terminate (Fig. 2.1).

The next colossal breakthrough in the history of magnetism took place in Denmark in 1820, when Hans Christian Oersted discovered that a magnetic compass needle is deflected close to an electric current. This was a huge discovery for
mankind as a new science was born which unified two existing sciences, electricity and magnetism, and was called electromagnetism. After Oersted’s discovery, the

French physicist Andre-Marie Ampere began developing a mathematical and physical theory to understand the relationship between electricity and magnetism. He showed experimentally that the magnetic field of a small current loop is identical to that of a small bar magnet, and the north (N) pole of the magnet corresponds to current circulating in a counter-clockwise direction, whereas clockwise current is equivalent to the south (S) pole (Fig. 2.2). Also, Ampere observed that the

magnetic field generated by an electrical circuit depends on both the shape of the circuit and on the amount of current being carried, and gave the famous expression of Ampere’s circuital law (Eq. 2.2), which says that the integral of magnetic field intensity, $H$, along an imaginary closed path is equal to the product of current, $I$, enclosed by the path.

$$\oint_c H \cdot dl = I$$

The units of magnetic field in the cgs system are oersteds (Oe), while in the SI
system the units are amperes per meter (A/m). The hypothesis of Ampere that all magnetic effects are due to current loops, and that the magnetic effects in magnetic materials such as iron are due to so-called ”molecular currents” was remarkably insightful, considering that the electron was discovered much later in 1897 by the British physicist J. J. Thomson.

Nowadays, it is known that magnetic effects are caused by the orbital and spin angular momenta of electrons. A typical example is the Zeeman effect, which was discovered by the Dutch physicist Pieter Zeeman in 1896, and is the change in the absorption spectrum of an atom in the presence of an external static magnetic field. The theoretical explanation of the origin of magnetism is as follows: From classic electromagnetism, the magnetic dipole moment of a circulating current loop is given by the expression

\[ m = IA \]  

(2.3)

where \( I \) is the current and \( A \) is the area of the circulating current loop. Since the current is, by definition, the charge passing from a point per unit time, in the case of an electron orbiting at a distance \( r \) from a nucleus in an atom it equals to the charge of the electron multiplied by its velocity, \( v \), divided by the circumference of the orbit:

\[ I = \frac{ev}{2\pi r} \]  

(2.4)

Here, the \( e = -|e| \) because the charge on the electron is negative and so, the direction of current flow is opposite to that of the electron motion. Substituting Eq. (2.4) in Eq. (2.3) and considering that the area of the orbit is \( A = \pi r^2 \), the magnetic dipole moment can be written as:

\[ m = IA = -\frac{|e|vr}{2} \]  

(2.5)

Eq. (2.5) can be processed further by extracting an expression for the velocity \( v \) of the electron using some basics from classical and quantum mechanics. According to classical mechanics the angular momentum of a cyclical rotating mass is equal to the product of the radius of rotation \( r \) and the linear momentum of the mass
\[ p = m_e v, \] where \( v \) in our case is the equivalent linear (tangential) speed of the electron at the radius \( r \) and \( m_e \) its mass. Also, based on quantum mechanics the components of angular momentum of an electron along a magnetic field depend on the so called magnetic quantum number \( m_l \) and are equal to \( m_l \hbar \), where \( \hbar = h/2\pi \) is Plank’s constant. This number is quantized and allowed to take integer values from \(-l\) to \(+l\), where \( l \) is the orbital quantum number and determines the magnitude of the angular momentum. Values of \( l \) equal to 0,1,2 and 3 correspond respectively to the familiar labels s, p, d, and f for the atomic orbitals. For example, the p orbital, with \( l = 1 \), can have \( m_l \) values of -1, 0, +1. This means that p orbitals can exist with three orientations relative to an externally applied magnetic field (Fig. 2.3).

**Figure 2.3:** Directions of the angular momentum with respect to the magnetic field \( H \) for a p orbital \((l = 1)\)

Thus, the angular momentum projected on the axis of the external magnetic field can be written as:

\[ L = m_e v r = m_l \hbar \]  \hspace{1cm} (2.6)

From Eq. (2.6) the velocity \( v \) can be expressed as:

\[ v = \frac{m_l \hbar}{m_e r} \]  \hspace{1cm} (2.7)

and by substituting it to Eq. (2.5) gives the final expression of the magnetic dipole moment

\[ m = -\frac{|e| \hbar m_l}{2m_e} \]  \hspace{1cm} (2.8)
In terms of energy, a magnetic dipole moment, $m$, in a magnetic field, $H$, has an energy that is given by

$$E = -\mu_o m \cdot H$$

which with the help of Eq. (2.8) can be written as:

$$E = \mu_o \frac{e\hbar m_l}{2m_e} H = \mu_o \mu_B m_l H$$

The quantity $\mu_B = e\hbar/2m_e$ is called the Bohr magneton and is the elementary unit of orbital magnetic moment in an atom. In the SI units system its value is $9.274 \times 10^{-24}$ J/T, while in cgs units system it may written as $\mu_B = e\hbar/2m_ec = 0.927 \times 10^{-20}$ erg/Oe, where $c$ is the speed of light. From Eq. (2.10) it is clear that the energy of an atomic orbital is dependent on the presence of a magnetic field, and it changes by an amount proportional to the angular momentum of the orbital and the applied field strength. The example of the p orbital discussed above is shown in Fig. 2.4. In the absence of an applied magnetic field, $H$, the s and p orbitals each have one energy level. By applying the field, the p level splits on three different energy levels, corresponding to the $m_l$ values of -1, 0 and 1, while the s level remains as it was since the s electrons has no orbital angular momentum, and therefore no orbital magnetic moment. Fig. 2.5 shows interferometer images of the Zeeman effect in cadmium (Cd) vapors corresponding to the p orbital. In Fig. 2.5a the applied magnetic field is zero and the interferometer reveals only...
single lines (circles) from the emission spectra of cadmium. However, with the application of the magnetic field (Fig. 2.5b) each line (circle) splits up into three parts, as with Fig. 2.4.

All the discussion above about the Zeeman effect has to do with the case where the total spin angular momentum of atoms is equal to zero, and it is called the "normal Zeeman effect". However, much more common is the so-called "anomalous Zeeman effect", which gives a more complex arrangement of lines in the spectrum and is the consequence of spin-orbit coupling of electrons in the atoms (a simple and clear discussion can be found in the book [1]).

2.2 Magnetic materials

Modern technology relies heavily on magnetic materials as they are used in a diverse range of applications (creation, distribution and appliance of electricity, data storage, body scanners, implants, electric vehicles, etc) and becoming more important in the development of modern society.

Starting with the basics, when a magnetic material is subjected to an external magnetic field, $H$, the elemental structures of the material react to the appearance of the field. This reaction is called magnetic induction, $B$, and the equation relating $B$ and $H$ in SI units is expressed as:

$$B = \mu_0(H + M)$$  \hspace{1cm} (2.11)
where $\mu_0$ is the permeability of free space and $M$ is the magnetization of the material. The magnetization, $M$, is a property of the material and is defined as the magnetic moment per unit volume ($M = m/V$). It depends on both the individual magnetic moments of the constituent ions, atoms or molecules, and on how these dipole moments interact with each other. The SI units of magnetization and magnetic induction are A/m and Tesla (T), respectively, while the corresponded cgs units are emu/cm$^3$ and gauss (G) ($1\text{ gauss} = 10^{-4}\text{T}$), respectively. The magnetic induction, $B$, is also called magnetic flux density because it is related to another important quantity, the magnetic flux $\Phi$, which is defined as follows: it is the surface integral of the normal component of a magnetic field passing through a unit area (Fig. 2.6).

![Figure 2.6: Magnetic flux passing through a surface S. The $B_\perp$ corresponds to the normal component of the magnetic field to the surface.](image)

In general, the flux density inside a material is different from that outside. This characteristic can sort the magnetic materials into categories. For example, materials which present less flux density inside than the outside are known as diamagnetic materials. Such diamagnetic materials are the bismuth (Bi) and helium (He), which tend to exclude the magnetic field from their interior due to their zero magnetic dipole moment of atoms or ions. Another example are the paramagnetic (e.g Na or Al) and antiferromagnetic (e.g MnO or FeO) materials. In these materials the inside flux density is slightly more than the outside, and the magnetic dipole moment of atoms or ions is not zero. For paramagnets these dipole moments are randomly oriented (Fig. 2.7a), while for antiferromagnets they are ordered antiparallel to each other (Fig. 2.7b). There is also another example of magnetic materials of great technological interest and these are the ferromagnets
and ferrimagnets. In these materials the magnetic flux is very much greater inside than the outside. In ferromagnets, the magnetic dipole moments of the atoms tend to line up in the same direction (Fig. 2.7c). Ferrimagnets on the other hand are somewhat like antiferromagnets but some of the dipole moments, which are facing one way, are larger than the opposite dipoles (Fig. 2.7d), so the material has a net overall magnetic moment.

Each of the aforementioned categories of magnetic materials has its own magnetization curve, which shows the variation of $M$ or $B$ with the applied magnetic field, $H$. The ratio of $M$ to $H$ is called the susceptibility ($\chi=M/H$) and indicates how responsive a material is to an applied magnetic field. The ratio of $B$ to $H$ is called the permeability ($\mu=B/H$) and indicates how permeable the material is to a magnetic field. Materials which tend to concentrate a large amount of flux density in their interior have high permeability. In SI units the permeability is in units of henry/m and the susceptibility is dimensionless. Fig. 2.8 shows the magnetization curves of the magnetic materials mentioned above. In the case of dia-, para- and antiferromagnets (Fig. 2.8a) the $M - H$ curves are linear. For these magnetic materials large fields (25000 Oe = 2.5 T) are required to cause small changes in magnetization, and no magnetization is retained when the applied field is removed. In diamagnets the susceptibility (slope of the $M - H$ curves) is small and negative, and the permeability is slightly less than unity, while in para- and antiferromagnets the susceptibility is small and positive, and the permeability is slightly greater than unity. On the other hand, the magnetization curves of ferro- and ferrimagnets in Fig. 2.8b are completely different from those in Fig. 2.8a. First of all, it can be noticed that the axes scale are different. In this case, a
very large magnetization is obtained on the application of a small magnetic field. Also, the magnetization saturates above a certain applied field and an increase in field causes only a very small change in magnetization. Here, it is clear that susceptibility, $\chi$, and permeability, $\mu$, are large and positive, and are non-linear functions of the applied field.

Another remarkable behavior of ferro-ferrimagnetic materials is that, decreasing the field to zero after saturation does not reduce the magnetization to zero. This phenomenon is known as hysteresis and is very important in technological applications. Fig. 2.9 shows a schematic of a hysteresis loop plotted in $B - H$ axes. The original state of the material is the unmagnetized state and the magnetic induction follows the curve from 0 to $B_s$, as the field is increased in the positive direction. Although the magnetization is constant after saturation (Fig. 2.8b), here
the $B$ continues to increase due to Eq. (2.11). The value of magnetic induction at $B_s$ is called the saturation induction, and the curve of $B$ from the demagnetized state to $B_s$ is called the normal induction curve. When $H$ is reduced to zero, the $B$ decreases from $B_s$ to $B_r$, which is the residual induction or retentivity. This is useful in applications with magnetic memory devices. The reversed field required to reduce the induction to zero is called the coercivity, $H_c$. Depending on the value of $H_c$, ferromagnetic materials are classified as either hard or soft. Hard magnets retains a large fraction of the saturation field when the driving field is removed, and are desirable for permanent magnets, magnetic recording and memory devices. Soft magnets are easy to saturate but also easy to demagnetize, and are desirable for device applications where the minimization of the energy loses are crucial for the optimum operation of the device (e.g transformers).

### 2.3 Ferromagnetic domains

The phenomenon of magnetism in ferromagnetic materials as mentioned above is very strong, and dominates over other magnetic materials. Ferromagnetism is similar to paramagnetism except that there is a strong quantum interaction (exchange interaction) between the atoms which tends to align electron spins, and hence their magnetic dipole moments parallel to each other, even in the absence of an external magnetic field. This orientation does not appear uniformly throughout the material but separately in areas known as magnetic domains (Fig. 2.10). A

![Figure 2.10:](image.png)

**Figure 2.10:** (a) Magnetic domains in a single grain (outlined with a black line) of steel photographed under Kerr-effect microscope. The arrows show the direction of magnetization in each domain. (b) Schematically representation of two magnetic domains.

magnetic domain is an area of a material in which the magnetization of the atoms
is uniform and with a common direction. Therefore, in each domain the oriented magnetization of all its atoms can be replaced by a single magnetization vector \( M_d \). Most ferromagnetic materials are not magnetized due to the interplay between the magnetization of the individual domains. This condition is also defined as the minimum energy state in the material. However, in the presence of an external magnetic field, the magnetic domains tend to be oriented, thereby increasing the total magnetic energy in the material. While in the diamagnetic and paramagnetic materials the orientation of the individual magnetizations, in the presence of an external field, is weak and non-permanent, in the ferromagnetic materials the orientation is almost complete and permanent.

The exchange energy provides a strong driving force for parallel alignment, therefore ferromagnetic materials should be expected to consist of one domain with all dipoles aligned in the same direction. However, although a single domain would certainly minimize the exchange contribution to the total energy, there is a number of other contributions to the total magnetic energy of a ferromagnet. The formation of domains allows a ferromagnetic material to minimize its total energy, of which the exchange energy is just one component. The other main contributors to the magnetic energy are the magnetostatic energy, which is the principal driving force for domain formation, and the magnetocrystalline and magnetostrictive energies, which influence the shape and size of domains.

### 2.3.1 Magnetostatic energy

A magnetized block of ferromagnetic material containing a single domain has a macroscopic magnetization. The magnetization causes the block to behave as a magnet, with a magnetic field around it. Fig. 2.11a illustrates a magnetized block with its associated external field. From the figure it is apparent that the field acts to magnetize the block in the opposite direction to its own magnetization. For this reason it is called the demagnetizing field, \( H_d \). The demagnetizing field causes a magnetostatic energy which depends on the shape of the ferromagnetic material. It is this magnetostatic energy which allows the block to do work such as lifting another ferromagnet against the force of gravity.
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The magnetostatic energy can be reduced by reducing the external demagnetizing field, and one way to do this is to divide the block into domains, as shown in Fig. 2.11b. Here the external field is lower, so the block is capable of doing less work, and (conversely) is storing less magnetostatic energy. To reduce the magnetostatic energy to zero there should not be magnetic poles at the surface of the block, and one way to achieve this is shown in Fig. 2.11c. However, the magnetic moments at the boundary between the two domains are not able to align parallel, so the formation of domains increases the exchange energy of the block.

2.3.2 Magnetocrystalline energy

The magnetization in ferromagnetic crystals tends to align along certain preferred crystallographic directions. The preferred directions are called the "easy" axes, since it is easiest to magnetize a demagnetized sample to saturation if the external field is applied along a preferred direction. Fig. 2.12a show the magnetization curves of the iron single crystal with the field applied along the easy, medium and hard axes (Fig. 2.12b shows schematically the easy, medium and hard magnetization directions in a unit cell of bcc iron). Since bcc iron is a cubic crystal, all six cube edge orientations (⟨100⟩, ⟨010⟩, ⟨001⟩, ⟨100⟩, ⟨010⟩ and ⟨001⟩) are in fact equivalent easy axes. The body diagonal (⟨111⟩) is the hard axis of magnetization, and other orientations, such as the face diagonal (⟨110⟩), are intermediate. In all cases the same saturation magnetization is achieved, but a much larger applied field is required to reach saturation along the medium and hard axes than along the easy axis.
The phenomenon that causes the magnetization to align itself along a preferred crystallographic direction is the magnetocrystalline anisotropy. The crystal is higher in energy when the magnetization points along the hard direction than along the easy direction, and the energy difference between samples magnetized along easy and hard directions is called the magnetocrystalline anisotropy energy. In fact the area between the hard and easy magnetization curves of iron in Fig. 2.12a is a measure of the magnetocrystalline energy. To minimize the magnetocrystalline energy, domains will form so that their magnetizations point along easy crystallographic directions. For example, the vertical axis in Fig. 2.11 should correspond to a cube edge in bcc iron. Because of the cubic symmetry the horizontal direction is also an easy axis for bcc iron, therefore the domain arrangement shown in Fig. 2.11c has a low magnetocrystalline energy. The horizontal domains at the top and bottom of the crystal in Fig. 2.11c are called "domains of closure" and they form readily when a material has easy axes perpendicular to each other. In such materials this configuration is particularly favorable because it eliminates the demagnetizing field, and hence the magnetostatic energy without increasing the magnetocrystalline anisotropy energy. One more point to note is that, the magnetocrystalline energy clearly affects the structure of the domain boundaries. Within the region between domains the direction of magnetization changes, and therefore cannot be aligned along an easy direction. So, like the exchange energy,
the magnetocrystalline energy prefers large domains with few boundaries.

2.3.3 Magnetostrictive energy

When a ferromagnetic material is magnetized it undergoes a change in length known as its magnetostriction. Some materials, such as iron, elongate along the direction of magnetization and are said to have a positive magnetostriction. Others, such as nickel, contract and have negative magnetostriction. The length changes are very small (tens of parts per million) but do influence the domain structure.

In iron, magnetostriction causes the triangular domains of closure to try to elongate horizontally, whereas the long vertical domains try to elongate vertically, as shown in Fig. 2.13a. Clearly the horizontal and vertical domains cannot elongate at the same time, and instead an elastic strain energy term is added to the total energy. The elastic energy is proportional to the volume of the domains of closure and can be lowered by reducing the size of the closure domains, which in turn requires smaller primary domains. Of course making smaller domains introduces additional domain walls, and the corresponding increase in exchange and magnetostatic energy. The total energy is reduced by a compromise domain arrangement such as that shown in Fig. 2.13b.

Figure 2.13: Effects of magnetostriction in domain structures in bcc iron: (a) Formation with high magnetostriction energy and (b) Formation with low magnetostriction energy.
2.3.4 Domain walls

The boundaries between adjacent domains in bulk ferromagnetic materials are called domain walls or Bloch walls. The thickness of the wall is very small (compared to the domain size) and across this distance the direction of magnetization changes usually by either 90 or 180 degrees. The width of domains walls is again determined by a balance between competing energy contributions. The exchange energy is optimized if adjacent magnetic moments are parallel, or as close to parallel as possible to each other. This favors wide walls, so that the change in angle of the moments between adjacent planes of atoms can be as small as possible. However the magnetocrystalline anisotropy is optimized if the moments are aligned as closely as possible to the easy axes. This favors narrow walls with sharp transition between the domains, so that few moments have unfavorable crystalline alignment in the transition region. In practice a compromise is reached which minimizes the total energy across the boundary.

![Figure 2.14](image)

**Figure 2.14:** (a) Change in orientation of the magnetic dipoles in a Bloch wall (180° twist boundary), (b) Change in orientation of the magnetic dipoles in a 90° tilt boundary.

The most energetically favorable types of domain wall are those which do not produce magnetic poles within the material, and therefore don’t introduce demagnetizing fields. One such is the twist boundary illustrated for a 180° boundary in Fig. 2.14a. Here the magnetization perpendicular to the boundary rotates gradually out of the wall plane. Also, stable are 90° tilt boundaries, as shown in Fig. 2.14b. The magnetic moments rotate through the wall in such way that they make a constant angle of 45° with both the wall normal and the surface. Another
kind of domain wall, called a Néel wall, occurs in thin films of magnetic materials because they are energetically more favorable. In Néel walls the spins rotate around an axis normal to the surface of the film rather than around an axis normal to the domain wall. The spin rotation in a Néel wall is shown in Fig. 2.15.

![Figure 2.15: Rotation of the atomic dipoles in a Néel wall.](image)

### 2.4 Magnetoelastic materials

The magnetic and elastic properties of ferromagnetic materials, as mentioned in Section 2.3.3, depend on each other. The different couplings between these properties are called magnetoelastic effects. These effects can be separated into two main categories, namely direct and inverse effects. One direct effect that is common and occurs in these materials is the magnetostriction, which is divided into isotropic and anisotropic, while one inverse effect, that is also common, is the Villary effect.

#### 2.4.1 Magnetostriction

The magnetostriction itself, isotropic and anisotropic, is the phenomenon by which ferromagnetic materials deform due to magnetic interactions that can be either within the sample itself (spontaneous magnetostriction) or caused as a consequence of an external magnetic field (forced magnetostriction), and is related to the rotation of the non-spherical (elliptical) atoms of the material [2]. In the case of isotropic magnetostriction the deformation occurs in all directions and on average is the same, while in the case of anisotropic magnetostriction the deformation
occurs in a specific direction, usually caused by an external magnetic field. The magnetoelastic material used in this thesis (Metglas) is an amorphous metallic alloy of iron and nickel mostly, and has an isotropic magnetostriction.

When magnetoelastic materials are cooled down from a high temperature through their Curie point (the Curie point is defined as the temperature above which a ferromagnetic substance loses its ferromagnetism and becomes paramagnetic) an anomalous expansion is observed near this point. This is the aspect of the spontaneous magnetostriction. Fig. 2.16 is a graphical representation of spontaneous magnetostriction. A single magnetic domain of a magnetoelastic material has a magnetization \( M_d = 0 \) above its Curie point, because of the randomly oriented atomic dipoles (Fig. 2.16a). In this situation the thermal energy of the domain prevails over exchange energy. When the material temperature drops below the Curie point the magnetic domain gets spontaneous magnetized with the alignment of the magnetic dipoles. Due to the alignment of the magnetic dipoles the domain is lengthened by \( \Delta L = L - L_o \), while at the same time contracting to the other two directions.

In the case of the forced magnetostriction an external magnetic field applied to a magnetoelastic material, which temperature is below Curie point, tends to align all the magnetic domains along its direction. Since the domains are already elongated due to their spontaneous magnetostriction, they are further elongated to the direction of the magnetic field. Fig. 2.17 shows schematically the forced
magnetostriction. Dotted lines symbolize the original shape of the domain, before applying the magnetic field. As the magnetic field strengthens all individual magnetic dipoles of the domain rotate until they are perfectly aligned with the field. This happens to all domains and the result is the total elongation of the magnetoelastic material. Magnetostriction is measured in ppm (parts per million) and symbolized by the letter $\lambda$, which is equal to $\lambda = (L-L_0)/L$. When all magnetic domains inside the magnetoelastic material are aligned to the magnetic field, saturation occurs. At this situation the magnetostriction is maximum and called saturation magnetostriction ($\lambda_s$). The magnetoelastic material used in this thesis has a saturation magnetostriction of $\lambda_s = 12$ ppm.

![Figure 2.17: Schematic representation of the forced magnetostriction in a magnetic domain with increasing applied magnetic field.](image)

### 2.4.2 Villary effect

In addition to the magnetostriction effect, the magnetoelastic materials also exhibit the Villary effect according to which, when external mechanical stress is applied to the material this causes a change in its magnetic state [3]. The Villary effect is essentially the opposite of the magnetostriction effect. Thus, while the latter expresses the change in the dimensions of the material due to an external magnetic field, $H$, Villary effect describes the change in the magnetization of the magnetoelastic material due to the application of external mechanical stress, $\sigma$.

Under a given uni-axial mechanical stress $\sigma$, the flux density $B$ for a given magnetizing field strength $H$ may increase or decrease. The way in which a material responds to stresses depends on its saturation magnetostriction $\lambda_s$. For
this analysis, compressive stresses $\sigma$ are considered as negative, whereas tensile stresses are positive. This means that, when the product $\sigma \lambda_s$ is positive the flux density $B$ increases under stress. On the other hand, when the product $\sigma \lambda_s$ is negative the flux density $B$ decreases under stress [4]. In the case of a single stress $\sigma$ acting upon a single magnetic domain, the magnetic strain energy density $E_\sigma$ can be expressed as [5]:

$$E_\sigma = \frac{3}{2} \lambda_s \sin^2 \theta$$

(2.12)

where $\theta$ is the angle between the saturation magnetization and the direction of stress. When $\lambda_s$ and $\sigma$ are both positive (like in iron under tension) the energy is minimum for $\theta = 0$, i.e. then tension is aligned with the saturation magnetization. Consequently, the magnetization is increased by tension.

Fig. 2.18 summarizes the two aforementioned effects for the geometry of the samples used in this thesis which have the form of thin ribbons of magnetoelastic material. In the case of thin ferromagnetic ribbons the non-magnetized state can occur in several ways, however the most likely and dominant is the one shown in the Fig. 2.18a, because it minimizes the field lines outside the material and therefore the system has the smallest possible energy. Applying a magnetic field $H$ or a mechanical stress $\sigma$ to the material has the same effect: Changing its magnetic state in such a way that the magnetization of the individual domains
follows these changes (Figs. 2.18b and 2.18c). The close relationship between the elastic and the magnetic properties of these materials, through the magnetization and deformation, led to the term "magnetoelastic materials”.

2.4.3 Metglas

Metglas is a thin amorphous metal alloy ribbon produced by using rapid solidification cooling process of approximately 1.000.000 °C/s. This rapid solidification creates unique ferromagnetic properties that allows the ribbon to be magnetized and de-magnetized quickly and effectively with very low core losses and a maximum relative permeability. Metglas is based on technology developed at AlliedSignal research facilities in Morristown, New Jersey and Vacuumschmelze in Hanau, Germany. The development of amorphous metals began in 1970 and over the years many new alloys have been found, using the same principles of rapid solidification.

The magnetoelastic material used in this thesis, as vibration sensor, is the Metglas 2826MB series (Table 2.1 lists all the physical and geometric properties of this material, as provided by the manufacturer). It is a soft ferromagnetic alloy, mainly

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Saturation Induction (T)</td>
<td>0.88</td>
</tr>
<tr>
<td>Annealed permeability</td>
<td>800,000</td>
</tr>
<tr>
<td>As-cast permeability</td>
<td>&gt;50,000</td>
</tr>
<tr>
<td>Saturation magnetostriction (ppm)</td>
<td>12</td>
</tr>
<tr>
<td>Electrical resistivity (μΩ·cm)</td>
<td>138</td>
</tr>
<tr>
<td>Curie Temperature (°C)</td>
<td>353</td>
</tr>
<tr>
<td>Width (mm)</td>
<td>6</td>
</tr>
<tr>
<td>Density (g/cm³)</td>
<td>7.9</td>
</tr>
<tr>
<td>Vicker’s hardness (50g Load)</td>
<td>740</td>
</tr>
<tr>
<td>Tensile strength (GPa)</td>
<td>1-2</td>
</tr>
<tr>
<td>Elastic modulus (GPa)</td>
<td>100-110</td>
</tr>
<tr>
<td>Lamination Factor (%)</td>
<td>&gt;75</td>
</tr>
<tr>
<td>Thermal expansion (ppm/°C)</td>
<td>11.7</td>
</tr>
<tr>
<td>Crystallization temperature (°C)</td>
<td>410</td>
</tr>
<tr>
<td>Continuous Service Temp (°C)</td>
<td>125</td>
</tr>
</tbody>
</table>

made of iron (Fe) and nickel (Ni), with average stoichiometry Fe_{40}Ni_{38}Mo_{4}B_{18}. It is manufactured by Metglas Inc., in Conway, SC, in the form of a long thin ribbon, 29 μm thick and 6 mm width, while its manufacturing way is based on a
rapid quenching technique [6]. It has a medium saturation induction, relatively low magnetostriction and high corrosion resistance, and it is mostly used in shielding and high frequency cores applications, as well as field and magnetomechanical sensors. It can be also annealed for high permeability, round or square BH loop.

2.4.4 Applications of magnetoelastic sensors

Physical parameters such as elasticity, dimensions, mass loading etc. affect the resonance conditions of magnetoelastic sensors [7]. Therefore, recording of the resonance of the sensor can give information about the changes in the environment around it. For example, previous works has shown that the resonance frequency of a magnetoelastic sensor responds to changes in atmospheric pressure [8, 9], in temperature [10, 11], in density and viscosity of liquids [12–15]. Mass charging has also been found to alter the resonance frequency of the sensor, and in conjunction with sensitive layers, chemical sensors can be created, which can detect changes in the pH of liquids [16, 17], the concentration of CO$_2$ [18], NH$_3$ [19], glucose [20], microorganisms [21–23] and humidity [9, 10, 24]. Because the intensity of the applied magnetic field also affects the resonance frequency of the magnetoelastic sensor, special care was given during the aforementioned studies in order to keep it constant. It is worth noting that, most of the aforementioned applications were carried out with Metglas magnetoelastic material, as used in the present thesis.

The resonance frequency of a magnetoelastic sensor is just one of its parameters that can be used to detect changes in its environment. The Q factor, the resonance amplitude, or the phase at a particular frequency can also be used for the same purpose. However, the reason why the resonance frequency is the one used primarily as a measured quantity is that its value does not depend on signal strength and orientation. The value of the resonance frequency depends on the shape of the sensor, the external excitation and the type of the oscillation, which is usually identified with the first longitudinal mechanical harmonic frequency due to its magnetoelastic character. The sensor alignment and the applied field intensity determine the oscillation amplitude of the sensor but they do not affect the type of the oscillation. Therefore, the resonance frequency changes can be
detected and correlated with environmental conditions without having to worry about the position or orientation of the sensor. In applications where the location or orientation of the sensor play a role, quantities other than the frequency of the sensor must be used.

The main advantages of using magnetoelastic sensors are their low cost, the absence of electrical connections between the transmitter and the receiver, and their good mechanical properties due to their metallic nature. Apart from the aforementioned cases in which the magnetoelastic materials are used to detect the environmental changes, they also have been used as magnetic field detectors and position sensors. The above applications are based on the $\Delta E/E$ phenomenon, which changes the sensor resonance frequency in the presence of an magnetic field of variable intensity [25]. The position can be determined if the object has a permanent magnet near it, and therefore the characteristics change every time it moves [26].

Similar sensors can be used as magnetostrictive delay lines (MDLs) [27, 28] which are made of magnetostrictive materials with small coils wrapped around them to trigger and detect a pulse traveling along the material. The wave and its characteristics change under the appearance of a mass load between the two coils, or the presence of external magnetic field. The deposition of sensitive and thin layers between the transmitters improves their function as Surface Acoustic Wave (SAW) sensors [29]. SAW sensors can detect position and displacement in one or three dimensions, tensile or compressive stresses, force and pressure in one and two dimensions, as well as the distribution of magnetic field intensity on flat and cylindrical surfaces.

All the aforementioned applications of the magnetoelastic materials as sensing devices focus on the response of the material when it is triggered by an alternating magnetic field. Here, in this thesis the study focuses mainly on the exact opposite behavior, which is the response of the magnetoelastic material (Metglas) when it is triggered by a vibrating mechanical stress field. In this direction there have been few studies that showed the applicability of magnetoelastic materials as mechanical vibration sensors, and can be found in these references [30, 31].
2.5 Bibliography


3.1 Introduction

In the broadest definition, a sensor is a device, module, machine, or subsystem whose purpose is to detect physical parameters in its environment, such as temperature, humidity, motion, vibration etc., and send the information to other electronics, frequently a computer processor. As far as vibration is concerned, vibration is one of the most popular phenomena that exists in our daily life and it is generated as a result of mechanical disturbances from sources such as music-sound, engines, wind and others. Detection of vibration is an important sensor technology for monitoring the operation of machines and mechanical structures such as engines, buildings and bridges, to sense and predict natural disasters like earthquakes and more. Vibration measurements usually include parameters such as vibration displacement, velocity and acceleration, and the device that converge these parameters into a electrical signal is called vibration sensor. These parameters are mathematically related (velocity is the first derivative of the displacement and acceleration is the first derivative of the velocity) and can be measured by a variety of vibration sensors, based on the frequencies of interest and the signal levels involved.
3.2 Current vibration sensing technologies

In this section, several types of vibration sensor technologies are described which are widely used both in industry and in research. The basic measurement principle of a vibration sensor is shown in Fig. 3.1 and it is divided into three parts which are the vibration source (object), the sensing mechanism (vibration sensor) and the signal analysis technique. The sensor detects the vibration parameter of an object, through its mechanical structure, and converts this parameter into an electrical signal by some physical effect in order to convert a non-electrical signal to an electrical signal.

As it has been mentioned, there are different types of vibration sensors and can be separated into displacement (amplitude), velocity and acceleration sensors, according to the measured vibration parameters. Because the displacement, the velocity and the acceleration can be translated into each other in the way of simple calculus, any of these three different types of sensors can be used equivalently. For example, the output signal of an accelerometer can be doubly integrated to obtain the relative displacement. However, in most of the cases it is inadvisable because of significant low frequency instability associated with the integration process. Currently, according to different methods of detecting vibration, vibration sensing devices with different kinds of physical effects are invented, and can be separated in the following categories which will be presented in the next subsections.
• Piezoelectric sensors

• Magnetic sensors

• Capacitive sensors

• Inductive sensors

• Laser Doppler Vibrometers (LDV)

3.2.1 Piezoelectric sensors

A piezoelectric sensor is one of the most common sensors used for vibration detection. It is based on the piezoelectric effect that some piezoelectric crystals exhibit, which is the generation of voltage or charge when they are stressed mechanically (the Greek root word “piezein” means “to squeeze”). Motion in the axial direction stresses the crystal due to the inertial force of the crystal-mass and produces a signal proportional to acceleration of that mass. This small acceleration signal can be amplified for acceleration measurements or converted (electronically integrated) within the sensor into a velocity or displacement signal. This is commonly referred to as the ICP (Integrated Circuit Piezoelectric) type sensor. Most industrial piezoelectric sensors used in vibration monitoring today contain internal amplifiers.

The two basic piezoelectric materials used in vibration sensors today are synthetic piezoelectric ceramics and quartz. While both are adequate for successful vibration sensor design, differences in their properties allow for flexible designs. For example, modern “tailored” piezoceramic materials have better charge sensitivity than natural piezoelectric quartz materials. Most vibration sensor manufacturers now use piezoceramic materials developed specifically for sensor applications. Special formulations yield optimized characteristics to provide accurate data in extreme operating environments. The exceptionally high output sensitivity of piezoceramic material allows the design of sensors with increased frequency response when compared to quartz. Much has been said of the thermal response
of quartz versus piezoceramics. Both quartz and piezoceramics exhibit an output during a temperature transient (pyroelectric effect) when the material is not mounted within a sensor housing. Although this effect is much lower in quartz than in piezoceramics, when properly mounted within a sensor housing the elements are isolated from fast thermal transients. The difference in materials then becomes insignificant. The dominant thermal signals are caused by metal case expansion strains reaching the base of the crystal. These erroneous signals are then a function of the mechanical design rather than sensing material (quartz or piezoceramic). Proper sensor designs isolate strains and minimize thermally induced signals.

Fig. 3.2 shows the most common structure of a piezoelectric sensor along with a real picture of such a sensor. It main parts are the piezoelectric material, which is usually a synthetic ceramic, and the positive and negative electrodes (electrode and metal plate, respectively). The advantages of piezoelectric sensors are mainly the wide frequency range, high sensitivity, accuracy in phase measurements, high signal to noise ratio (SNR), reliability, and the simplicity and light weight of the structure. The disadvantages include the resonant frequency, which alters the stability of the sensor’s frequency response, vulnerable to interference from the external environment, high output impedance and the non-moving parts. The last disadvantage is mostly related with their use in structures where there are moving-rotating parts (e.g. rotating shafts of a motor).

Figure 3.2: (a) Schematic of the principle structure of a piezoelectric sensor, (b) A real piezoelectric sensor.
3.2.2 Magnetic sensors

Magnetic sensors are also known as electric sensors because they transform the mechanical vibration energy into electrical energy through the Faraday’s law of induction. Faraday’s law of induction (briefly, Faraday’s law) is a basic law of electromagnetism predicting how a magnetic field will interact with an electric circuit to produce an electromotive force (EMF), a phenomenon called electromagnetic induction. It is the fundamental operating principle of transformers, inductors, and many types of electrical motors, generators and solenoids [4]. According to this law, the electrical voltage or EMF that is induced in a coil is given by \( V = -N \frac{d\Phi}{dt} \), where \( N \) is the number of turns of the coil and \( \Phi \) is the magnetic flux that passes through the coil. Thus, this law states that the measure of the induced voltage on the coil is proportional to the \( N \) turns of the coil and the rate of the magnetic flux change through the coil. The minus symbol refers to the fact that the induced voltage tends to create a current that will produce a magnetic field opposite to the existing one, in the case of a close circuit.

The operating principle of magnetic sensors is shown in Fig. 3.3a. Typically, a cylindrical magnet (Fig. 3.3b) is mounted on the surface of a vibrating object, close to which there is a solenoid coil. As the object vibrates, it transmits the vibration to the magnet which in turn changes the magnetic flux through the coil, thus inducing a voltage on the coil. This voltage can be detected at the electrodes of the coil by connecting them to any kind of oscilloscope. The advantages of magnetic vibration sensors are mostly the large output signal, the simple post-
processing circuit, the high anti-interference ability and the non-contact parts between the transmitter (magnet) and the receiver (coil). The disadvantages are mainly related to their sensitivity in high frequency vibrations, due to the very small vibration amplitude at these frequencies. However, the vibration sensor used in this thesis is in the category of magnetic vibration sensors, and have showed a very good and stable behavior in high frequencies, mostly due to its magnetoelastic character [5].

3.2.3 Capacitive sensors

Capacitive sensors can be also used as vibration sensors. These devices convert the vibration parameter into electrical capacity changes, then convert these changes into voltage or current using the principle of a capacitor. According to this principle, the capacitance of a conductor increases appreciably when a grounded conductor is brought near it. In the field of mechanical vibrations, capacitive sensors generally are divided into two types, the variable-clearance type and the variable-area type.

In Fig. 3.4 it can be seen the principle operation of a variable-clearance capacitive sensor. Two metallic plates, one movable (plate 2) and one fixed (plate 1), are at a distance d from each other and the movable plate is in touch with a vibrating object. When the vibration of the object cause the moving plate to move up at a distance $\Delta d$, the capacitance increment is given from the following equation:

$$
\Delta C = \frac{\varepsilon A}{d - \Delta d} - \frac{\varepsilon A}{d} = C_o \frac{\Delta d}{d - \Delta d}
$$

(3.1)
where \( \epsilon \) is the dielectric constant between the two plates, \( A \) is the surface area of the two plates and \( C_o \) is the initial capacitance value when both plates are at a distant \( d \) from each other. As it can be seen from the above equation, the change of the capacitance relates to the displacement of the moving plate and when \( \Delta d \ll d \), \( \Delta C \) is approximately linear with \( \Delta d \). So, by measuring the changes of capacitance, the vibration displacement of the object can be detected.

The variable-area capacitive sensor, on the other hand, has a different principle of operation. Fig. 3.5 shows schematically the structure diagram of this type of capacitive sensors. In these sensors the movable plate instead of moving vertically it is moving horizontally, and thus the movement changes the relative surface area between the two plates. When the relative surface area between the two plates changes at a value of \( \Delta A \), the capacitance change is given as:

\[
\Delta C = \epsilon \frac{\Delta A}{d}
\]  

(3.2)

Seen from the above equation, the capacitance \( \Delta C \) is linear with the relative surface area \( \Delta A \). Usually, variable-clearance type capacitive sensors are used for linear vibration displacement measurements, while the variable-area type sensors for torsional angular displacement measurements.

The prominent features of the capacitive sensors include high resolution and precision, short dynamic response time, suitable for online dynamic measurements and non-contact measurements [6]. The disadvantages include mostly the small frequency range, high output impedance, parasitic capacitance, low-grade anti-
jamming ability and vulnerability to electromagnetic interference [7]. However, with the in-depth research of capacitive sensor measurement principle and structure, and the development of new circuits, materials and processes, some of their shortages gradually are being overcame. The accuracy and stability of capacitive sensors are increasing, and more and more design engineers are prefer them over others due to their versatility, reliability and robustness.

### 3.2.4 Inductive sensors

Inductive sensors are an electronic proximity sensors which detect metallic vibrating objects without being in contact with them. Their principle of operation is based on electromagnetic induction, by using self-inductance or mutual inductance coils to achieve detecting the electrical signal through vibrations. An inductor develops a magnetic field when a current flows through it, and alternatively, a current will flow through a circuit containing an inductor when the magnetic field through it changes. This effect can be used to detect metallic objects that interact with a magnetic field.

Fig. 3.6 shows schematically the principle structure of an inductive sensor. The sensor consists of an induction loop or detector coil which is wounded around a high magnetic permeability core, such as a ferrite ceramic rod. As the coil is connected to a capacitance and an AC source, to form a tuned frequency oscillator, the produced AC magnetic field from the coil induces eddy currents in proximal conductors (target). Eddy currents are closed loops of electrical currents induced

![Figure 3.6: Principle structured of an inductive vibration sensor.](image-url)

Figure 3.6: Principle structured of an inductive vibration sensor.
within conductors with the application of an external AC magnetic field. The closer the target is and the greater its conductivity (metals are good conductors, for example), the greater the induced eddy currents are. Thus, as the metallic object vibrates, it periodically affects the size and the frequency of the coil’s magnetic field, by altering its mutual inductance, thereby allowing vibration to be detected. A change in oscillation magnitude may be detected with a simple amplitude modulation detector, like a diode that passes the peak voltage value to a small filter to produce a reflective DC voltage value, while a frequency change may be detected by one of several kinds frequency discriminator circuits, like a phase lock loop detector, to see in what direction and how much the frequency shifts. Either the magnitude change or the amount of frequency change, can serve to define the vibration state of the metallic object.

The prominent features of inductive sensors include the high range of sensitivities and output powers, the contactless nature of the detection process and their use in wet or dirty environments, due to the non-interaction of substances such as liquids or some kinds of dirt with magnetic fields. The main disadvantage of these sensors is that they cannot be used in detection of non-metallic objects.

3.2.5 Laser Doppler Vibrometers (LDV)

The LDV is an interferometric device which measures the absolute velocity of a vibrating body through the measurement of the Doppler shift of scattered laser light coming from the target. It works on laser Doppler principle, in which a frequency-modulated coherent laser beam is reflected from a vibrating surface, and Doppler shift, $f_{shift}$ of the reflected beam is compared with the reference beam, $f_{ref}$. A Doppler signal processor demodulates the photo detector signal, which generates a time resolved velocity of the vibrating body [8]. These techniques are capable of high accuracy and sensitivity, and considered to be a unique instrument for repeatable vibration measurements to overcome the various problems, such as frequency response, intrusivity etc. These instruments are used for modal analysis measurement, high temperature surfaces and noisy environments or to analyze small structures [9, 10]. Laser Doppler Vibrometer is used for observing changes
in mode shapes and natural frequencies for various types of damages such as cut-out, impact and delamination in graphite-epoxy composite beams. The result is virtually independent of the transmitter/receiver motion due to the occurrence of the frequency shift at reflection, so it measures the absolute rather than relative velocity.

These vibrometers require less data storage and processing as compared with a full field measuring instruments like real time and time-average holography. The disadvantages include “speckle drop out” (the speckle noise can distort LDV signal), line of sight required from the laser head to the target, the time required for testing and demodulation of data [11]. The improvements in speckle noise interference would be necessary for a successful application to actual field structures [12]. This technique has a great advantage that the measurement point can be changed easily and quickly by deflecting the laser beam without loading the target object. These have a very limited application in structural health monitoring (SHM) due to its expensiveness. At present, due to its bulkiness and non-portability, these are unsuitable for intermittent monitoring. Currently, they are not really a viable option for regular condition monitoring, even though they are used in production quality control measurements [13].

3.3 Vibration-based structural health monitoring (SHM)

Mechanical systems with reciprocating or rotating parts create vibrations due to mechanical disturbances from various sources such as engines, sound, noise, etc. In mechanical engineering, the term “vibration” is often reserved for systems that can undergo an oscillatory motion freely without applied forces. Vibration is one of the characteristics of mechanical machinery that if uncontrolled, can cause minor or serious performance, operational or safety problems. Machine vibration is a key element for structural health monitoring (SHM), so there is massive interest in acquiring, analyzing and quantifying it for improving reliability,
life, quality control, productivity and safety against catastrophic failure. Although there are various causes of vibration problems, only about 90% of them arise due to unbalancing or misaligning of rotating parts [14].

As it has been mentioned before, vibration can be characterized in terms of three parameters: displacement, velocity and acceleration. The sensitivity of sensors used for measuring these parameters varies with the frequency of the vibration. The general selection guideline is to use displacement sensors to pick up low frequency signals, velocity sensors in the middle ranges, and accelerometers at higher frequencies [15]. Machine tool structural components, such as gateways, bearings and ball screws are liable to deterioration in their performance level with respect to time due to gradual wear and tear, aging, unbalance, looseness of parts, etc. On a regular basis, the testing of such components is needed to reduce the risk of failures and machinery breakdown. The present global market competition has attracted the attention of manufacturing industries towards the monitoring of manufacturing processes and improved equipment conditions.

3.3.1 Vibration as a diagnostic tool

Diagnosis is the art of identifying machine condition from its signs or symptoms to determine its “cause and effect”. It is usually used for monitoring, detection and analysis of machine condition during an operation. Vibration signatures of the machine can offer an early warning to the operator for time based maintenance or to make a crucial decision before any serious problem or unscheduled downtime. The amplitude of the vibration signature gives an indication of the severity of the problem, whilst the frequency can indicate the source of the defect [16]. The extraction of these signals can be considered as a valuable diagnostic tool to predict run-up failures of the machine components. It is a very challenging task to extract the feature from the acquired signal as interferences are occurring due to the presence of noise. The need arises for developing a processing and analysis system of high level data to provide veracious extraction of data about the health of the machine. Various feature extraction techniques like statistical domain, frequency domain, time domain and time-frequency domain are used for obtaining diagnostic
CHAPTER 3. VIBRATION SENSORS AND STRUCTURAL HEALTH MONITORING

Vibration analysis is a very powerful and reliable technique for monitoring the operating conditions of the machine. It is becoming more famous and familiar in industry due to non-destructive in nature and allows sustainable monitoring without any interfering in the process.

3.3.2 Introduction to SHM

The environment of machining operations changes unpredictably and thus implementing a well consulted SHM is an imperative need to raise structural safety, reduce maintenance cost and avoid human and economic losses. SHM is a continuous and an autonomous tool for measuring the various parameters (i.e. vibration, performance, bearing temperature, etc.) to diagnose the real time condition of the distinct components of the structural and mechanical systems which is very useful for improving structural models. SHM can be classified as online and offline. In the long term, SHM provides current information from time to time about the machine’s ability to perform the intended operations in light of unavoidable failures and degradation from the working place. The SHM [17] consists of three steps which are the signal monitoring, the signal processing and the data interpretation used in damage identification system, as shown in Fig. 3.7.

Signal processing techniques includes statistical time series models, Fourier transform, short time Fourier transform, Cohen’s class, wavelet transform, Hilbert – Huang transform, whereas data interpretation includes artificial neural networks, fuzzy logic, support vector machine, Bayesian classifiers, Hybrid classifiers. SHM is also categorized into local and global methods analogous to perform a human
health checkup [18]. This classification is commonly established on the relation of the wavelength of the experimented signals with respect to the defect size as well as to the whole structure dimensions. SHM technology can also contribute to intelligent aircraft structures for reducing its weight and maintenance [19]. They showed that SHM is the imitation of human nervous system. A popular approach to SHM is the recording of modal parameters (natural frequencies, mode shapes, mode shape curvatures, flexibility, and changes in strain energy distributions etc.) which are functions of the physical properties of the structure. So, any alterations in these properties (mass, stiffness, damping) due to boundary conditions, damage, or other internal defects cause variations in modal parameters of the structure. Natural frequencies are most widely used parameters in vibration based SHM, but alterations in these frequencies are not much valuable to determine the damages if the measurements are not very accurate or large levels of damage have appeared [20]. The upgradation in sensor technology for vibration measurement is required to minimize the drawbacks of mode shape parameters that require a single excitation point and a roving exciter for the measurement [21]. The changes in the mode shape curvatures are more suitable for damage detection due to their higher sensitivity in respect to the damage location [22].

The vibration-based SHM has become a significant and fast growing research domain in different fields such as mechanical engineering, aeronautics and civil engineering to assess the real time dynamic characteristics of a structure. A crack may be undetectable at early stages by visual inspection techniques because it doesn’t make changes in properties of the adjacent material, but only changes in geometry with respect to the undamaged state. For example, current observations such as, visual inspection, non-destructive techniques for stress corrosion and fatigue cracks are performed in the US Navy’s P-3 Orion aircraft [23], and a fatigue life enhancement program has been employed to prolong its life for 10 years. It is estimated that nearly half of all operating costs of machining operations can be assigned to repair and maintenance. This is also an ample motivation to study the unmanned SHM that makes the manufacturing industries to operate at low cost with greater customer satisfaction. It is anticipated that an accurate and
CHAPTER 3. VIBRATION SENSORS AND STRUCTURAL HEALTH MONITORING

reliable SHM systems could result in an increase in cutting speeds, reduction in downtime and an overall increase in savings between 10 and 40% [24]. The basic objective of SHM is to determine the location, size, existence and type of damage, and prognosis. In general, a SHM system is used for fault detection, breakdown anticipation and problem diagnostics in a mechanical structure. A bathtub curve describes the life of a machine as vibration levels are used to indicate the failure of a machine [25]. Sometimes, the deterioration in machine takes place so rapidly that there is hardly any gap between fault detection and total failure [26]. In such cases continuous or real time monitoring with automatic shutdown is needed.

3.3.3 SHM methods

The SHM methods can be divided into two types: the model based and the feature based methods [27]. Starting with the model-based methods, which are also called failure detection methods, involve finding a model in which monitoring is carried out by either the variations in model parameters, such as damping ratio or predict errors in the model outputs to reveal defects. These methods correlate the observational signatures of abnormal structures with analytical or quantitative models to detect the damage parameters, and also need a precise computational model. Various models such as auto-regressive and moving average (ARMA) model, hidden Markov model (HMM), artificial neural network (ANN) etc. are falling into this category.

Feature-based methods, on the other hand, are performed in two steps, one is feature extraction from the signal and another is the decision-making based on these features. Various features can be used, including:

1. Time-domain features such as mean, standard deviation, range, root mean squares, skewness, kurtosis, crest factor, etc.

2. Frequency-domain features such as frequencies, damping ratios, energy in different frequency bands, etc.

3. Spatial domain features
4. Time-frequency domain features such as time-frequency distribution.

The extraction of features using signal processing is considered as one of the major elements and challenging aspect of vibration based machine condition monitoring (MCM). The aim is to accentuate features from the sensed signals, which are usually noisy and complex and have to be further processed to yield the features of our interest. The selection of the signal processing method is paramount to effectively identify and diagnose defects that are indicative of potential machine failure. Due to the complex processes involved in the structural response to dynamic loading, a significant research effort has been dedicated towards health monitoring of a machine with the emergence of a broad range of methods, techniques and algorithms. The various methods based on level of identification are classified to extract the features for obtaining the current health of a machine [28, 29].

Level 1: To determine the presence of damage in structure.

Level 2: To determine the geometric location of the damage.

Level 3: To quantify the severity of the damage.

Level 4: To predict the remaining service life of the structure.

Level 5: Self healing structures.

Park et al. [29] suggests ‘level 5’ in the context of smart structures with emergence of shape memory alloys i.e. Cu-Al-Ni and Ni-Ti (NiTi) alloys. The ultimate aim of SHM is to predict the remaining service life of the damage. Too much noise in the vibration signals makes it impossible to extract the useful information through signal processing techniques so adaptive noise cancellation (ANC) is one such technique that enhances the signal-to-noise ratio [30]. The various vibration transducers, like accelerometer, measure raw data obtained from the structure in time domain. Although, the time domain analysis is direct and provide useful information, still has a limitation that it doesn’t provide clear information about all the frequencies present in the response of the structure. Fast Fourier transform
(FFT), a real time analysis is a method to perform quick transformations of functions from time domain to frequency domain or vice versa. The further analysis is often undertaken to obtain modal domain data from the frequency domain data. Now, modern techniques are also available to directly convert the time domain data to modal domain data [31]. During the conversion between the domains, some compression of the data takes place, but for linear systems, there is a little loss of information between the time and frequency domains [32]. Currently, signal processing methods based on modal domain are widely discussed in the literature because mode shapes, modal damping factors and natural frequencies are more easily interpreted than features extracted from the time domain (e.g. residuals of auto-regressive model) and frequency domain (e.g. the distortion identification functions) [22].

### 3.3.4 Signal processing and feature extraction techniques

Obviously, there are no universal physical variables and signal processing techniques that would be appropriate for all applications, and also it is not practically possible to express all signal processing methods for various applications [33]. To keep it general, this section summarizes some of the most common ones.

#### Statistical Time Series Models

The earliest used models in SHM are time series (TS) models which are developed as an approximate mathematical model based on a set of input-output measurements. They are classified into two categories, first, the linear statistical TS model (e.g. auto-regressive model, moving-average model, auto-regressive moving average etc.) and second, the nonlinear statistical TS models (e.g. non-linear auto-regressive with exogenous inputs, non-linear auto-regressive moving average with exogenous inputs) [34]. In 1951, Peter Whittle introduced the general ARMA model which is considered as a time series model based condition monitoring technique carried out by either the variations in model parameters, such as damping ratio or predict errors in the model outputs to reveal defects. The time series anal-
ysis of vibration signals was used to diagnose the damages in mechanical structures [35]. These are among the most utilized methods for parametric representations of a signal and condition assessment of structures under dynamic loading. The time series analysis of acoustic emission is also performed to monitor the wear of cutting tool [36]. It is employed to extract the wear sensitive features for the estimation of tool major flank wear in a turning process and establish the direct relation to the natural frequency of the tool/holder set by developing a metric calculated based eigenvalue system [37]. It provides a measure of the wear accelerating stage and the region using eigenvalues and dispersion ratios. ARMA is successfully accepted to analyze and forecast the rotational speed signal of aero-engine and also to predict the value transformation trend of signal [38]. ARMA can’t model noise contaminated measured structural response and can’t model problems like non-linear behavior of civil structures [17]. Hence, Statistical non-linear TS models have been proposed to overcome the statistical linear TS models.

**Probability Distribution and Density Function**

It is a way of characterizing random signal to distribute the value at any arbitrarily chosen time. It can be expressed in probability distribution, i.e. the probability of its variable is less than or equal to a specified value [39]. The probability function is a non-decreasing function, and it lies between 0 and 1. Its variation with variable is described in [40]. The probability density function (PDF) is used to obtain the different statistical parameters of a signal by taking various moments for mean value, \( \bar{x} \):

\[
\bar{x} = \int_{-\infty}^{+\infty} xp(x)dx \quad (3.3)
\]

The first moment defines the ‘center of gravity’, which occurs when the area under the curve is unity from Eq. (3.3). Second moment defines the variance about the mean value, and it corresponds to the ‘moment of inertia’. The third moment defines the ‘skewness’ (S), which is zero for symmetrical functions and large for asymmetrical functions. The fourth moment defines ‘kurtosis’ (K) that measures spiky or impulsive signals. The statistical parameter ‘kurtosis’ is used
for predicting the condition of rolling element bearing by measuring the bearing housing vibration [41].

**Fast Fourier Transform (FFT)**

During vibrations, the vibration signal produced by a machine component, which consists of certain frequencies, does not change, although their levels may change from one location to another. Frequency analysis of the vibration signal is widely used to diagnose the machine faults [27]. One way of analysis is the forward or inverse Fourier Transform (FT), a frequency domain representation that estimates the strength of different frequency components (the power spectrum) of a time-domain signal. The forward FT is used to convert the signal from time to frequency domain, while the inverse FT is used to convert the signal from frequency to time domain. However, the Fast Fourier Transform (FFT) is considered as more effective and efficient diagnosis technique to obtain the FT of discretized time signal. This signal is considered for a finite time called the “frame” or “time window”, which is then digitized and stored for feature extraction. The selection of an appropriate sampling rate is important for signal digitization to avoid false frequency components that take place due to aliasing [26]. According to the Nyquist theorem, the sampling frequency should be at least twice of the maximum frequency present in the signal. FFT has been used in various types of structures for damage detection such as pipes, scaled bridge model made of three steel girders subjected to impact test, identification of a 3D truss-type structures subjected to earthquakes and forced excitation [42–44].

In a real environment, the majority of the signals is non-stationary, i.e. the spectra vary with time. During the working mode of machines, the vibrations generated have a different and distinct frequencies at each instant of time [45]. The FFT has significant limitations that it can’t depict the change in frequency signals (vibration, speech or biomedical signals) content over the time, also can’t be used for monitoring real structures subjected to dynamic excitations [34]. So the joint time-frequency analysis technique is used to characterize a signal simultaneously into a two-dimensional representation (time vs. frequency), by comparing
them with elementary functions such as the frequency modulated Gaussian functions [46]. The major difference between each time-frequency method is to handle the problem of uncertainty. The uncertainty principle states that “one can’t simultaneously have good frequency resolution and good time resolution” [47]. To overcome the time information loss problem of FFT, three signal processing techniques are described: the short-time Fourier transform (STFT), the Wigner-Ville distribution (WVD), and the wavelet transform (WT).

**Short-Time Fourier Transform (STFT)**

This technique is introduced to overcome the problems of FFT analysis and it is an extension of FFT capability of analyzing non-stationary or noisy signals. The basic idea is to divide the initial signal into small time windows and apply the Fourier transform to each time segment for representing the variation in signal frequency content over time that existed in that segment. It has been used in various applications for e.g. to estimate the modal parameters of a 3D truss-type structures, 7-story RC frame, 3-story 3D steel frame and beams [42, 48–50]. It provides constant absolute bandwidth analysis to identify harmonic components and offers constant resolution in two-dimensional representation, irrespective of the actual frequency [51]. The size of the window determines the accuracy of time, with the larger the window is, the poorer the time resolution, the better the frequency resolution and vice versa.

The STFT has a limitation of low-resolution problem. This problem is resulted choosing a small window which doesn’t permit the dynamic transient behavior of the structure adequately. Also, it doesn’t allow observing two closed natural frequencies [52].

**Wigner-Ville Distribution (WVD)**

The WVD is one of the most general time-frequency analysis techniques, seems to violate the uncertainty principle to provide excellent resolution for accurate examination in both domains, which leads to use this method for monitoring the nonlinear behavior of civil structures. In 1932, Eugene Wigner developed the
original Wigner distribution to study the problem of statistical equilibrium in quantum mechanics. After 15 years, Jean Ville modified the Wigner distribution and first introduced it in signal analysis to eliminate interference between the frequency components. Now, in the field of signal processing it is commonly known as Wigner-Ville Distribution.

Cohen’s class (CC) distribution is a technique to evaluate the energy of time-varying systems [53]. The WVD is one of the so-called ‘Cohen’s class’ of time frequency distributions [47]. The WVD is also applied based on empirical mode decomposition (EMDs) for the fault diagnosis of bearing [54]. Although the WVD has advantages of simplicity, excellent accuracy, effectiveness, no requirement of window function for its analysis, better resolution than STFT, and invariant to window effects, it also has drawbacks of cross-term interference (i.e. generation of spurious frequency not contained in the original signal). Choi-William [55] developed alternative kernels to overcome this problem, but at the cost of lower resolution and an increase in computational resources. Various smoothing techniques are suggested to retain optimum resolution by first locating the interference components, and then withdraw from the unsmoothed WVD [56]. A smooth pseudo WVD is used to monitor the condition of the gearbox by detecting the broken tooth in a spur gear [57]. WVD also suggested to spread the energy of the frequency components by using a Gaussian window [58]. It is also used for contrasting the acoustic and vibration signals to detect the failures in gear, and compare the time-frequency feature of cracked rotor [59, 60].

**Wavelet Transform (WT)**

In 1982, Jean Morlet introduced the idea of fine grained time-frequency analysis approach to achieve an optimal balance between frequency resolution and time resolution for seismic wave analysis. In STFT, complex cosine and sine functions are used to map the signal from time to frequency domain, whereas WT consists of a family of elementary functions (as a function of time) that can be independently dilated and shifted, known as wavelets. Discrete wavelet transform (DWT) and continuous wavelet transform (CWT) are two methods to calculate the wavelet
transform of a signal. In DWT, the scaling factor is chosen as a power of two and it is generally implemented through a pair of low-pass and high-pass wavelet filters [61]. In CWT, the scaling factor can be chosen arbitrarily or by means of convolution.

This method is more sensitive to stiffness variation as compared to WVD [60], and also a crack can be easily found if the unbalance increases significantly. WT gives a better time localization at high frequencies and leads to analysis with constant percentage (or relative) bandwidth, which makes WT more effective than STFT. WT has great potential for detecting abrupt changes in a signal or locate a sudden change in response from an acceleration time response. WT has been used extensively in different fields such as mechanical engineering, transportation engineering, biomedical engineering, power engineering, robotics, earthquake, structural vibration control, image processing, and due to its advantage of noise elimination, data compression and computational efficiency [34]. Complex harmonic wavelets (typically of one-octave bandwidth) which are particularly useful for de-noising purposes in the frequency domain, are infinite in the time domain due to their narrowness and the appearance of windowed sinusoids (harmonic functions) [62]. Various wavelets are used for SHM [34]. Complex wavelets have the advantage of non-sensitivity to the phasing of the event being transformed, because the imaginary part of the wavelet is orthogonal to the real part (sine rather than cosine).
3.4 Bibliography


Chapter 4

Experimental setups and methods

4.1 Introduction

This chapter presents the experimental part of the thesis and contains descriptions of all the experimental setups and methods used to measure and collect the data. It is structured in two different sections, each with a different experimental setup and method. The first section presents the devices and methods used to test the proof of concept of the thesis, which is the usage of magnetoeelastic ribbons as vibration sensors in order to detect the natural frequencies of mechanical structures, such as cantilever beams, for damage detection. The second section presents the devices and methods used to characterize these sensors for structural health monitoring applications (SHM), based on the dynamic behavior of the structure.

4.2 Proof of concept experiment

The heart of the presented technique is the detection of the natural frequencies of a mechanical structure by metglas ribbons. In the current thesis, a cantilever beam (CB) was chosen as a mechanical structure, due to its simplicity and usefulness. Shown in Fig. 4.1 is the experimental setup which detects the natural frequencies of the beam and includes, among other things, a hydraulic press, a homemade
solenoid detection coil, an arbitrary waveform generator, a linear power amplifier, a homemade mechanical stimulator and a laptop. The hydraulic press is a 12-ton press which was used to firmly tighten the one end of the beam so as to achieve the fixed-end boundary condition of a CB. The other end of the beam, which had two attached metglas ribbons on it, was free to vibrate and it was located inside the detection coil, which was connected to the sound card of the laptop. Lastly, there is a beam stimulator, which consisted of a sharp metallic edge attached to a magnetic actuator and was driven by a waveform generator, through a power amplifier, and it was used to excite mechanically the CB.

Figure 4.1: Experimental setup for the detection of the natural frequencies of the CB.

For this experiment a total number of 31 test-beams were used, out of which, 1 was crack free and the other 30 had cracks at fixed positions and with fixed depths (Fig. 4.2a and Fig. 4.2b). Cracks were chosen to be at locations 50 mm, 100 mm, 150 mm, 200 mm, 250 mm and 300 mm from the fixed end and the crack depth for each location was varied from 1 mm to 5 mm with a step of 1 mm. The beam material is aluminum alloy 6063 and all test-beams were cut at same dimensions by CNC machinery (the physical and geometric properties of the test-beams are given in Table 4.1). Each test-beam included two equal-length magnetoeelastic ribbons of material Metglas 2826MB attached on it by double-sided tape (Fig. 4.2a). The ribbons were attached using a 3M\textsuperscript{TM} VHB\textsuperscript{TM} 4930 double-sided tape (Table 4.2 shows the physical properties of the tape).
CHAPTER 4. EXPERIMENTAL SETUPS AND METHODS

Figure 4.2: (a) The undamaged test-beam with two ribbons of Metglas2826MB attached on it, (b) All damaged test-beams together as they were prepared for experiment.

Table 4.1: Physical and geometrical properties of the aluminum alloy 6063 material test-beam.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>350 mm free + 50 mm fixed</td>
</tr>
<tr>
<td>Width</td>
<td>30.25 mm</td>
</tr>
<tr>
<td>Height</td>
<td>8.18 mm</td>
</tr>
<tr>
<td>Density</td>
<td>2.69 gr/cm$^3$</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>68.3 GPa</td>
</tr>
</tbody>
</table>

Table 4.2: Physical and geometrical properties of 3M$TM$ VHB$TM$ 4930 double-sided tape.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adhesive Type</td>
<td>Modified Acrylic Adhesive</td>
</tr>
<tr>
<td>Tape Thickness</td>
<td>0.64 mm</td>
</tr>
<tr>
<td>Peel Adhesion</td>
<td>350 N/mm</td>
</tr>
<tr>
<td>Normal Tensile</td>
<td>1.1 MPa</td>
</tr>
<tr>
<td>Dynamic Shear</td>
<td>620 kPa</td>
</tr>
<tr>
<td>Static Shear</td>
<td>1.5 kg (22 °C)</td>
</tr>
</tbody>
</table>

The sticking of the ribbons was carried out in the following steps:

1. In order to align the two ribbons, the setup of Fig. 4.3a was used which consists of three parts, the alignment beam, the spacer and the magnetic tape. The alignment beam together with the spacer were used to maintain both ribbons parallel to each other and at a certain distance, while the magnetic tape aimed to flatten the ribbons in order to make the alignment procedure easier.

2. A piece of the double-sided tape was carefully attached over the ribbons and let for few seconds to stick well to them (Fig. 4.3b).
3. The tape was carefully de-attached so as to have the two ribbons stick on its back side (Fig. 4.3c).

4. Finally, the tape with the two ribbons on it were attached on the CB so as to have the ribbons parallel to its long axis. The excess tape was stripped thoroughly using a strong plastic blade to prevent the beam from scratching (Fig. 4.3d).

![Figure 4.3: (a) Ribbons alignment setup, (b) A piece of a double-sided tape attached over the ribbons, (c) Two aligned ribbons ready to be attached on the test-beam, (d) Attached metglas ribbons on the one end of the test-beam.](image)

The reason of using two ribbons instead of one is that the detectable signal was sufficiently strong, without the need for further amplification. Having the two ribbons parallel to each other and along the beam’s length causes them to vibrate in-phase during bending oscillations, which were the kind of oscillations studied in this thesis, thus to avoid any anti-phase effects that could cause a reduction to the recorded signal.

Following the ribbons attachment on the beam, the beams were placed to the experimental setup. In order to achieve the CB fixed end boundary condition,
two craved plates (Fig. 4.4a) composed of aluminum alloy 7075 were used to fix one end of the test-beam (Fig. 4.4b). The geometrical properties of the plates are shown in Fig. 4.4c, with L = 50 mm is the beam’s fixed length, leaving out a free length of 350 mm to vibrate. The total internal opening of the carving inside the joined plates is 2H = 7 mm (3.5 mm on each side) while the beams have a thickness of 8.18 mm which means that a gap of 1.18 mm appears between the plates which allows the application of stress by the press. The clamping process took place in 3 steps:

1. Centering the beam plates on the cylindrical platens of the press.

2. Closing the expansion valve of the press.

3. Clamping the beam at the desired pressure using the compression lever.

Figure 4.4: (a) Aluminum alloy 7075 plates with interior carving in order to clamp the test-beams, (b) A clamped test-beam, (c) Geometrical properties of the curved plates [H = 3.5 mm, L = 50 mm, W = 30.25 mm].

Once the test-beam was clamped, a detection solenoid coil was placed at the free end of it using a stand. Table 4.3 shows the physical and geometrical properties of the coil. Fig. 4.5 shows the schematic of the setup. The principle of operation of this experiment is the following: The actuator is excited once by a pulse (50 V - 500 ms), produced from the generator and amplified by the amplifier, which activates the metallic edge to move vertically and stimulate the CB. This mechanical stimulation forces the CB into free vibrations. The metglas ribbons attached at the free end of the CB follow the vibrations and due to their magnetoelastic character, their magnetic state (magnetization) continuously changes. The change of the ribbon’s magnetization produces an AC magnetic field around them, which in turn can induce an AC electric voltage to a nearby coil due to

60
Faraday’s law of induction. The sound card of a laptop is used as a Data Acquisition System (DAQ), through the mic input, so as to record this AC voltage. The data process in the current thesis was done by Matlab software and the algorithms used can be found in the appendix A.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coil Diameter</td>
<td>60 mm</td>
</tr>
<tr>
<td>Wire Diameter</td>
<td>0.16 mm</td>
</tr>
<tr>
<td>Tread Number</td>
<td>1475</td>
</tr>
<tr>
<td>Wire Material</td>
<td>Cu</td>
</tr>
<tr>
<td>Electrical Resistance</td>
<td>232 Ω</td>
</tr>
<tr>
<td>Coefficient of Inductance</td>
<td>0.215 H</td>
</tr>
</tbody>
</table>

**Figure 4.5:** Schematic diagram of the experimental setup.

### 4.3 Characterization experiment

The characterization of the metglas ribbons as vibration sensors was accomplished using two different experimental setups. One setup was to characterize the frequency response parameter, while the other setup was to characterize the rest of the parameters, such as linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. From now on, the experimental setup for measuring the frequency response will be referred as the FR (Frequency Response) setup, whereas the setup for measuring the rest of the parameters will be referred as the NFR (Non-Frequency Response) setup.
4.3.1 The FR experiment

Fig. 4.6a shows the FR setup with all the devices used, while Fig. 4.6b presents schematically the connections among the devices. A homemade single layer solenoid detection coil was placed in the center of a pair of homemade Helmholtz coils (Table 4.4 contains the physical and geometrical properties of the coils). The electrodes of the detection coil were connected to a digital oscilloscope which monitors the output voltage of the coil, while the electrodes of the Helmholtz coils were connected to a waveform generator and parallel to it a multimeter, in order to monitor the voltage of the Helmholtz coils. During the FR experimental procedure, the Helmholtz coils were generating AC magnetic field and the procedure was divided in two parts. The first part was to find the optimum operating parameters of the Helmholtz coils, while the second part was the measurement of the ribbon’s frequency response based on these parameters.

As far as the first part is concerned, its necessary to determine the resonant frequency of the complex RL circuit (Resistor - Inductor circuit) which is formed
because of the connections of the Helmholtz coils to the waveform generator. This is done in order to define a suitable frequency range away from the resonance. Fig. 4.7a presents the results of this part in a logarithmic scale. The magnitude was calculated using the equation:

\[
\text{Magnitude(dBs)} = 20 \log \left( \frac{V_D}{V_H} \right)
\]  

(4.1)

where \(V_D\) is the inductive voltage on the detection coil due to the Faraday’s law of induction and \(V_H\) is the voltage of the Helmholtz coils. It is clear that there are three distinct areas in this plot, the low-frequency area (below \(f_1 = 500\) Hz), the mid-frequency area (between \(f_1\) and \(f_2 = 50\) kHz) and the high-frequency area where the resonance peak is at \(f_r = 313\) kHz. Since this thesis deals with the use of the metglas ribbons as structural vibration sensors, all frequency response measurements were performed in the range of \([0.01 - 50]\) kHz, so as to have a clear sensor signal free of the coil resonance. Another important experimental parameter that had to be optimized was the homogeneity domain of the magnetic field within the Helmholtz coils. Fig. 4.7b shows the change of the magnetic field along the Helmholtz coils axis, where the position \(z=0\) mm corresponds to the center of one of the two coils (the center of the other coil is located at position \(z=210\) mm). The magnetic field was produced by a DC current of \(I = 1\) A and it was measured with the help of a gaussmeter (FW Bell 7010 Single Channel Gauss/Tesla Meter, 50 kHz). It is clear from Fig. 4.7b that the field reaches the

\[ \text{Figure 4.7:} \quad (\text{a})\text{ The frequency response of the Helmholtz coils - waveform generator circuit,} \\
\text{ (b) Magnetic field along the axis crossing the center of the Helmholtz coils.} \]

maximum point and starts to stabilize about \( L = 50 \text{ mm} \) from the center of one of the two coils, and remains stable up to the value \( L = 150 \text{ mm} \), where it starts to decrease. In both FR and NFR experimental setups, the detection coils were located entirely within the homogeneous region of the Helmholtz coils.

### 4.3.2 The NFR experiment

For the case of the NFR setup (Fig. 4.8), the aluminum alloy 6063 was used as the beam material for the characterization process and two metglas ribbons were attached on the CB’s free end using double-sided tape. The dimensions of the CB were set as before using a CNC machinery, and they are length=700 mm, width=30 mm and height=10 mm. A number of three natural frequencies of the CB were chosen for the process, namely the 2nd, the 4th and the 6th bending modes (Fig. 4.9 shows the 3D visualization of these modes using ANSYS program). The reason for choosing bending modes rather than torsional is because the torsional modes are very difficult to excite and detect in CBs, due to their very

![Figure 4.8](image-url)

**Figure 4.8:** (a) Panoramic view of the non-frequency response (NFR) experimental setup. A detailed view of the (b) left and (c) right side of the NFR setup, (d) Schematic diagram of the connections between the devices in the NFR setup.
small oscillation amplitude. Also, the reason for choosing these specific bending modes is because they cover a decent range in the frequency response of the ribbons.

Figure 4.9: 3D visualization of the cantilever beam’s bending mode shapes using ANSYS program: (a) 2nd (102.10 Hz), (b) 4th (550.16 Hz) and (c) 6th (1322.08 Hz) bending modes. The fixed end of the beam is colored with deep blue and it is the upper left part in each 3D graph.

Fig. 4.8d shows the schematic diagram of the connections between the devices and it can be seen that more devices are involved in this setup (compared to the FR setup). On this setup the waveform generator was connected to a mechanical vibrator which was attached to the CB close to its fixed end. Parallel to the mechanical vibrator, a multimeter was connected to monitor the vibrator excitation voltage. The free end of the beam was, along with the attached metglas ribbons, located inside a homemade solenoid detection coil, of which its electrodes were connected both to the multimeter and the DAQ of a laptop. Finally, a DC power supplier was connected to the Helmholtz coils in order to create the bias field (DC magnetic field) for the magnetization of the metglas ribbons. In particular, the following devices were used in the current experiment:

- **Waveform Generator:** KEITHLEY 3390 50MHz Arbitrary Waveform Generator
- **Digital Oscilloscope:** Tektronix TDS 1002 Digital Oscilloscope (60 MHz - 1GS/s)
- **Multimeter:** KEITHLEY 2000 Multimeter
- **DC Power Supplier:** TTi QL355P Power Supply
- **Mechanical Vibration Generator:** Frederiksen Vibration Generator no.2185.00
- **Hydraulic Press:** CARVER 4350 Manual Pellet Press
Similarly to the FR setup procedure, optimization was performed also in the NFR setup but with different parameters, such as the boundary conditions of the CB and the bias field. Concerning the first parameter, Fig. 4.10a presents the dependence of the 2nd bending mode frequency on the pressure load applied at the fixed end of the CB. It can be seen that as the applied load increases, the frequency comes to a saturation value of $f_s = 102.10$ Hz. So, it can be considered that the boundary conditions of the fixed end are optimum under high applied loads, such as 10 kN. Concerning the second parameter, Fig. 4.10b presents the dependence of the signal voltage of the 2nd bending mode on the bias field for an applied load of 10 kN. Here, the signal voltage shows an increase with the applied field, up to a certain point, and then a gradual decrease. The peak of the curve occurs at $B = H_A = 0.64$ Gauss, where $H_A$ is an important property of the magnetic materials known as "anisotropy field", and is defined as the magnetic field required to disorient all the magnetic domains from the anisotropy direction. This optimum field was used to study the characterization parameters of linearity, stability and reversibility.

**Figure 4.10:** (a) Frequency dependence of the 2nd bending mode of the cantilever beam with the applied load at the fixed end, (b) Change of the signal voltage of the 2nd bending mode versus the bias field.
Chapter 5
Usage of magnetoelastic ribbons as vibration sensors: Proof of concept

5.1 Abstract

This chapter discusses the proof of concept of the thesis which is the usage of magnetoelastic ribbons as vibration sensors in detecting the natural frequencies of mechanical structures such as cantilever beams (CB). The ribbons used are composed of an amorphous metallic alloy known as Metglas 2826MB series. In order to examine the concept, various cantilever beam specimens of the same dimensions but with a single transverse crack at different positions and depths were tested, fixed at one end by using a hydraulic press so as to have consistent boundary conditions. The beams material was aluminum alloy 6063 and each one of them was prepared and tested according to the experimental procedure described in Section 4.2. The FFT (Fast Fourier Transform) spectrum of the recorded signal, for each specimen, revealed seven dominant peaks in the frequency range of [0 - 7] kHz, thus showing the ability of the ribbons to transmit the dynamic state of the CB. The frequency values of the peaks were compared to the ones predicted by ANSYS modal analysis and the results showed an excellent agreement between the predicted and the measured frequency values, showing thus the accuracy of the magnetoelastic ribbons in detecting the undamaged and
damaged state of mechanical structures such as CBs.

5.2 Introduction

Large-scale civil infrastructures such as buildings, bridges, dams, wind turbines, and pipeline systems are exposed to various external loads throughout their lifetime. Vibration caused by earthquakes, wind, temperature, or human-made excitation initiates structural damage during their service lives and sometimes triggers catastrophic failure. Structural Health Monitoring (SHM) is an emergent and powerful diagnostic tool for damage detection and disaster mitigation of large-scale structures. The SHM comprises four key elements: data acquisition, system identification, condition assessment and decision making/maintenance. The traditional SHM methods use responses from the entire structure (global) such as vibrations or local responses such as strains, or a combination of both to assess the structure during in-service conditions or extreme climatic events. Unlike displacement or strain-based methods, vibration-based SHM strategies [1–3] are very effective in evaluating global health state of structures and performing a rapid risk assessment and hazard mitigation. Most of these techniques primarily rely on acceleration measurements that require the installation of either contact or non-contact sensors collecting rich quality of data.

Several studies have been presented over the years in sensing methods which involve the use of different materials and devices as vibration sensors on mechanical structures [4–10]. One of the most known are the piezoelectric-based material sensors. Abramovich and Pletner [11] proposed a piezo-laminated sandwich type structure for actuation and sensing of harmonically excited thin walled structures. The numerical results are compared with experimental ones obtained during a test series on a cantilever sandwich beam equipped with piezoceramic sensors and actuators, and constructed according to the proposed concept. Seeley and Chattopadhyay [12] developed a multiobjective optimization technique which includes actuator locations, vibration reduction, power consumption, minimization of dissipated energy and maximization of the natural frequency peak
as design objectives by using piezoelectric sensors. The technique was demonstrated through a cantilever beam problem and showed that performance control can be obtained with only a few optimally placed actuators. Wang and Wang [13] presented a theoretical analysis of the application of piezoceramic transducers to cantilever beam modal testing by considering four pairs of sensors and actuators including accelerometer-point force, accelerometer-PZT, PVDF-point force and PVDF-PZT. Results showed that any sensor-actuator pair can successfully determine natural frequencies and damping ratios. In refs [14–16] the authors investigated the concept of a continuous, grid-type sensor array composed of lead-zirconium-titanium oxide (PZT) sensors, for detecting vibrations and stress waves in mechanical structures for cracks detection. Park et al. [17] reviewed 69 articles related to impedance-based SHM applications, based on piezoelectric materials, and made a brief overview on the methods that use high-frequency structural excitation to monitor the local area of a structure, for changes in structural impedance that would indicate imminent damage.

Optical fibers are another category of materials that can be used as vibration sensors. The authors in [18–21] developed several synthetic detection devices, incorporating optical fibers, to monitor the structural health of concrete and non-concrete structures, by measuring the static and dynamic loads in the structures. Kuang et al. [22] examined the potential of an extrinsic plastic optical fiber (POF) sensor for vibration-based structural health monitoring (SHM) applications. The POF sensor was surface-bonded to a cantilever beam and the time-history responses of the sensor, following an impulse-type excitation of the beam, were obtained and analyzed using classical Fourier transform techniques. The results showed that the POF sensor is capable of detecting the relevant modal frequencies. Perrone et al. [23] presented a non-contact method to measure vibrations by using a low-cost optical system that is able to provide a sub-micrometer resolution. The system used POFs for the realization of the sensing head and a simple spectral analysis to evaluate the amplitude of the vibrations. Comparison results with other techniques or reference systems have shown the capability of the system to measure the amplitude of vibrations up to about 40 kHz with a
resolution of below 1 $\mu$m. Another optical method to measure the dynamic state of a mechanical structure is the use of laser vibrometers. Sriram et al. [24] applied a time-domain sorting algorithm to demonstrate the use of a scanning LDV (Laser Doppler Vibrometer) to simulate multiple discrete sensors distributed over the test structure. They illustrated the technique by measuring the second mode shape of a light-weight cantilever beam through the processing of the LDV output signal in the frequency domain. Okafor and Dutta [25] recorded and analyzed with wavelet transform, the first six mode shapes of a damaged and undamaged aluminum cantilever beam using scanning laser vibrometer. A finite-element model of the beams showed a close correlation to the corresponding experimental beam results.

In the current work magnetoelastic ribbons made of an amorphous magnetic alloy material are tested as vibration sensors. Many studies have been carried out on magnetoelastic materials to exploit the magnetoelastic property for sensing purposes [26]. Magnetoelasticity (or magnetostriction) is the property of some ferromagnetic materials to change their magnetic state (magnetization) by the application of mechanical stresses. The opposite effect is also occurs in these materials, which is their continuous deformation under the application of a magnetic field. An important parameter in these materials is the magnetoelastic coupling coefficient $k$ which is equal to the ratio of conversion of the magnetic to the elastic energy ($0 < k < 1$). Authors in refs. [27, 28] have measured this coefficient of some ferromagnetic metallic glasses ribbons and found values between 0.70 < $k$ < 0.97, with the latter one being the highest among the known magnetoelastic materials. Ausanio et al. [29] examined the influence of stress on the amplitude of the resonant mechanical waves inside a Fe$_{62.5}$Co$_6$Ni$_{7.5}$Zr$_6$Cu$_1$Nb$_2$B$_{15}$ ribbon for strain-stress real-time monitoring in civil buildings. The results exhibited good reliability and stability as well as better sensitivity [up to 200 times higher in proper conditions using resistive and vibrating wire strain gauges].
5.3 Measured and calculated frequency data

In Section 4.2 an extensive description of the experimental setup and procedure was given for the proof of concept of the thesis. As it has been mentioned there, the data recording process was done using the Matlab software program. Fig. 5.1a shows the recorded signal for the undamaged CB after a single short pulse excitation. The total recording time was set to 10 s, so there was enough time to stimulate the beam and let it vibrate, and the sampling rate at 14 kHz. With these recording settings the frequency resolution was calculated to be 0.1 Hz. Fig. 5.1b shows the FFT analysis of the signal where seven in total peaks are revealed within the frequency range of [0-7] kHz. These peaks correspond to the first seven bending modes of the CB to the easy axis (easy modes) and are the identity of both the physical and the geometrical characteristics of the CB. As easy axis is defined the geometric axis in which the CB exhibits the least moment of inertia. This is also the main reason for not seeing other modes than bending ones. Modes like hard-axis-bending and torsional (hard modes) are difficult to stimulate in CBs due to their high values of moment of inertia.

![Figure 5.1: (a) Recorded signal voltage versus time and (b) its corresponding FFT analysis using Matlab software program.](image)

The amplitude of the peaks seems to vary with frequency, with the 1st bending mode having the smallest and the 4th the largest amplitude. There are three factors that determine the amplitude of the peaks and are related to the nature of the detection process. As described in Section 4.2, the magnetoelastic ribbons attached to the free end of the CB follow its vibrations, and because of their
magnetoelastic character their magnetization changes continuously. This produces an AC magnetic field around it, which in turn induces an AC electrical voltage to a nearby coil (solenoid, ring etc.) due to the Faraday’s law of induction (the amplitude of the peaks is proportional to the strength of the recorded voltage). According to this law the induced electrical voltage at the electrodes of any coil is given by the equation:

\[ V_{\text{induced}} = -N \frac{d\Phi}{dt} = -N A \frac{dB}{dt} \]  

(5.1)

where \( N \) is the number of the turns in the coil, \( A \) the cross section of the coil and \( B \) the external magnetic field within the coil. In the case of a single ring coil \( N=1 \). The first two parameters (\( N \) and \( A \)) are constant and relate to the geometrical characteristics of the coil. The third parameter (\( B \)) has to vary with time, otherwise the \( V_{\text{induced}}=0 \) V, and does not relate to the coil. In our case, the parameter \( B \) is the magnetic field produced from the ribbons due to their magnetization change and it is proportional to the amplitude of the CB’s bending oscillation. The reason that the ribbons magnetization is proportional to the amplitude of the CB’s oscillation is because in these materials the magnetization is directly proportional to the applied mechanical stress on them, and because with higher amplitude oscillations the deformation on the ribbons is larger, the stress is also larger. In the simple case where there is only one vibration frequency \( \omega \) in the beam, then the corresponding produced magnetic field has the form \( B = B_0 \sin(\omega t) \), where \( B_0 \) is its magnitude. By substituting the expression of \( B \) in Eq. 5.1 the final form of the induced voltage is given as:

\[ V_{\text{induced}} = -\omega N A B_0 \cos(\omega t) \]  

(5.2)

Thus, the first factor that determines the amplitude of the peaks is the geometrical characteristics of the detection coil (\( N, A \)), the second one is the radial frequency \( \omega \) and the third is the magnitude \( B_0 \) which is directly proportional to the mechanical stress applied to the ribbons due to their magnetoelastic character. All three factors contribute to the measured signal.
In this part of the thesis the measured frequencies are compared with computational ones using ANSYS modal analysis. Since the mechanical structure used were CB specimens, fixed end boundary conditions are assumed. While in ANSYS simulations these conditions are ideal, experimentally we had to use a hydraulic press in order to approach them as much as possible. This is because the measured frequency of a CB depends strongly on the boundary conditions. This is evident in Fig. 5.2 where the measured frequencies of each bending mode of the undamaged CB depend on the applied load at the fixed end. The load range was [1-10] kN with a step of 0.5 kN, and for each step 5 different frequency measurements were made to derive the mean value of the frequency. For each plot shown in Fig. 5.2 there is a frequency saturation value that occurs at high values of applied load and can be calculated by fitting the data with exponential saturated functions. The fitting process was accomplished using the OriginPro 2015 software program, where a total number of 84 exponential saturated functions were used, provided by the software, in order to get the best fitted function. The functions were fitted by using the rank model option in the fitting analysis of the program and the first criterion for selecting the best fitted function was the Adjusted R-Squared ($R^2$) of the fitting results.

### 5.4 Computational frequency data

In beam theory the natural frequencies of an undamaged beam are calculated simply by applying the differential equations of elasticity with the appropriate boundary conditions. However, when defects are introduced into the beam, the differential equations become intractable. In such a case, only numeric solutions are possible through appropriate design model and FEM (Finite Element Method) analysis. Here, the designing of each CB model was done using an open source parametric 3D modeler software named FreeCAD. A total number of 31 designs of different CBs were made which correspond to the CB specimens used in the experiment. Fig. 5.3a shows the CB model as it was designed in FreeCAD. The design model was divided in 7 free blocks plus 1 fixed, which are bonded together,
Figure 5.2: Bending modes frequency measurements with respect to the applied load at the fixed end of the undamaged CB: a) 1st, b) 2nd, c) 3rd, d) 4th, e) 5th, f) 6th, g) 7th modes.
Figure 5.3: (a) CB model as designed in FreeCAD with the metglas ribbons, the fixed end and the picked crack positions pointed out, (b) Side view of the CB model with the shape of the crack inserted (Width $V_w = 1$ mm, V-notch height $V_h = 0.5$ mm, CD varies from 1 to 5 mm with a step of 1 mm).

and in-between the free blocks the crack positions were set. The reason for this division was the achievement of a better local meshing at the crack location. As with the experimental beams, the crack positions here were chosen to be at the locations 50 mm, 100 mm, 150 mm, 200 mm, 250 mm and 300 mm from the fixed end. The crack design was picked to have the shape of a V-notch, as shown in Fig. 5.3b, in order to approach as possible the real shape of the cracks, and the depth was varied from [1-5] mm with a step of 1 mm, the same with the real CB specimens. The designed metglas ribbons were attached to the opposite CB face to the face where the cracks are located, so as to keep them an-effected by the cracks.

Once the preparation of all CB models was finished, each one of them was simulated using ANSYS 2016 Workbench software program. The designed models from the FreeCAD program were imported into ANSYS Workbench in the form of '.step' files and simulations were performed in pre-stress conditions which includes static and modal analysis (Fig. 5.4). Three steps were followed in the Workbench environment to analyze the beam model and extract the natural frequencies of it. The first step involved the selection of the material, the contact parts, the meshing, the boundary conditions and the environment conditions (such as standard earth gravity direction). The second step included the solution of the problem with these settings and the third step was the extraction of the mode shape results. Fig. 5.5a shows the meshed model of the undamaged CB in rest in top view. A
rectangular mesh was used to discretize all the CB models into cubic elements,

Figure 5.4: ANSYS Workbench interface with the Pre-Stress Modal analysis completed.

Figure 5.5: (a) A rectangular meshing geometry was used to discretize the CB design model into elements. The 3D representation of the first 7 bending modes of the undamaged CB: (b) 1st, (c) 2nd, (d) 3rd, (e) 4th, (f) 5th, (g) 6th and (h) 7th.
with the element edge being 1 mm for the main body of the model and 0.1 mm close to the crack location. Meshing the defect region (crack) with smaller elements increases the precision of the modal results (natural frequency values). Figs. 5.5b to 5.5h show the 3D graphs of the deformation of the undamaged CB for the first seven bending modes along the easy axis. The color legend in the left side of the graphs describes the displacement degree of the elements, with the blue color corresponding to zero and the red the maximum. As expected, the fixed and free ends of the CB are blue and red, respectively, and are located correspondingly in the far left and right side of the CB. Also, the blue areas that appear along the beam (Figs. 5.5c to 5.5h) correspond to the nodes of the CB and they are characteristic for each bending mode.

5.5 Comparison results

In this section the experimental and simulations results of the two previous sections are going to be compared and discussed. Table 5.1 shows the experimental

Table 5.1: Comparison of the computational and experimental frequency values of the bending modes for the CB specimens with the crack location at L=50 mm from the fixed end and different crack depths ($\alpha$ varied from [0-5] mm with step of 1 mm).

<table>
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<tr>
<th>$\alpha$ (mm)</th>
<th>$f(Hz)$</th>
<th>$f_1$</th>
<th>$f_2$</th>
<th>$f_3$</th>
<th>$f_4$</th>
<th>$f_5$</th>
<th>$f_6$</th>
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<td>3020.3</td>
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<td>0.41</td>
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<td></td>
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<td>943.8</td>
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<td>0.02</td>
<td>0.01</td>
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<td>Error(%) 0.19</td>
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<td>0.08</td>
<td>0.18</td>
<td>0.21</td>
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<td>0.12</td>
<td>1.01</td>
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</tbody>
</table>
CHAPTER 5. USAGE OF MAGNETOELASTIC RIBBONS AS VIBRATION SENSORS: PROOF OF CONCEPT

saturated (EXP) and the computational calculated (ANSYS) frequency values of each bending mode along with the absolute percentage error values between them, for the case with a crack at the location L=50 mm from the fixed end and depth $\alpha=0$ mm (undamaged), 1 mm, 2 mm, 3 mm, 4 mm and 5 mm. It can be seen from the error values that there is an excellent agreement between the EXP and ANSYS values, since almost in all cases the error is below 1%, with only one exception for the case of $\alpha=5$ mm, at the 7$^{th}$ bending mode, where the error value is 1.01%.

Shown in Figs. 5.5a to 5.5g is the summary of all comparison data in 3D bar graphs where each graph corresponds to a different bending mode. The purpose of the inset in Fig. 5.6h is to show clearly the axis numbers and units of the graphs. The vertical axis corresponds to the percentage error between the measured and computational frequency values (similar to the error values in Table 5.1), while the two horizontal axes correspond to the location of the crack and its depth. Starting with the 1$^{st}$ bending mode it can be seen from Fig. 5.6a that almost all of the error values are lower than 0.5%, with the only exception the case with crack location L=50 mm and crack depth $\alpha=3$ mm, where the error value is between [0.5-1]%. For this mode the average percentage error value was calculated to be 0.09% and it was the lowest value of all bending modes. The 2$^{nd}$ bending mode (Fig. 5.6b), along with the 3$^{rd}$, 4$^{th}$, 5$^{th}$ and 6$^{th}$ modes (Figs. 5.5c to 5.5g), also had most of the error values below 0.5%. For the 2$^{nd}$ mode the maximum error was at the point (L=200 mm, $\alpha=5$ mm) with a value of 2.0%, while for the 3$^{rd}$, 4$^{th}$, 5$^{th}$ and 6$^{th}$ modes the maximum error was at the points (L=250 mm, $\alpha=5$ mm), (L=300 mm, $\alpha=2$ mm), (L=300 mm, $\alpha=2$ mm), (L=300 mm, $\alpha=3$ mm) with values of 2%, 2%, 4% and 2% respectively. The average percentage error value for each one of these modes (2$^{nd}$, 3$^{rd}$, 4$^{th}$, 5$^{th}$ and 6$^{th}$) was calculated to be 0.38%, 0.29%, 0.34%, 0.51% and 0.44%, respectively. Finally, the 7$^{th}$ bending mode had the highest error values in relation to the other modes as most of them were between [0.5-1.5]%. The maximum error for this mode was at the point (L=300 mm, $\alpha=3$ mm) with a value 2% and the average percentage error value was calculated to be 0.68%. This excellent agreement between the ANSYS and the measured frequency values shows the capability of the magnetoelastic ribbons to be used as vibration sensors.
CHAPTER 5. USAGE OF MAGNETOELASTIC RIBBONS AS VIBRATION SENSORS: PROOF OF CONCEPT

Figure 5.6: Percentage error values between experimental and computational frequency values, versus crack location and depth, for all bending modes: (a) 1\textsuperscript{st}, (b) 2\textsuperscript{nd}, (c) 3\textsuperscript{rd}, (d) 4\textsuperscript{th}, (e) 5\textsuperscript{th}, (f) 6\textsuperscript{th} and (g) 7\textsuperscript{th} (h) An inset that indicates the axes numbers and units, and the error color scale.

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5.6 Bending modes frequency variation

The frequency comparison results presented in the previous section showed the accuracy of the metglas ribbons in detecting the dynamic behavior of mechanical structures such as CBs. They also showed that the dynamic state of the CB is changing when a crack is introduced to it (in Table 5.1 the bending mode frequency values are changing with crack depth).

Fig. 5.7 contains the 3D graphs of the normalized frequency of each bending mode versus crack depth and location. As normalized frequency (NF) is defined the ratio $f_{\text{damaged}}/f_{\text{undamaged}}$, where $f_{\text{damaged}}$ and $f_{\text{undamaged}}$ are the experimental bending frequency values of the cracked and uncracked CBs, respectively (the vertical axis in all graphs is dimensionless and reversed, and it is starting from 1 and going up to 0.8). Notice that in all graphs the dependence of the NFs with the crack depth is monotonous, while with respect to the crack position the behavior is more complicated. Starting with the $1^{st}$ bending mode (Fig. 5.7a), the NF varies monotonically with respect to the distance of the crack from the CB fixed end (it is low close to the fixed end and it converges non-linearly to unity as the crack location moves to the free end). Also, it appears that for crack locations near to the fixed end, the NF is changing rapidly versus crack depth. Comparing the NF behavior of the $1^{st}$ bending mode with the rest of the modes (Figs. 5.7b to 5.5g), it can be seen that in the case of the latter, the NF changes are not monotonic and show a partial complementary behavior with each other. For example, the $2^{nd}$ bending mode (Fig. 5.7b) shows marked NF changes mainly in the region between [100-300] mm and intense in the region [150-250] mm. The behavior is quite the opposite in the case of the $6^{th}$ bending mode (Fig. 5.7f) where the NF changes are more active at regions [50-100] mm and [250-300] mm, and less to almost none in the region [150-250] mm. The same complementary behavior appears between the $3^{rd}$ (Fig. 5.7c) and the $5^{th}$ bending mode (Fig. 5.7e). It is apparent that the $3^{rd}$ mode shows intense maxima at the crack locations L=50 mm and L=250 mm from the fixed end, while the $5^{th}$ mode shows deep minima there.
Figure 5.7: Normalized frequency values versus crack location and depth for all bending modes: (a) 1st, (b) 2nd, (c) 3rd, (d) 4th, (e) 5th, (f) 6th and (g) 7th. (h) Inset that indicates the axis numbers and units, and the normalize frequency color scale.
Fig. 5.8 is a schematic color illustration of the CB deformation for each bending mode except the 1st one, as given by ANSYS for the 5 mm crack depth (little white bar) at different locations, indicated by numbers to the right of Fig. 5.8a. The top illustration in each figure corresponds to the undamaged case and the fixed end of the CB is on the right side of each figure. As it has been mentioned above (Section 5.4), the blue color regions correspond to zero displacement of the CB elements in that location and can be considered as standing wave nodes, while the red color regions correspond to exactly the opposite, a maximum displacement of the CB elements, and can be considered as standing wave antinodes. It can be seen from all schematics of Fig. 5.8 that the position of the 5 mm depth crack has a major effect on the shape of the bending modes. In particular, when the crack is in the vicinity of a node no much is happening, whereas when the crack is midway between two nodes it creates there a local maximum (red colors around crack) which means that the mode has been strongly altered. This is also in accordance with the data shown in Fig. 5.7. For example, examining the 2nd bending mode at the schematic illustration of Fig. 5.8a, two nodes are apparent, one at the fixed end and one around 275 mm from it. When the crack location is at 150 mm, which is about half way between the two nodes, some red color appears around the crack, where it was yellow before at the undamaged case. Taking a closer look at (Fig. 5.7b), which corresponds to the NF changes of the 2nd bending mode,
it can be confirmed that the highest change happens for crack locations between [150-200] mm. Similarly, in (Fig. 5.8b), which shows the deformation shape of the 3rd mode, the crack at the locations 100 mm and 250 mm creates a local maximum which is in agreement with the NF changes of Fig. 5.7c. Another example is the deformation schematic of the 7th bending mode (Fig. 5.8f) where there are quite a few nodes along the CB. As the crack location is shifting away from the fixed end, the color around it is getting redder till the point of 200 mm. At this location the crack is situated almost symmetrical between two successive nodes and thus it is expected the highest change in the NF values. Indeed from Fig. 5.7g, it can be seen that the most intense change of the NF appears at the crack location 200 mm.

The fact that the frequency changes are minimal, when the crack is close to a node, can be explained physically: A standing wave does not carry energy and thus at the nodes the energy is very low which means that the crack cannot absorb energy from the beam. On the contrary, when the crack is midway between two nodes, there is plenty of energy available which is absorbed by the crack in order to behave like an almost free-end. Also, from Figs. 5.7 and 5.8 it can be seen that as the number of the nodes is getting larger in higher modes, the NF changes are less intense, and can be explained through the amount of energy in higher modes. In higher modes the vibration energy is divided in more portions and thus the appearance of a crack has a smaller effect on the frequency values, due to the less energy that it absorbs.

5.7 Conclusions

In this part of the thesis, magnetoelastic ribbons of Metglas 2826MB series material were tested as vibration sensors. The experimental procedure involved the detection of the first seven bending modes of damaged and undamaged cantilever beams (CBs), and comparison of the results with computational ones predicted by ANSYS modal simulations. The detection method was accomplished with the help of a coil, connected to sound card, which was placed close to the ribbons and
recorded the signal through the Faraday’s law of induction. A total number of 31 CB specimens were tested, where 1 was undamaged and the other 30 had a single crack at fixed locations and depths. Special care was taken with the boundary conditions of the CBs where a hydraulic press was used and the frequency measurements were done in different applied loads at the fixed end of the CBs. The comparison results showed an excellent agreement between the experimental and computational frequency values, with the average percentage error values for all bending modes being below 1%, showing thus the effectiveness of the metglas ribbons to be used as vibration sensors. Also, an analysis was made on the variation of the normalized frequencies of each bending mode versus crack location and depth, where a correlation appeared between the crack positions with respect to the node and anti-node positions.
CHAPTER 5. USAGE OF MAGNETOElastic RIBBONS AS VIBRATION SENSORS: PROOF OF CONCEPT

5.8 Bibliography


Chapter 6
CHARACTERIZATION OF MAGNETOELASTIC RIBBONS AS VIBRATION SENSORS

6.1 Abstract

This chapter discusses the characterization of the Metglas 2826MB series ribbons as vibration sensors. The process involves seven different sensor parameters such as the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. Two experimental setups were used for the characterization process, one for the frequency response parameter (FR setup) and one for the rest of the parameters (NFR setup). The frequency response parameter was examined for two different states of the ribbon, the non-annealed and the annealed states, and better characteristics were revealed for the annealed state.

In the NFR setup, a cantilever beam (CB) was used as a vibrating platform with two annealed metglas ribbons attached on its free end using a double-sided tape. The 2nd, 4th and 6th bending modes of the CB were used for the characterization process.

Concerning linearity, the ribbons showed an extremely linear behavior, with an average value of the adjusted R-square being $R_a^2 = 0.99995$. The SNR and Quality factor parameters were studied versus the DC magnetization field of the Metglas ribbons (bias field) and the results showed that the implementation of a
DC magnetic field increased the strength of the detectable signal without reducing its quality. The ribbons stability was examined within a time period of 2 hours, with the average percentage deviation from the mean frequency being as small as 0.005 % and the average change of the frequency with time as small as $F_a = (1.1 \pm 0.4) \cdot 10^{-4}$ Hz/min. As for the repeatability parameter, the ribbons were subjected to alternating CB clumping force and showed an excellent behavior during repetition cycles. The last parameter studied was the sensitivity of the ribbons in detecting the shift of the natural frequencies versus CB stiffness, when a crack was introduced. Each bending mode revealed a different value of sensitivity, with the 2nd mode having the lowest and the 6th mode having the highest. The average sensitivity value among the three modes was calculated to be $S_a = (38 \pm 1) \cdot 10^{-3}$ Hz/(N·m$^{-1}$).

6.2 Introduction

Sensors are an integrable part of modern technology and even common people are using indirectly 5-10 sensors daily on average through their devices like, smart phones, air-conditions, televisions, refrigerators etc. This is the reason why there is an intense research for the design and implementation of various types of sensors, with focus mainly towards the miniaturization and the low cost, without compromising important sensor qualities. These qualities are typically the linearity, sensitivity, repeatability, stability, selectivity, frequency response etc. Thus, it is imperative for each new sensor design to have the sensors fully characterized so as to know if they fulfill certain specifications.

Several studies have been conducted around the development of vibration sensors, with greater emphasis being placed on their characterization. Abas et al. [1] characterized a vibration sensor made of piezoelectric paper by attaching it on an aluminum cantilever and measuring its frequency response function when it was subjected to impulse loading and sinusoidal excitation. They compared their measurements with a piezoelectric polymer sensor which was placed at the same cantilever and found that the two spectra were very similar with the natural fre-
frequencies differing by no more than 1%. Jenq and Chang [2] have used a piezofilm sensor on both a Glass-fiber-reinforced-plastic cantilever and an aluminum cantilever, and characterized it with respect to its frequency response. They found that the sensor behaves as expected above $3 \ kHz$, while it underestimates the structural response below that limit. Kim et al. [3] have designed a shear-mode piezoelectric accelerometer for high temperature vibration sensing applications. They measured the frequency response of their sensor as sensitivity versus temperature from $1 \ Hz$ to $3 \ kHz$ and found that the functional frequency range was $1–350 \ Hz$. Also, they showed that the sensor exhibited linearity with the g values and its resolution was quite stable in the temperature range $0–1000 \ C$. Laflamme et al. [4] characterized the dynamic behavior of soft elastomeric capacitors (SECs) as vibration-based SHM sensors on steel and concrete specimens. They examined the frequency response and the sensitivity of the sensor, and the results revealed that the sensor was capable of detecting frequency inputs in the range [1-40] Hz, but the presence of noise showed that the SEC was limited below 15 Hz in the time domain. Lin et al. [5] have implemented an embedded micro-strain sensor inside a polyurethane (PU) thin film to measure its stress/strain in situ. They presented data for the linearity of the sensor. Dai et al. [6] have designed a strain sensor using carbon nanotube-based non-woven composites. They did a very detailed sensor characterization as well as mechanical characterization of their composite material. The sensor characterization involved linearity, reversibility and stability. Ge et al. [7] have put together a review on flexible and wearable strain sensors for human-motion monitoring and in this review they presented data for linearity and repeatability of various sensors.

In recent years, more and more interest has grown around magnetoelastic materials, and lots of studies have been carried out on the exploitation of their magnetostrictive-magnetoelastic properties for sensing purposes [8–16]. Grimes et al. [17] reviewed 68 articles on magnetoelastic resonance sensors, and presented a comprehensive review on the theory, operating principles, instrumentation and key applications of magnetoelastic sensing technology. In general, the magnetoelastic effect (or Villari effect) appears in certain ferromagnetic materials, where
their magnetic susceptibility changes with the application of mechanical stress. By far the best known magnetoelastic materials are the metallic glasses. One of the most common used material in this category is the Metglas 2826MB series. It is an iron-nickel based alloy material, with a medium saturation induction and lower magnetostriction, and is usually made in the forms of thin foils or ribbons. Authors in Refs. [18–20] used these series of metallic glasses to develop contactless and wireless strain sensors. Tan et al. [21] developed a wireless and passive stress sensor by measuring changes in the induced magnetic field of a Metglas 2826MB series strip attached to a solid body in the form of a bridge-like structure. Their measurements were the frequency changes of the 2nd-order harmonic of the strip, and showed that there is a very good correlation between the sensor’s signal and the applied lateral compressive stress on the structure. Ausanio et al. [22] proposed an elastomagnetic material made of Sm$_2$Co$_7$ microparticles as vibration sensor. They detected vibrations amplitudes from 0.1 to 1 mm with a sensitivity of 3.5 mV/mm, in the frequency range from 5 up to 50 Hz, and showed the usefulness of the sensor in low vibration frequency detection. Zhang et al. [23] made a different approach on developing a magnetic stress detection system, by presenting a non-contact stress detection device based on the magnetoelastic effect, which does not include metallic glasses as transmitters. The device setup included two coils, one excitation and one detection, winded on a magnetic core out of soft magnetic ferrite, and placed orthogonally without contact on a ferromagnetic steel plate. They measured the induced voltage on the detection coil with respect to the applied stress on the plate, and showed the linear dependence between those variables, both experimentally and theoretically. Wichmann et al. [24] presented an advanced measurement device, using magnetoelastic coil sensors, for the direct stress monitoring in prestressed steel elements used in various infrastructure buildings and other structures subjected to high loads such as bridges.

In the current part of the thesis magnetoelastic ribbons of Metglas 2826MB series material are characterized as vibration sensors, where the process includes seven overall sensor parameters and are the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. The above
parameters are very useful for structural health monitoring (SHM) applications, where the monitoring is carried out by measuring the vibration frequencies of the structure. In the next section, all the specified characterization parameters are defined individually and in detail, with their experimental data fully presented.

6.3 Characterization results

6.3.1 Frequency response

The procedure of measuring the sensor frequency response is similar to the aforementioned frequency response measurement of the Helmholtz coils in Section 4.3.1, with the only difference being that one ribbon was placed inside the detection coil. Shown in Fig. 6.1 are the sensor’s frequency response results. The measurements were taken within the frequency range of $[10 \, Hz - 50 \, kHz]$ and the intensity of the applied AC magnetic field was kept fixed at 0.045 Gauss for all frequencies, by feeding the Helmholtz coils with a constant AC current of 20 mA. This intensity was chosen because the ribbon at this value exhibits linearity with respect to the AC field.

There are three different sets of measurements in the graph, one for the detection coil without any ribbon inside it (reference state), one for the detection coil...
coil with a non-annealed metglas ribbon inside it and one with a thermally annealed ribbon. As it was expected, the detection signal changes (increases) with the insertion of the ribbons. This happens due to the ferromagnetic nature of the metglas ribbon which acts as a magnetic core inside the coil and increases the intensity of the field. The extra gain on the magnitude between the annealed and the non-annealed ribbons is directly linked to the saturation induction increase with the annealing process. Also, the curvature of both the annealed and the non-annealed ribbons seems to follow the curvature of the reference state (single coil without ribbon).

It is clear from the graph that there are three distinct ranges corresponding to different curvatures, the low-frequency range \([10 \, Hz - 100 \, Hz]\), the mid-frequency range \([100 \, Hz - 500 \, Hz]\) and the high-frequency range \([500 \, Hz - 50 \, kHz]\). In the low-frequency range the magnitude seems to increase linearly, especially after 20 \(Hz\), for all three curves, but with a different rate (slope). Applying a linear fitting analysis in this range for each curve, gives a slope of \((0.10 \pm 0.01) \, dB/Hz\) for the single coil and the slopes of \((0.11 \pm 0.01) \, dB/Hz\) and \((0.12 \pm 0.01) \, dB/Hz\) for the non-annealed and the annealed state, respectively. At the mid-frequency range the linear character of the magnitude changes, still maintaining an increase but more gradual, till the point where the saturation begins \((500 \, Hz)\). The total change of the magnitude within this region is 2.54 \(dB\) for the single coil, 3.67 \(dB\) for the non-annealed metglas ribbon and 3.51 \(dB\) for the annealed metglas ribbon. Finally, saturation occurs at the high-frequency range for all three case. Comparing the top two sets of data with the reference state, the extra gain of the signal was calculated to be 2.10 \(dB\) for the non-annealed ribbon and 5.39 \(dB\) for the annealed. On the other hand, comparing the two top sets of data with each other, reveals a gain difference of 3.29 \(dB\), which corresponds approximately to 157\% increase on the magnitude between the two states. Based on these results, the annealed state of the ribbon presents better frequency response characteristics compared to the non-annealed state, and especially in the extra gain on the signal. For the rest of this work, all experiments were done with the use of annealed ribbons.
6.3.2 Linearity

One of the most important sensor characteristics is the linearity which means how linear is the sensor output signal with respect to the changes to its input. Here, the metglas ribbons are used as a vibration sensors attached on a CB, which acts as a mechanical vibration platform. The input was chosen to be the vibrator excitation voltage, which is proportional to the excitation amplitude of the beam, and the measured output was the voltage developed on the detection coil. The principle of operation of this linearity characterization experiment is the following: The mechanical vibrator sets the beam in oscillatory motion and the metglas ribbons, attached to the free end of the CB, follow the oscillation, which alters their magnetization due to the their magnetoelastic nature. The change on the magnetic state of the ribbons can be detected using a detection coil, through the Faraday’s Law of induction.

![Linearity graphs](image)

**Figure 6.2**: Metglas ribbons linearity graphs for the (a) 2nd, (b) 4th and (c) 6th bending modes of the CB. (d) Linearity comparison plot for these three bending modes.
Figs. 6.2a to 6.2c present the sensor’s linearity plots for the 2\textsuperscript{nd}, 4\textsuperscript{th} and 6\textsuperscript{th} bending modes of the CB. From these data plots it is apparent that there is a strong linearity, with $R^2$ values being very close to unity. This linearity is a combination of the vibrator linearity (oscillating amplitude versus input voltage) and the sensor linearity (output signal versus oscillating amplitude). As the vibrators are known to be very linear devices, it is concluded that the sensor shows very high linearity as well, which reveals the high coupling between the magnetic and the elastic properties of the ribbons. Plotting all three linear graphs together for comparison (Fig. 6.2d), reveals a different sensitivity for each bending mode. For example, the slope of the 4\textsuperscript{th} and 6\textsuperscript{th} bending modes is 5.21 and 11.15 times greater than the slope of the 2\textsuperscript{nd} bending mode, while the slope of the 6\textsuperscript{th} bending mode is 2.14 times greater than the slope of the 4\textsuperscript{th} bending mode.

### 6.3.3 Signal to noise ratio (SNR)

The signal to noise ratio (SNR) is another important sensor parameter that shows how high is the signal with respect to the noise, which is always present in every measurement. It is a clear number and it is defined as the ratio of the sensor signal ($V_s$) under a given stimulus, divided by the same signal ($V_n$) when the stimulus is absent. As it has been shown in Fig. 4.10b of Section 4.3.2, the magnetic state of the metglas ribbons, and thus their sensor signal, changes with the application of a DC magnetic field, which is known as the “bias field”. So, here the SNR was studied versus the bias field for a certain dynamic stress.

Figs. 6.3a to 6.3c show the SNR results for the 2\textsuperscript{nd}, 4\textsuperscript{th} and 6\textsuperscript{th} bending mode of the CB, correspondingly. All measurements were taken with the mechanical vibrator voltage fixed at $V_{rms} = 172$ mV and the SNR was calculated in dBs using Eq. (4.1), where the voltage $V_H$ was replaced with the background noise voltage $V_n$. The background noise was measured under the same operational conditions of the experiment but without the metglas ribbons attached on the CB. This was accomplished by using a second CB specimen with the same physical and geometrical properties. It can be seen that SNR increases rapidly with the magnetic field up to a certain maximum, and then decreases gradually with the...
increasing magnetic field. The magnetic field values on the maximum point seem to not change drastically between the bending modes, and their values vary within the range of $[0.8 - 0.9]$ Gauss. The percentage increase between the minimum and the maximum SNR value for the 2nd, 4th and 6th bending modes is 151%, 41% and 27%, respectively. Presenting all curves in Fig. 6.3d for comparison, it can be seen that although the SNR values for the 2nd bending mode is lower compared to the other two bending modes, the percentage increase for this bending mode is larger.

![Graphs of Metglas ribbons SNR (a) 2nd, (b) 4th, (c) 6th bending modes and (d) SNR comparison plot for these three bending modes.](image)

**Figure 6.3:** Metglas ribbons SNR graphs for the (a) 2nd, (b) 4th and (c) 6th bending modes of the CB. (d) SNR comparison plot for these three bending modes.

### 6.3.4 Quality factor

Another parameter that was studied versus the bias field was the quality factor of the resonance peaks of the bending modes, which gives an indication of the sharpness of the peak. In this case the measurements were carried out using the Matlab software program, where a simple algorithm was created to record and
calculate the FFT spectrum of the signal from the detection coil. The total record time was settled to be 50 s and the sampling rate 5 kHz. Based on these settings the frequency resolution was calculated to be 0.02 Hz.

Figure 6.4: Resonance data and fitted peaks for the (a) 2nd, (c) 4th and (e) 6th bending modes of the CB and their corresponding quality factor plots (b), (d) and (f) versus bias field.

Figs. 6.4a, 6.4c and 6.4e show the resonance data for \( B_{\text{bias}} = 0 \) Gauss, for the 2\(^{nd}\), 4\(^{th}\) and 6\(^{th}\) bending modes, respectively. The data are fit by a Cauchy-Lorentz distribution for which the quality factor is given by the relationship \( Q = f_p / \text{BW} \), where \( f_p \) is the frequency corresponding to the maximum value and \( \text{BW} \) is the bandwidth which corresponds to the full width at half maximum (FWHM).
Figs. 6.4b, 6.4d and 6.4f presents graphically the calculated values of the Q factor versus the bias field for the aforementioned modes. The range of the applied field was picked to be the same with this of the SNR experiment ([0 − 1.4] Gauss). As it can be seen, the changes of the Q factor are mostly stable for the 2nd mode, while in the case of the 4th and 6th modes there is a slight increase and saturation. Particularly, the mean value for the 2nd mode is $Q_{\text{mean}}^{2\text{nd}} = (4355 \pm 0.1\%)$, and the saturation values for the 4th and 6th modes are $Q_{\text{satur}}^{4\text{th}} = (21335 \pm 0.01\%)$ and $Q_{\text{satur}}^{6\text{th}} = (76510 \pm 0.004\%)$, respectively. Comparing these results with the SNR experiment, it can be concluded that the implementation of a DC magnetic field for the magnetization of the metglas ribbons improves the characteristics of the detectable signal, by increasing its strength (SNR) without reducing its quality (Q factor).

### 6.3.5 Stability

The time stability of the metglas ribbons was examined with respect to the frequency, for each of the three bending modes studied. The mechanical vibrator was oscillating the CB at a constant frequency for a total time of 120 min and a measurement was taken from the detection coil, through the laptop’s sound card, every 2 min. The signal was FFT analyzed with the help of the Matlab software in order to extract the frequency and the recording settings of the algorithm were the same as those used in the quality factor measurements.

Fig. 6.5 shows the stability results for all three bending modes. For the 2nd mode (Fig. 6.5a) the mean value along with the percentage deviation is $f_{\text{mean}}^{2\text{nd}} = (102.10 \pm 0.01\%)$, and for the 4th and 6th modes (Fig. 6.5b and Fig. 6.5c) is $f_{\text{mean}}^{4\text{th}} = (550.14 \pm 0.004\%)$ and $f_{\text{mean}}^{6\text{th}} = (1322.08 \pm 0.002\%)$, respectively. Also, a linear fitting on the data of Fig. 6.5 shows very small changes on the frequency values with time. Specifically, the slope of the linear fit for the 2nd, 4th and 6th modes is $(1.4 \pm 0.3) \cdot 10^{-4}\ Hz/min$, $(1.1 \pm 0.8) \cdot 10^{-4}\ Hz/min$ and $(0.7 \pm 0.9) \cdot 10^{-4}\ Hz/min$, respectively. From these results it’s clear that the metglas ribbons exhibit excellent stability with time.
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6.3.6 Repeatability

In order to study the repeatability of the sensor, the boundary conditions (applied load) of the CB were cycled between two states for a total of three cycles. Every cycle was consisted of 20 minute intervals, with each state sharing an equal time of 10 min per cycle (frequency measurements were taken every minute). The reason of picking the applied load as the changing variable in this experiment was because, as it has been shown in Fig. 4.10a the frequency values of the bending modes are directly dependent on it, and also it is easy to cycle it in order to have the precise same conditions on each cycle. The results are shown in Fig. 6.6 where the red line represents the cycling of the applied load between 2 and 9 kN, and the blue circles the measured frequency data. Table 6.1 contains the calculated mean values per cycle and the percentage deviations of the plotted data. The averages of the zeroth cycle are considered to be the references for the other cycles. The mean values are calculated in each state of the cycles and the percentage deviation...
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Figure 6.6: Metglas ribbons repeatability graphs versus time and applied load, for the (a) 2nd, (b) 4th and (c) 6th bending modes of the CB.

Table 6.1: Calculated mean values and percentage deviations for each cycle and bending mode.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Cycle</th>
<th>Load = 2 kN</th>
<th>Load = 9 kN</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Mean (Hz)</td>
<td>Deviation (%)</td>
</tr>
<tr>
<td>2nd</td>
<td>Reference</td>
<td>100.845</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>100.857</td>
<td>0.012</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>100.847</td>
<td>0.002</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>100.844</td>
<td>0.001</td>
</tr>
<tr>
<td>4th</td>
<td>Reference</td>
<td>542.677</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>542.693</td>
<td>0.003</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>542.675</td>
<td>0.000</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>542.669</td>
<td>0.001</td>
</tr>
<tr>
<td>6th</td>
<td>Reference</td>
<td>1303.244</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>1303.299</td>
<td>0.004</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>1303.273</td>
<td>0.002</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>1303.269</td>
<td>0.002</td>
</tr>
</tbody>
</table>

gives the % difference of each cycle with respect to the reference state. The average percentage deviation from the reference state is calculated to be 0.004%, 0.002% and 0.003% for the 1st, 2nd and 3rd cycle, respectively. As these percentages are very small, it can be concluded that the sensor exhibits high repeatability.
6.3.7 Sensitivity

The last parameter studied was the sensitivity of the ribbons in detecting the shift \( \Delta f \) of the natural frequencies versus CB stiffness change \( \Delta k \), after a crack was introduced. Special attention was given to the crack location because if it happens to be close or right on a mode nodal point, then it has little or no effect to the mode resonance frequency. So, the crack location for all three bending modes that were tested was picked to be close to the middle of the CB, an area which happened to be free of nodal points for all the three modes. For the calculation of the stiffness \( k \), two different variables were measured, the static deflection \( \delta \) and the applied force \( F \) at the free end of the CB, and the stiffness was calculated using the relationship \( k = F / \delta \). The measurement procedure was accomplished for seven different crack depths (CDs) and was carried out in the following steps:

1. The frequency values of the 2nd, 4th and 6th bending mode of the CB were measured dynamically, recording the sensor signal and using the FFT analysis in Matlab software program.

2. With the use of a digimatic indicator (Mitutoyo 543-690 Absolute LCD Digimatic Indicator) and a homemade applied force deflector, ten different static measurements were made on the deflection of the CB with the applied force. The calculation of the stiffness \( k \) was accomplished using the linear fitting analysis on the \( F-\delta \) plots.

3. A transverse crack was introduced close to the middle of the CB, away from the nodes of the measured bending modes.

Shown in Table 6.2 are the experimental results of the procedure described above. As it was expected, the presence of the crack alters the stiffness of the beam by reducing it, and also affects the beam’s dynamic behavior through the frequency change of each bending mode. The frequency change data for the three modes versus CD are plotted in Figs. 6.7a to 6.7c and the stiffness change data versus the CD in Fig. 6.7d. As can be seen both, frequency and stiffness, values are decreasing non-linearly versus CD, with the rate of change exponentially increased.
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From these plots no sensitivity information can be extracted as their non-linearity prevents it. However, plotting the frequency change $\Delta f$ data versus the stiffness change $\Delta k$, the non-linearity changes significantly.

Table 6.2: CB’s stiffness and bending modes frequency values, as measured experimentally for different values of crack depth, including the undamaged beam (0%). The values within the brackets correspond to the percent values of the crack depth in relation to beam thickness.

<table>
<thead>
<tr>
<th>Crack depth (mm)</th>
<th>CB stiffness (N/m)</th>
<th>Bending modes (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>2nd</td>
</tr>
<tr>
<td>0 (0 %)</td>
<td>13911</td>
<td>102.10</td>
</tr>
<tr>
<td>1 (10 %)</td>
<td>13908</td>
<td>101.82</td>
</tr>
<tr>
<td>2 (20 %)</td>
<td>13895</td>
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<tr>
<td>3 (30 %)</td>
<td>13788</td>
<td>100.98</td>
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<tr>
<td>4 (40 %)</td>
<td>13670</td>
<td>99.78</td>
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<tr>
<td>5 (50 %)</td>
<td>13372</td>
<td>97.86</td>
</tr>
<tr>
<td>6 (60 %)</td>
<td>12968</td>
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</tr>
<tr>
<td>7 (70 %)</td>
<td>12267</td>
<td>88.82</td>
</tr>
</tbody>
</table>

Figure 6.7: Frequency changes with respect to the crack depth for the (a) 2nd, (b) 4th and (c) 6th bending modes of the CB. (d) Reduction of the CB stiffness with the crack depth.

Fig. 6.8 shows the plots of $\Delta f$ data versus $\Delta k$, for all three bending modes. As it seems, a much better linearity behavior appears in these graphs and their slopes can be used as the definition of the sensor sensitivity. The vertical axis corresponds to the $\Delta f$ change of the frequency between the undamaged and damaged state of
the beam, while the horizontal axis corresponds to the $\Delta k$ change of the stiffness, for the same states of the beam. The slopes (sensitivity) of the linear fits for the 2nd, 4th and 6th bending modes are calculated to be $S_{2nd} = (7.9 \pm 0.2) \times 10^{-3}$, $S_{4th} = (34.2 \pm 0.6) \times 10^{-3}$ and $S_{6th} = (77 \pm 3) \times 10^{-3} \text{ Hz/(N}\cdot m^{-1})$, respectively. It’s clear from these results that each mode presents its own sensitivity to stiffness, with the 2nd having the smallest and the 6th the largest sensitivity. The average sensitivity value among the three bending modes is calculated to be $S_a = (38 \pm 1) \times 10^{-3} \text{ Hz/(N}\cdot m^{-1})$.

![Graphs](image)

**Figure 6.8:** Metglas ribbons sensitivity graphs with stiffness for the (a) 2nd, (b) 4th and (c) 6th bending modes of the CB.

### 6.4 Conclusions

A full characterization of the metglas ribbons as vibration sensors was discussed in this part of the thesis. Seven different sensor parameters were studied using two different experimental setups. The first setup was used to examine the frequency response of the ribbons, while the second setup, which contained a vibration platform (cantilever beam), was used to examine the rest of the parameters (linearity,
signal to noise ratio SNR, quality factor, stability, repeatability and sensitivity), through the measured bending modes frequencies of the vibrating platform (2nd, 4th and 6th bending modes).

In the case of the frequency response, two different states of the ribbon were examined, the non-annealed and the annealed state. The annealed state showed better frequency response characteristics in relation to the non-annealed, especially on the extra gain of the signal magnitude ($\approx 157\%$ increase), and for the rest of the characterization process only the annealed ribbons were used. Concerning linearity, the ribbons showed an extremely linear behavior with respect to the oscillation amplitude of the cantilever beam (CB), with the average value of the adjusted R-square from all three bending modes being $R_a^2 = 0.99995$. The SNR parameter of the ribbons was examined with respect to the DC magnetization field (bias field), and it was observed that it reaches a maximum within the range $[0.8-0.9]$ Gauss for all three bending modes. The improvement on the detectable signal, upon the application of the bias field, was about 151%, 41% and 27% for the 2nd, 4th and 6th bending modes, respectively. The resonance peak’s quality factor of each bending mode was also examined versus the bias field. It was observed that within the range $[0.8-0.9]$ Gauss where the SNR was optimum for all modes, the quality factor was stable for the 2nd mode and slightly increased for the 4th and 6th modes, concluding that the implementation of a DC magnetic field for the magnetization of the ribbons improves the characteristics of the detectable signal, by increasing its strength without reducing its quality.

The sensor stability parameter was examined within a time period of 2 hours with all the experimental setup settings (beam’s boundary conditions, vibrating voltage, ribbon’s magnetization field, etc.) fixed and stable, and showed an excellent behavior, with the average percentage deviation from the mean frequency for all bending modes being as small as 0.005%. The changes on the frequency with time for each bending mode were also small, with slopes being calculated as $f_{2nd} = (1.4 \pm 0.3) \cdot 10^{-4}$ Hz/min, $f_{4th} = (1.1 \pm 0.8) \cdot 10^{-4}$ Hz/min and $f_{6th} = (0.7 \pm 0.9) \cdot 10^{-4}$ Hz/min. On the other hand, the repeatability of the sensor was examined by subjecting the CB to two applied load values cyclically at its fixed
end (2 and 9 kN). The average percentage deviation from the reference state was 0.004%, 0.002% and 0.003% for the 1st, 2nd and 3rd cycle, respectively. Finally, the sensor’s sensitivity parameter was examined versus the stiffness change of the CB, when a transverse crack was introduced. The results showed sensitivity values of $S_{2nd} = (7.9 \pm 0.2) \cdot 10^{-3} \text{ Hz/(N \cdot m^{-1})}$, $S_{4th} = (34.2 \pm 0.6) \cdot 10^{-3} \text{ Hz/(N \cdot m^{-1})}$ and $S_{6th} = (77 \pm 3) \cdot 10^{-3} \text{ Hz/(N \cdot m^{-1})}$, indicating an increase of it as the mode number increases.
6.5 Bibliography


CHAPTER 6. CHARACTERIZATION OF MAGNETOElastIC RIBBONS AS VIBRATION SENSORS


Chapter 7

Identification of cracks on cantilever beams through their bending frequency modes

7.1 Abstract

In this chapter a methodology of identifying cracks in CBs by having as an input the first 8 bending modes of the CB is presented. The beam modeling part of the method was developed based on a finite element analysis and fracture mechanics theory, and implemented using Matlab programming. Bending mode normalized frequencies were extracted from the model, versus crack location (CL) and crack depth (CD), and all possible normalized frequency ratios (NFRs) were calculated in order to be stored in a database. This database information was used to find the optimum CL and CD values from input data, through a pattern matching process. Two different approaches were presented to validate the method, one experimental and one numerical. In the first case, a number of 19 beam specimens of aluminum alloy 6063 were used with various fixed crack locations and depths, and the frequency measurement of each bending mode was accomplished non-invasively using the sensing technique described in section 4.2. On the other hand, for the numerical approach the beam specimens were designed by ANSYS with the same physical and geometrical characteristics, and the bending modes were extracted numerically. Both validation tests showed that the proposed method is
extremely capable of predicting the crack location and quite capable of predicting
the crack depth.

7.2 Introduction

In the last few decades, damage detection appeared as one of the main fields of
research integrating many interdisciplinary fields to ensure structure safety. The
damage produces a local change in stiffness, which produces a discontinuity in
the mode shape, reduces frequency and increases damping. These changes in
the dynamic properties (modal parameters) can be measured and leads to the
identification of the damage location and severity. Based on these changes, many
engineers and scientists have devoted their time and efforts towards developing a
different method for damage detection. In [1–7], the authors reviewed articles on
the damage detection through the dynamic properties.

Among these three modal parameters the use of the frequency changes becomes
the most inexpensive structural assessment technique, as the process of measuring
the natural frequencies of a mechanical structure is simpler and easy, and can
be accomplished using only one sensor. Salawu [8] reviewed 65 articles in the
damage detection through changes in the natural frequencies. Cawly and Adam [9]
demonstrated that the ratio of frequency changes in two modes is a function of only
the damage location, and proposed a method to locate the damage by comparing
the theoretical frequency shifts due to damage at selected location with measured
ones. Gillich and Praisach [10] showed that the appearance of the curve formed
by the natural frequency shifts does not change as function of damage depth but
changes only as function of damage location. Therefore, the curves of the natural
frequency shifts are plotted for each damage location and are compared with the
measured ones, and those which most resemble indicate the location of the damage.
Gillich et al. [11, 12] showed the relationship between the frequency changes and
the normalized square mode shape curvature, and developed a correlation method
where the set of values used in the correlation are dependent only on the damage
location. Dahak et al. [13] showed that if the natural frequency of one of the
vibration modes remains unchanged, the damage is symmetrical of one of the vibration nodes of this vibration mode, and the authors used the classification of the frequencies changes approach to distinguish the node, hence locate the damage.

Many optimization algorithms have been also applied in order to compare the calculated frequencies with the measured ones. Moradi et al. [14] applied the bees algorithm, where the objective function is the weighted sum of the squared errors between the measured natural frequencies and the calculated ones using an analytical approach based on a rotational spring model. Later, Moradi et al. [15] generalized the method for multiple damages detection. The results show that the number of cracks as well as their sizes and locations can be predicted. Moezi et al. [16] extended the method using a modified cuckoo optimization algorithm. Kaminski et al. [17] used artificial neural networks as an algorithm of optimization. The Genetic Algorithms are used by the authors in [18–21]. Krawczuk [22] combined a genetic algorithm, the wave propagation approach and the gradient technique. Recently, the authors in Ding et al. [23] applied an improved artificial bee colony algorithm and Boubakir et al. [24] applied an improved accelerated random search algorithm.

Since the frequency reductions are a function of the damage location and severity, the frequency contour method appears as an attractive approach. Liang [25] and Lele and Maiti [26] used a weightless torsional spring model to develop a relationship which determines a stiffness value for a given natural frequency for various locations. In order to localize the damage, contours of various values of stiffness along a various crack location which gives constant natural frequencies are plotted. The intersections of the superposed contours will provide the damage location. Then, the damage severity is given by the use of a relationship between the damage magnitude and the stiffness. Rosales et al. [27] proposed an efficient algorithm to solve the governing differential equation of the beam named a power series technique (PST). Chinchalkar et al. [28] presented the same approach of contour method, but instead of the analytical solving of the governing differential equation to obtain the contours, the authors modelled the beam with a finite element method and the plotted superposed contours are given by the inverse problem of
finding in each location the spring stiffness value that gives the frequencies which are equal to the measured ones. Nikolakopoulos et al. [29] plotted a variation of crack depth instead of the stiffness, for various crack locations and the intersection point of the superposed contours gives directly the crack depth and its location. The authors applied the approach in frame and beam structures. Owolabi et al. [30] made experimental investigations of the effects of cracks depth and location on the first three natural frequencies. The results are used as database in order to plot the contour and detect the crack location and depth. Dong et al. [31] applied the contour line method in the rotor based on the idea that the cracked rotor is a Euler–Bernoulli beam with circular cross-section. Nahvi and Jabbari [32] proposed the use of the mode shape to determine the cracked element. Then, the authors applied the contour line method to specify the location and give the crack depth. Swamidas et al. [33] proposed to add an off-centre mass in the beam with symmetrical boundary condition to eliminate symmetrical solutions. Sinou [34] improved the contour method to predict also the crack orientation in addition to its location and depth. Mazanoglu and Sabuncu [35] combined the contour line method with an algorithm in order to detect double cracks. Banerjee et al. [36] proposed two techniques to detect a crack in beam with functionally graded materials. The first one using a frequency contour method, and the second using an optimization technique based on genetic algorithm.

As it was mentioned earlier, the process of measuring the natural frequencies of a mechanical structure can be accomplished by using only one sensor. In chapter 5 of this thesis a detailed experimental procedure was presented on detecting the natural frequencies of damaged and undamaged CBs with the use of magnetoelastic ribbons, and where the accuracy of these materials was examined in order to be used as vibration sensors. Thus, the experimental validation of the methodology presented in this part of thesis was done by using as an input the measured natural frequencies of the damaged CBs by these sensors.
7.3 Beam modeling

The modeling of the dynamic behavior of the CB through its bending modes with the presence of a crack was accomplished by using a finite element method and fracture mechanics theory. The task was carried out in three stages which were later combined in order to be programmed using Matlab software. The first stage involves the analytical extraction of the stiffness and mass matrices of the undamaged beam element using the Euler-Bernoulli beam finite element method. The second stage involves the extraction of the stiffness matrix of the damaged beam element using fracture mechanics theory. The final stage involves the assembling of all beam elements and the extraction of the global stiffness and mass matrices, considering the boundary conditions, and solving the eigenvalue problem for the bending modes of the cantilever beam.

7.3.1 The undamaged beam element

Fig. 7.1 illustrates a schematic diagram of an undamaged beam element under bending. The uniform element of length $L_e$ consists of two nodes at its two ends (red dots). Each node has two degrees of freedom (DOFs), one for translational $(u_1(t), u_3(t))$ and one for rotational $(u_2(t), u_4(t))$ displacements. The parameters $E$, $I$, $\rho$ and $A$ correspond to the modulus of elasticity, moment of inertia, material density and cross section of the element, respectively.

![Figure 7.1: Schematic diagram of the undamaged beam element under bending.](image)
Following typical approaches of FEM analysis, the equation of motion of the beam element can be derived in the form:

\[
[m_e] \{\ddot{u}(t)\} + [k_e] \{u(t)\} = \{f(t)\} \tag{7.1}
\]

where \(u(t)\) is the column matrix with the four displacements \(u_i(t)\), \(f(t)\) is the column matrix of node's force vectors and \([k_e], [m_e]\) are the elemental stiffness and mass matrices, respectively. Eq. (7.1) constitutes a different form of the Lagrange's equation of motion, which integrates the analytical expressions of the kinetic and the dynamic energies of each element (a detailed analysis on the derivation of the Lagrange's equation of motion and extraction of the mass and stiffness matrices can be found in chapter 8 of book [37]). The corresponding elemental mass and stiffness matrices can be expressed as follows:

\[
m_e = \frac{\rho A L_e}{420} \begin{bmatrix}
156 & 22L_e & 54 & -13L_e \\
22L_e & 4L_e^2 & 13L_e & -3L_e^2 \\
54 & 13L_e & 156 & -22L_e \\
-13L_e & -3L_e^2 & -22L_e & 4L_e^2
\end{bmatrix}, \quad k_e = \frac{EI}{L_e^3} \begin{bmatrix}
12 & 6L_e & -12 & 6L_e \\
6L_e & 4L_e^2 & -6L_e & 2L_e^2 \\
-12 & -6L_e & 12 & -6L_e \\
6L_e & 2L_e^2 & -6L_e & 4L_e^2
\end{bmatrix} \tag{7.2}
\]

These matrices refer to the intact state of the beam element and consists the basic information for studying the vibration behavior of the beam without damage.

### 7.3.2 The damaged beam element

The presence of a damage in the form of a crack in a structural element affects significantly the vibration behavior by reducing its local stiffness. Although the crack alters both the elemental mass and stiffness matrices, the changes on the mass matrix can be neglected due to the insignificant mass removal. Fig. 7.2 shows the schematic diagram of the beam element with only bending moments \((M_1, M_2)\) and shearing forces \((T_1, T_2)\) acting on it, under the presence of a transverse crack with depth \(s\), halfway along the element’s length. The static equilibrium condition
of the element in Fig. 7.2 is expressed as follows:

\[ \sum_{i=1}^{2} T_i = 0 \iff T_1 + T_2 = 0 \iff T_1 = -T_2 \] (7.3)

\[ \sum_{i=1}^{2} M_i = 0 \iff M_1 + M_2 + T_2 L_e = 0 \iff M_1 = -T_2 L_e - M_2 \] (7.4)

which can be summarized in matrix form as:

\[
\begin{bmatrix}
T_1 \\
M_1 \\
T_2 \\
M_2
\end{bmatrix} =
\begin{bmatrix}
-1 & 0 \\
-L_e & -1 \\
1 & 0 \\
0 & 1
\end{bmatrix}
\begin{bmatrix}
T_2 \\
M_2
\end{bmatrix} \iff
\begin{bmatrix}
T_1 \\
M_1 \\
T_2 \\
M_2
\end{bmatrix} = [A]
\begin{bmatrix}
T_2 \\
M_2
\end{bmatrix}, \quad [A] =
\begin{bmatrix}
-1 & 0 \\
-L_e & -1 \\
1 & 0 \\
0 & 1
\end{bmatrix}
\] (7.5)

The matrix \([A]\) in Eq. (7.5) is the transformation matrix of the equilibrium state of the beam element and contains all the information needed to transform the force system at the right node into the force system at the left node. In general, the transformation matrix of the beam element is related to the elemental stiffness matrix through the expression:

\[ [k_e] = [A]^T[C]^{-1}[A] \] (7.6)
CHAPTER 7. IDENTIFICATION OF CRACKS ON CANTILEVER BEAMS THROUGH THEIR BENDING FREQUENCY MODES

where $[A]^T$ is the transpose of the transformation matrix and $[C]^{-1}$ the inverse of the flexibility matrix. The flexibility matrix is associated with the strain energy $U$ of the beam element and is a property of the structure alone, and does not depend upon the loads on the structure. The general expression for the flexibility matrix coefficients is:

$$C_{ij} = \frac{\partial^2 U}{\partial F_i \partial F_j}$$

(7.7)

where $F_i$ and $F_j$ correspond to the type of the forces applied to the element. In the case of a beam element under bending the $F_i = T_i$ and the $F_j = M_j$, with $i, j = 1, 2$ due to the number of the nodes, which is 2. Using Eqs. (7.5) to (7.7) the elemental stiffness matrix is obtained as:

$$[k_e] = \begin{bmatrix}
    C_{22} & L_e C_{22} - C_{12} & -C_{22} & C_{12} \\
    L_e C_{22} - C_{21} & L_e^2 C_{22} - L_e (C_{12} + C_{21}) + C_{11} & -L_e C_{22} + C_{21} & L_e C_{12} - C_{11} \\
    -C_{22} & -L_e C_{22} + C_{12} & C_{22} & -C_{12} \\
    C_{21} & L_e C_{21} - C_{11} & -C_{21} & C_{11}
\end{bmatrix}$$

(7.8)

In order to calculate the flexibility coefficients $C_{ij}$, the strain energy of the beam element must be derived. For the undamaged beam element, which is subjected to total a bending moment $M_T = TL + M$, the strain energy is given by:

$$U_o = \frac{1}{2} \int_0^{L_e} \frac{M_T^2}{EI} dL = \frac{1}{2} \int_0^{L_e} \frac{(TL + M)^2}{EI} dL = \frac{L_e}{2EI} \left[ \frac{(TL_e)^2}{3} + M^2 + TMLe \right]$$

(7.9)

Using Eq. (7.7) and Eq. (7.9) to calculate the coefficients of the flexibility matrix and introducing them into Eq. (7.8), we get the stiffness matrix of Eq. (7.2) for the undamaged beam element. However, when a crack is introduced to the beam element the strain energy in Eq. (7.9) must be written with an additional energy term that is related to the properties of the crack, and specifically with the relative modes of the crack surface displacements.

There are three basic modes of the crack surface displacements and they are shown in Fig. 7.3. Mode I (Fig. 7.3a) is the "opening" or "tensile" mode where the crack surfaces are displaced normal to themselves. Mode II (Fig. 7.3b) is the
"sliding" or "in-plane shear" mode where the crack surfaces slide over one another in a direction perpendicular to the leading edge of the crack. Finally, Mode III (Fig. 7.3c) is the "tearing and anti-plane shear" mode where the crack surfaces move relative to one another and parallel to the leading edge of the crack.

According to Tada et al. [38], the additional term (due to the crack) that comes into the expression of strain energy is given by:

\[ U_1 = \int_S \frac{1}{E'} \left[ K_I^2 + K_{II}^2 + \frac{E'}{2\mu} K_{III}^2 \right] dS \quad (7.10) \]

where \( E' = E/1 - v^2 \) is the plane strain Young’s modulus, \( \mu = E/2(1 + v) \) is the shear modulus, \( E \) and \( v \) the Young’s modulus and the Poisson’s ratio, respectively, \( K_I, K_{II} \) and \( K_{III} \) are the crack-tip stress intensity factors for each mode with the indexes indicating the respective mode, and \( S \) is the crack cross section. The \( K_I, K_{II} \) and \( K_{III} \) factors represent the strength of the stress field surrounding the crack tip and physically may be regarded as the intensity of the transmitted load through the crack-tip region. In our case, the cracked beam element of Fig. 7.2 has width \( w \), height \( h \) and crack depth \( s \). When only shear forces and bending moments are applied, as is the case of a beam under bending, these factors can be expressed as:

\[ K_I = K_{IT} + K_{IM}, \quad K_{II} = K_{IIT}, \quad K_{III} = 0 \quad (7.11) \]

The above relations express the fact that there are two contributing components to the \( K \) factors, one due to shear forces (\( T \)) and one due to bending moments (\( M \)). Because the \( K \) factors depend linearly on the stresses, the superposition principle
applies and these two components are added together. In particular, in Mode I both components are present, in Mode II only the shear component is present and in Mode III none of them contributes to the crack formation. By applying these relations to Eq. (7.10), and taking into account that the crack cross-section area is equal to \( S = w \times s \), we get:

\[
U_1 = \int_0^s \frac{w}{E'} \left[ (K_{IT} + K_{IM})^2 + K_{IIT}^2 \right] ds \quad (7.12)
\]

The factors \( K_{IT} \), \( K_{IM} \) and \( K_{IIT} \) in Eq. (7.12) are functions of both the external forces and the size of the crack, and according to Tada et al. [38] they are given as:

\[
K_{IT} = \sigma_{IT} \sqrt{\pi s F_I(\lambda)} , \quad K_{IM} = \sigma_{IM} \sqrt{\pi s F_I(\lambda)} , \quad K_{IIT} = \sigma_{IIT} \sqrt{\pi s F_{II}(\lambda)} \quad (7.13)
\]

where

\[
F_I(\lambda) = \sqrt{\frac{2}{\pi \lambda}} \tan \frac{\pi \lambda}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi \lambda}{2})^4}{\cos \frac{\pi \lambda}{2}} \right), \quad F_{II}(\lambda) = \frac{1.122 - 0.561 \lambda + 0.085 \lambda^2 + 0.18 \lambda^3}{\sqrt{1 - \lambda}} \quad (7.14)
\]

and

\[
\sigma_{IT} = \frac{3TL_e}{wh^2}, \quad \sigma_{IM} = \frac{6M}{wh^2}, \quad \sigma_{IIT} = \frac{T}{wh} \quad (7.15)
\]

The \( \sigma_{IT} \), \( \sigma_{IM} \) are the bending stresses applied by shear and bending forces, respectively, in order to open the crack, the \( \sigma_{IIT} \) is the shear stress applied by shear forces in order to slide the crack, the \( F_I(\lambda) \), \( F_{II}(\lambda) \) are the correction functions of the intensity factors of each respective mode and \( \lambda = s/h \) is the normalized crack depth. By substituting the above expressions (Eqs. (7.14) and (7.15)) into the factors of Eq. (7.13), the explicit forms of the \( K \) are:

\[
K_{IT} = \left[ \frac{3TL_e}{wh^2} \right] \sqrt{\pi \lambda h} \left[ \sqrt{\frac{2}{\pi \lambda}} \tan \frac{\pi \lambda}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi \lambda}{2})^4}{\cos \frac{\pi \lambda}{2}} \right) \right] \quad (7.16)
\]

\[
K_{IM} = \left[ \frac{6M}{wh^2} \right] \sqrt{\pi \lambda h} \left[ \sqrt{\frac{2}{\pi \lambda}} \tan \frac{\pi \lambda}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi \lambda}{2})^4}{\cos \frac{\pi \lambda}{2}} \right) \right]
\]

\[
K_{IIT} = \left[ \frac{T}{wh} \right] \sqrt{\pi \lambda h} \left[ \frac{1.122 - 0.561 \lambda + 0.085 \lambda^2 + 0.18 \lambda^3}{\sqrt{1 - \lambda}} \right]
\]
From Eqs. (7.9), (7.12) and (7.16) the total strain energy \( U = U_o + U_1 \) of the cracked beam element is given as:

\[
U = \frac{L_e}{2EI} \left[ \frac{(TL_e)^2}{3} + M^2 + TML_e \right] + \frac{\pi(1-v^2)}{Ew} \left[ \left( \frac{3TL_e}{h} + 6M \right)^2 \int_0^\lambda \lambda F_i^2(\lambda)d\lambda + T^2 \int_0^\lambda \lambda F_{II}^2(\lambda)d\lambda \right]
\]

This is the complete form of the strain energy for the beam element and contains both the damaged and the undamaged expressions of it. Finally, introducing the above strain energy equation into Eq. (7.7) the flexibility coefficients can be calculated. The result is:

\[
C_{11} = \frac{L_e^3}{3EI} + \frac{2\pi(1-v^2)}{Ew} \left[ \frac{9L^2}{h^2} \int_0^\lambda \lambda F_i^2(\lambda)d\lambda + \int_0^\lambda \lambda F_{II}^2(\lambda)d\lambda \right]
\]

\[
C_{12} = C_{21} = \frac{L_e^2}{2EI} + \frac{36\pi L_e(1-v^2)}{Ewh^2} \int_0^\lambda \lambda F_i^2(\lambda)d\lambda
\]

\[
C_{22} = \frac{L_e}{EI} + \frac{72\pi(1-v^2)}{Ewh^2} \int_0^\lambda \lambda F_{II}^2(\lambda)d\lambda
\]

The next step is to introduce these coefficients into Eq. (7.8) and calculate the elemental stiffness matrix. As it has been mentioned above, the changes on the mass matrix of the beam element with the presence of the crack can be neglected, leaving thus the matrix \([m_e]\) from Eq. (7.2) unchanged. The final step of the beam modeling involves the assembling of the whole beam, considering the boundary conditions, and the extraction of the bending modes through the solution of the eigenvalue problem (in chapter 8 of this book [37] there is an example of the assembling procedure of a discretized cantilever beam, as well as solution of the eigenvalue problem).

### 7.4 Beam model simulation results

The simulation of the model presented in the previous section was carried out using the Matlab software program. A cantilever beam with \(N_e = 2000\) elements was considered, the geometric and physical characteristics of which are shown in Table 7.1. These characteristics were selected based on the experimental specimens that were used for the evaluation of this method. As it is known, the number of
elements used to simulate the beam is related to the accuracy of the method because it discriminates the beam in smaller and smaller pieces, and thus the detection of the crack can be done with less error. Also, a sparse matrix method was used to reduce significantly the elapse time of the algorithm due to the high number of elements.

### Table 7.1: Physical and geometrical characteristics of the simulated beam.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>350 mm</td>
</tr>
<tr>
<td>Width</td>
<td>30.25 mm</td>
</tr>
<tr>
<td>Height</td>
<td>8.18 mm</td>
</tr>
<tr>
<td>Material density</td>
<td>2.69 gr/cm³</td>
</tr>
<tr>
<td>Cross section</td>
<td>247.45 mm²</td>
</tr>
<tr>
<td>Moment of inertia (easy axis)</td>
<td>1379.8 mm⁴</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>68.3 GPa</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.33</td>
</tr>
</tbody>
</table>

#### 7.4.1 Normalized frequency variations

Fig. 7.4 shows the variation of the first 8 bending modes of the simulated cantilever beam versus crack location (CL) and crack depth (CD). The vertical axis corresponds to the normalized frequency (NF) values of the beam, which is equal to the ratio $f_{\text{damaged}}/f_{\text{undamaged}}$, where $f_{\text{damaged}}$ and $f_{\text{undamaged}}$ are the bending modes of the cracked and cracked-free beam, respectively. The horizontal axis corresponds to the normalized values of the crack location (NCL) with values between 0 and 1 (with 0 corresponding to the fixed end and 1 to the free end), and $\text{step} = 1/N_e = 0.0005$.

There are a total number of 14 curves in each graph of Fig. 7.4, each corresponding to different CD ratios (the caption in Fig. 7.4a shows the variation of the CDs in percentages). A common feature of all the curves in Fig. 7.4 is that none of them contains values greater than 1. This indicates that, according to the model, the presence of a transverse crack on the beam always tends to reduce the frequency values of the bending modes, since the ratio $f_{\text{damaged}}/f_{\text{undamaged}}$ is always below 1, regardless of the CL and CD. Another common feature is that there is at least one point where all curves converge to unity. The magnification
CHAPTER 7. IDENTIFICATION OF CRACKS ON CANTILEVER BEAMS THROUGH THEIR BENDING FREQUENCY MODES

Figure 7.4: Normalized mode frequencies versus normalized crack location and depth of the simulated cantilever beam for the bending modes: a) 1st, b) 2nd, c) 3rd, d) 4th, e) 5th, f) 6th, g) 7th, h) 8th.
of the converging points shown in Fig. 7.4b reveal that the converge to unity is only approximate and not exact. The same is true for all graphs. For the first bending mode (Fig. 7.4a) the converging point is the free end of the beam, while for the rest bending modes (Figs. 7.4b to 7.4h) there are more than one converging points along the beam, including the free end. Since at the converging points \( f_{\text{damaged}}/f_{\text{undamaged}} \approx 1 \), the corresponding bending mode frequency is almost the same as the one in the undamaged beam. This is happening because the converging points of each mode are identical to the beam nodes, where no displacements take place. In Section 5.6 this behavior was also observed experimentally. Additionally, it was observed that when the CL was symmetrical in between two nodes (highly deformation point), the changes in the corresponding frequency were maximal. Although these rules are valid for all the nodes along the beam, there are two points on the cantilever beam which are exceptions and the opposite rule holds, and these points are the ends of the beam.

### 7.4.2 Normalized frequency ratio variations

The graphs in Fig. 7.4 show the behavior of each bending mode individually with the presence of the crack. Even though in principle these graphs could have been used to extract the crack location and depth, in practice this approach leads to large errors due to the small sample size (8 NFs values), and since the proposed methodology (described in the next section) is based on the evaluation of the variance between the predicted and experimental values, a larger number of sample values can increase the precision. Thus, this number can be greatly increased to 28 by taking all possible combinations of ratios of normalized frequencies between the different NFs (from now on, the ratios \( f_{nm} = f_n/f_m \), where \( f_n \) and \( f_m \) are the NF of the bending modes \( n \) and \( m \) respectively, will be referred as NFR for simplicity). For example, Fig. 7.5 shows some of the NFR graphs which have the same horizontal axis as the graphs in Fig. 7.4. Due to the high number of graphical representations (totally 28 different graphs), only a small number of them are shown for clarity in Fig. 7.5, particularly the combinations \( f_{12}, f_{13}, f_{23} \) and \( f_{24} \) (all 28 NFR graphs can be found in Appendix B).
CHAPTER 7. IDENTIFICATION OF CRACKS ON CANTILEVER BEAMS THROUGH THEIR BENDING FREQUENCY MODES

Comparing the graphs of Fig. 7.4 and Fig. 7.5 we can see that in case of the NFR graphs there are values higher than unity and with at least one cross-one point (the point where the curves cross the unity). For example, the curves in Fig. 7.5a intersect the unity at a point close to $NCL = 0.4$, which is the only cross-one point, while the curves in Figs. 7.5b to 7.5d intersects the unity in more than one points. Also, the cross-one point of each curve varies with CD. The three insets shown in Fig. 7.5c correspond to the cross-one points of the curves at a greater magnification and it can be seen that each curve intersects the unity in different NCL points (the scales are the same for all three insets). For the first cross-one point (down-left inset) the intersection does not appear to vary significantly, while for the other two points (middle-up and down-right insets) the variation seems to be larger. From the middle-up and down-right insets, it can been seen that the variation of the cross-one points can be forward or backward too. This characteristic appears in all 28 combinations of the bending modes.

Figure 7.5: Ratios of normalized frequencies of the simulated bending modes versus normalized crack location and depth: a) $f_{12}$, b) $f_{13}$, c) $f_{23}$, d) $f_{24}$. 
Taking now the graph of Fig. 7.5a, it can be said that it is divided into two non-symmetric regions. For these regions the ratio of $f_1/f_2$ varies depending on whether the crack appears on the left or the right side of cross-one point. On the left region the ratio $f_1/f_2$ is always below the unity ($f_1/f_2 < 1$), while on the right region is always above the unity ($f_1/f_2 > 1$), creating thus a condition which determines the location of the crack. The same conclusion is obtained on the curves of Figs. 7.5b to 7.5d but with four regions, where the ratio $f_n/f_m$ is switching alternatively ($f_n/f_m > 1$ and $f_n/f_m < 1$). Similarly, in Fig. 7.6 (NFR $f_{78}$) there are 14 such regions. From a physical point of view, the cross-one points of the NFR curves sets the boundaries between different regions on the beam and determines the relative behavior among the NFs with the presence of the crack.

Figure 7.6: Normalized frequency ratio $f_{78}$ versus normalized crack depth and location (The color indexing is the same as in the previous figures).

7.5 Crack localization and quantification method

According to the simulation results of the beam model described above, for each CL and CD on the beam there is a set of 28 NFR values which can be stored in a database in the form of patterns. These patterns can be described as holes in the puzzle whose shape is determined by the set of NFR values, and for which
there is ideally a single match. Thus, the idea of this methodology is to extract
the optimum match through a pattern matching process, given as an input a set
of 28 NFR values. The steps consisting this matching process are the following:

1. Collection of the set of the first 8 bending mode frequencies for the undam-
aged cantilever beam \( f_{\text{undamaged}} \).
2. Repetition of the step 1 for the case of the damaged beam \( f_{\text{damaged}} \).
3. Calculation of the NF values using \( f_{\text{damaged}} / f_{\text{undamaged}} \).
4. Determination of all NFRs by using all possible combinations between the
NF values in step 3.
5. Evaluation of the variance between the input NFRs \( (NFR_{\text{input}}) \) and database
NFRs \( (NFR_{\text{model}}) \) for each CL and CD value in the database, using the
following equation:

\[
V = \frac{1}{N} \sum_{i=1}^{N} (NFR_{i,\text{model}} - NFR_{i,\text{input}})^2 \tag{7.19}
\]

where \( N = 28 \).
6. Extraction of the optimum (CL, CD) values through the minimization pro-
cess of \( V \).

In order to give an example and test the validity of this damage detection method-
ology, two different test methods were used as described in the next two subsec-
tions. In Section 7.5.1 the eight bending modes were produced computationally
by ANSYS 2016 modal analysis simulations, while in Section 7.5.2 they were mea-
sured experimentally using magnetoelastic ribbons.

### 7.5.1 ANSYS verification

To make a thorough test on the validity of the method, a series of simulations in
ANSYS modal analysis were carried out using the physical and geometrical char-
acteristics of Table 7.1. In particular, 19 designs of different cantilever beams were
made which correspond to 18 beams with a single crack at prescribed positions,
plus one undamaged beam. CLs were chosen to be at the points 50 mm, 100 mm,
150 mm, 200 mm, 250 mm and 300 mm from the fixed end, and the CD for each location was varied from 1 mm to 3 mm with step of 1 mm. Table 7.2 shows the numerical frequencies calculated by ANSYS for all CLs and CDs. The next step was to calculate the NFRs from these values and fed them as inputs to the variance method described above using Matlab software program.

**Table 7.2:** Cantilever beam bending mode frequency values as calculated using ANSYS modal analysis for all fixed CLs and CDs, including the undamaged beam.

<table>
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<tr>
<th>CL (mm)</th>
<th>CD (mm)</th>
<th>( f_1 ) (Hz)</th>
<th>( f_2 ) (Hz)</th>
<th>( f_3 ) (Hz)</th>
<th>( f_4 ) (Hz)</th>
<th>( f_5 ) (Hz)</th>
<th>( f_6 ) (Hz)</th>
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<th>( f_8 ) (Hz)</th>
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Tables 7.3 and 7.4 show respectively the CL and CD results of the method as they are predicted by the model, and the percentage deviations from the fixed values used in ANSYS. It can be seen that in the CL case most of the error values are below 1%, thus indicating an excellent prediction ability of the method concerning the crack location. Also, the percent error values seem to drop as we move away from the fixed end of the beam, with the highest values being mainly at the point CL=50 mm. On the other hand, the predicted CD values appear to have error values higher than 1%, with most of the them being from [5-10]%. Although these deviations are higher compared to the CL ones, they are still within an acceptable range showing thus very good prediction ability of the method concerning the CD.
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Table 7.3: Results of comparison between the predicted CL values and the fixed ones used in ANSYS, by the proposed method.

<table>
<thead>
<tr>
<th>CD (mm)</th>
<th>ANSYS CL (mm)</th>
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<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
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<td>1</td>
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<td>200.5</td>
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<td>Error(%)</td>
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<td>0.2</td>
<td>0.0</td>
<td>0.2</td>
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<td>Model (mm)</td>
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<td>200.4</td>
<td>250.4</td>
<td>299.8</td>
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<td></td>
<td>Error(%)</td>
<td>2.2</td>
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<td>0.3</td>
<td>0.2</td>
<td>0.2</td>
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<td>250.6</td>
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<tr>
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<td>Error(%)</td>
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Table 7.4: Results of comparison between the predicted CD values and the fixed ones used in ANSYS, by the proposed method.

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<th>ANSYS CL (mm)</th>
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<td>Error(%)</td>
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<td>Error(%)</td>
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<td>Error(%)</td>
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<td>6.3</td>
<td>6.3</td>
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7.5.2 Experimental verification

The next step was to test the efficiency of the method on predicting the CL and CD values on real beam specimens through their bending modes measured by magnetoelastic ribbons. In order to do so, a total number of 19 beam specimens were used (Fig. 7.7) out of which, 1 was damage free and the other 18 had damages at fixed locations and depths, same as in the previous subsection. The frequency measurement procedure was the same as the one described in the Section 4.2.

Figure 7.7: The beam specimens as they were prepared for experiment. The single beam at the bottom of the figure is the damage-free beam.
Table 7.5 contains the experimentally frequency values of the bending modes for all the beam specimens. Making a quick comparison of these frequency values with those predicted in ANSYS modal analysis (Table 7.2), it can be seen that they are very close to each other. For example, comparing the frequency values for the undamaged beam the deviation is: Dev - $f_1 = 0.18\%$, Dev - $f_2 = 0.09\%$, Dev - $f_3 = 0.05\%$, Dev - $f_4 = 0.08\%$, Dev - $f_5 = 0.19\%$, Dev - $f_6 = 0.41\%$, Dev - $f_7 = 0.62\%$ and Dev - $f_8 = 0.44\%$.

Table 7.5: Cantilever beam bending modes frequency values as measured experimentally for all CLs and CDs, including the undamaged beam.

<table>
<thead>
<tr>
<th>CL (mm)</th>
<th>CD (mm)</th>
<th>$f_1$(Hz)</th>
<th>$f_2$(Hz)</th>
<th>$f_3$(Hz)</th>
<th>$f_4$(Hz)</th>
<th>$f_5$(Hz)</th>
<th>$f_6$(Hz)</th>
<th>$f_7$(Hz)</th>
<th>$f_8$(Hz)</th>
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<td>4492.9</td>
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<td>4442.5</td>
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<td>1785.0</td>
<td>2848.2</td>
<td>4329.2</td>
</tr>
</tbody>
</table>

Tables 7.6 and 7.7 show respectively the CL and the CD results as they are predicted by the model, and the percentage deviations from the fixed values used on the experiment. The deviations in these tables seem to be a bit higher compared to the ones in Tables 7.3 and 7.4. For example, on Table 7.3 the maximum CL error of 3.2% appears at the values ($CL = 50 \text{mm}$, $CD = 3 \text{mm}$), while the maximum CL error of Table 7.6 appears at the same pair (CL, CD) but with the value of 6.0%. At the same pace, the maximum CD error values on Tables 7.4 and 7.7 appears at the values ($CL = 50 \text{mm}$, $CD = 1 \text{mm}$), ($CL = 300 \text{mm}$, $CD = 1 \text{mm}$) and ($CL = 50 \text{mm}$, $CD = 3 \text{mm}$), and is equal to 14% and 20%, respectively.
CHAPTER 7. IDENTIFICATION OF CRACKS ON CANTILEVER BEAMS THROUGH THEIR BENDING FREQUENCY MODES

Table 7.6: Results of comparison between the predicted CL values and the fixed ones used in the experiment, by the proposed method.

<table>
<thead>
<tr>
<th>CD (mm)</th>
<th>EXP CL (mm)</th>
<th>50</th>
<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Model (mm)</td>
<td>51.6</td>
<td>102.2</td>
<td>150.5</td>
<td>199.1</td>
<td>247.1</td>
<td>294.2</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>3.2</td>
<td>2.2</td>
<td>0.3</td>
<td>0.4</td>
<td>1.2</td>
<td>1.9</td>
</tr>
<tr>
<td>2</td>
<td>Model (mm)</td>
<td>52.3</td>
<td>102.0</td>
<td>153.6</td>
<td>200.4</td>
<td>246.8</td>
<td>298.9</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>4.6</td>
<td>2.0</td>
<td>2.4</td>
<td>0.2</td>
<td>1.3</td>
<td>0.4</td>
</tr>
<tr>
<td>3</td>
<td>Model (mm)</td>
<td>53.0</td>
<td>101.8</td>
<td>151.4</td>
<td>201.2</td>
<td>250.4</td>
<td>298.2</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>6.0</td>
<td>1.8</td>
<td>0.9</td>
<td>0.6</td>
<td>0.2</td>
<td>0.6</td>
</tr>
</tbody>
</table>

Table 7.7: Results of comparison between the predicted CD values and the fixed ones used in the experiment, by the proposed method.

<table>
<thead>
<tr>
<th>CD (mm)</th>
<th>EXP CL (mm)</th>
<th>50</th>
<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
<th>300</th>
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<tbody>
<tr>
<td>1</td>
<td>Model (mm)</td>
<td>1.11</td>
<td>1.19</td>
<td>0.90</td>
<td>1.06</td>
<td>1.17</td>
<td>1.06</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>11.0</td>
<td>19.0</td>
<td>10.0</td>
<td>6.0</td>
<td>17.0</td>
<td>6.0</td>
</tr>
<tr>
<td>2</td>
<td>Model (mm)</td>
<td>2.25</td>
<td>2.37</td>
<td>1.96</td>
<td>2.13</td>
<td>2.29</td>
<td>2.13</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>12.5</td>
<td>18.5</td>
<td>2.0</td>
<td>6.5</td>
<td>14.5</td>
<td>6.5</td>
</tr>
<tr>
<td>3</td>
<td>Model (mm)</td>
<td>3.60</td>
<td>3.27</td>
<td>3.35</td>
<td>3.35</td>
<td>3.44</td>
<td>3.27</td>
</tr>
<tr>
<td></td>
<td>Error(%)</td>
<td>20.0</td>
<td>9.0</td>
<td>11.7</td>
<td>11.7</td>
<td>14.7</td>
<td>9.0</td>
</tr>
</tbody>
</table>

The higher deviations in the case of the experimental results are mainly due to the extra error factors that come into the environment of the experiment, such as, not all the beams specimens are at the perfect shape and dimensions, the temperature of the specimens, the boundary conditions, the efficiency factor of the magnetoelastic sensor etc. However, the predicted CL and CD from the method are in a very good agreement with the actual values, thus indicating the efficiency of the method in being able to detect the location and the intensity of the crack, as well as the capability of the magnetoelastic sensor to provide the vital information needed for the damage detection.

7.6 Conclusions

A methodology on localization and quantification of crack damages on cantilever beams with the use of magnetoelastic sensors is presented here. To develop the methodology, the modeling of the dynamic behavior of a damaged and undamaged cantilever beam was accomplished using finite element method and fracture mechanics theory. While the changes on the mass matrix can be neglected with
the presence of the crack on the beam element, the stiffness matrix is significantly affected, altering the vibrating behavior of the beam. The implementation of the beam model was carried out using the Matlab software program, and the normalized frequency (NF) values of the first 8 bending modes were calculated for every crack location (CL) lengthwise in steps of 1/2000 and every crack depth (CD) in the range of 0 to 65% with a step of 5%. A graphical representation of the calculated NFs is included in this work.

The validity of the method was tested both with numerical and experimental data from damaged beams with a variety of fixed CLs and CDs, and the predicted model results showed a very good agreement with them. Specifically, the average CL and CD error values were 0.7% and 8.5%, respectively, using numerical data and 1.7% and 11.4%, respectively, using experimental data. The numerical bending mode frequencies were produced using ANSYS modal analysis and the simulation of a total 19 cantilever beams (1 undamaged beam and 18 damaged beams in fixed locations and depths) were made. In correspondence, the experimental bending modes frequencies were measured using ribbons of magnetoelastic material Metglas 2826MB3 as vibration sensors. The geometry and the material of the beam specimens were the same as with the ANSYS model beams.
CHAPTER 7. IDENTIFICATION OF CRACKS ON CANTILEVER BEAMS THROUGH THEIR BENDING FREQUENCY MODES

7.7 Bibliography


8.1 Summary - Conclusions

In this thesis magnetoelastic ribbons of metallic glass alloy known as Metglas 2826MB were studied as vibration sensors. The whole study involved three different stages in which the ribbons were tested. The first stage related to the "proof of concept" of this work, which is the suitability of magnetoelastic ribbons as vibration sensors in detecting the natural frequencies of mechanical structures such as cantilever beams. The second stage involved the process of fully characterizing these materials as vibration sensors, while the third and final stage was the application of these sensors in identifying defects in the form of cracks in cantilever beams, through the shifting of the natural frequencies of the beam.

As far as the first stage is concerned, two main questions needed to be answered in order to prove the concept which are both related to the definition of magnetoelasticity, which is: Magnetoelasticity is the property of some ferromagnetic materials to change their magnetic state (magnetization) when they are stressed mechanically. The questions were the following:

1. Can these materials sense and transmit the vibrational state of a mechanical structure?
2. How do they react to the change of the vibrational state of the structure due to damages?

To deal with the first question, the metglas ribbons were attached at the free end of a cantilever beam. The beam was excited mechanically and let free to vibrate, along with the ribbons. The FFT spectrum of the recorded signal, using a coil, revealed different frequency peaks which corresponded to the bending modes of the cantilever beam, as revealed by the corresponding ANSYS modal analysis. With these results, it turned out that the magnetoelastic ribbons can sense and transmit the vibrational state of a mechanical structure. Next, a total number of 31 cantilever beam specimens of the same dimensions but with a single transverse crack at different positions and depths were prepared in order to deal with the second question. The measured natural frequencies of each beam were compared to computational ones using ANSYS program and the comparison results showed an excellent agreement between them, with most of the average deviation values being below 0.5%. Thus, with this analysis it was concluded that the magnetoelastic materials used can be used as vibration sensors as well as sensing mechanism for damage detection.

In the second stage, the magnetoelastic ribbons are characterized as vibration sensors. The process involved seven different sensor parameters such as the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. These parameters are very useful for structural health monitoring (SHM) applications where the monitoring is carried out by measuring the natural vibrations of the structure. From the results, it was shown that the ribbons can operate stably over a wide frequency range, including almost all the audio spectrum, and exhibit strong linearity versus the amplitude of the vibrations. Also, it was concluded that the implementation of a DC magnetic field, for the magnetization of the ribbons, improves the detectable signal, without reducing its quality.

The final stage involved the application of the under consideration vibration sensors to detect and identify cracks in cantilever beams, through a proposed crack identification methodology. The methodology was divided into two parts. The first
part was the modeling part, where the dynamic behavior of the cantilever beam, with the presence of a crack, was modeled by using FEM analysis and fracture mechanics theory, and stored as a database in the form of patterns. The second part was related to the implementation of a pattern matching process in order to identify the cracks, having as an input the measured natural vibrations of the cantilever beam. The results showed that the used method is extremely capable of predicting the crack location and quite capable of predicting the crack depth, due to the high natural vibration orders detected from the metglas ribbons.

8.2 Suggestions for future research

The following recommendations for future research contribute to continuing this thesis experimental work:

- Comparison of the characteristic parameters of this sensor with other contact vibration sensors such as, piezoelectric, fiber optic, strain gauge, accelerometer etc.

- Study and characterization of different types of metallic glass ribbons as vibration sensors. For example, metallic glasses with a high induction saturation value (Metglas 2605SA1, SI=1.56 T as-cast) may exhibit a better signal strength in the low-frequency range (<100 Hz), compared to the low induction saturation value metallic glasses (Metglas 2826MB, SI=0.38 T as-cast).

- Optimization of the sensor’s performance in terms of its length. The length of the ribbons used in this thesis were 100 mm.

- Computational and experimental determination of the radius of curvature at the bend, below of which the sensor is operating non-linearly. This study is more related to the bending frequencies of a mechanical structure, where the sensor tends to follow the motion of the structure.
➢ Investigate the applicability of the sensor in higher temperature vibration sensing applications, where the environment temperature is much higher than the room temperature and close to the curie temperature of the sensor’s material.

➢ Direct usage of the sensor in large scale mechanical structures, metallic and concrete, for vibration-based SHM process.

➢ Optimization of the current crack detection methodology in order to extent it in different material and geometry cantilever beams, and creating one global database which will include all the natural frequency correction factors related to the material and the geometry of the structure. This way, the time-consuming model simulations can be avoided whenever the material or geometry of the structure changes.

➢ Usage of the metglas ribbons in the experimental study on the amplitude of the frequency peaks, versus crack location and depth, in mechanical structures such as cantilever beams, for crack detection and/or identification.
Appendices
Appendix A

Matlab Scripts and Functions Used in the Thesis

Figure A.1: Function used to record the output signal from the detection coil. It takes as an input the total desired recording time and the sample rate, and outputs the voltage values of the signal.

```matlab
function [Data] = Recording(Rec_Time,Sample_Rate)
   GetData = audiorecorder(Sample_Rate, 16, 1, 1);
disp('Recording...')
recordblocking(GetData,Rec_Time);
disp('End of Recording.')
Data = getaudiodata(GetData);
end
```

Figure A.2: Function used to FFT the recorded signal in order to get the frequency spectrum of it. The function takes as an input the recorded data and the sample rate, and outputs all the frequency values along with their amplitudes.

```matlab
function [Freq,Ampl]=fastfourier(Data,Sample_Rate)
    T = 1/Sample_Rate; L = length(Data); t = (0:L-1)*T;
    NFFT = 2^nextpow2(L);
    Ampl = fft(Data,NFFT)/L;
    Freq = Sample_Rate/2*linspace(0,1,NFFT/2+1);
    Ampl=2*abs(Ampl(1:NFFT/2+1));
end
```

Figure A.3: Function used to detect and store the frequency peak values of the bending modes. As an input it takes the FFT spectrum of the recorded signal, and gives as as output the positions and the amplitudes of the resonance peaks.

```matlab
function [Mark_X,Mark_Y]=Pos(Freq,Ampl)
    [~,Position]=findpeaks(Ampl,'MinPeakHeight',1.5e-3,'MinPeakDistance',1000);
    Mark_X=Freq(Position); Mark_Y=Ampl(Position);
    plot(Freq,Ampl); hold on;
    plot(Mark_X,Mark_Y,'rs', 'MarkerFaceColor', 'b'); xlim([0 7000]);
end
```
APPENDIX A. MATLAB SCRIPTS AND FUNCTIONS USED IN THE THESIS

Figure A.4: Script used to calculate all the bending mode frequencies of the cantilever beam, for the crack identification methodology.

```matlab
% EN = Number of elements, DOFs = Global degree of freedoms, v = Total beam's length
% b = Beam's height, r = Material density, w = Normalized crack depth, n = Bending modes
% E = Modulus of elasticity, BC = Boundary conditions, s = Percentage crack depth, GFS = All frequencies
EN=2000; DOFs=2*EN+2; v=0.35; TL=0.35; b=0.00588; r=2650;
E=2.03025; [K,M]=E*El; nM=Ex*Bc; BC=[1 2]; mE=0.05*0.05EhK;
L = Element's length, L = Number of inertia, A = Beam's cross section
L=TL/EN; v=2*E/3*E; L=2*b; [K,M] = KM_Matrices(K,L,n,A); %
Co = Uncovered element flexibility matrix, T = Transformation matrix, Tt = Transposed transformation matrix
C = Flexibility matrix constant, K = Sparse global mass matrix
Co = ([L^3]/[3v^3]); ([L^3]/[3v^3]); ([L^3]/[2v^2]); L/E]; T = [-1 0 1 0; 0 -1 0];
C = ([2p1^1(1-v^2)p1^1]/Ew); ([w^1p1^1(1-v^2)p1^1]/Ew); ([2p1^1(1-v^2)p1^1]/Ew); ([w^1p1^1(1-v^2)p1^1]/Ew); Tt = trans(Q);

Figure A.5: Function used to calculate the mass and stiffness matrices of the undamaged beam element. It takes as an input the element's modulus of elasticity, length, moment of inertia, material density and cross section, and outputs the mass and stiffness matrices.

```matlab
function [GM,GK]=GlobalMatrices(DOFs,EN,M,K); %
GM = zeros(DOFs); GK=zeros(DOFs);
for k=1:EN
    GM(k,1:EN) = zeros(1,EN); GK(k,1:EN) = zeros(1,EN);
end

Figure A.6: Function used to calculate the global mass and stiffness matrices of the assembled beam. As an input it takes the degrees of freedom of the beam, the number of elements, the elemental mass and stiffness matrices, and gives as an output the global mass and stiffness matrices.

```matlab
function [GM,GK]=GlobalMatrices(DOFs,EN,M,K);
GM = zeros(DOFs); GK=zeros(DOFs);
for k=1:EN
    GM(k,1:EN) = zeros(1,EN); GK(k,1:EN) = zeros(1,EN);
end
```
APPENDIX A. MATLAB SCRIPTS AND FUNCTIONS USED IN THE THESIS

Figure A.7: Function used to calculate the bending modes of the undamaged cantilever beam. The function takes as an input the global mass and stiffness matrices of the beam, the boundary conditions and the number of the bending modes, and outputs the frequency values of the bending modes.

```matlab
function [UnF] = UnDamaged_Beam(GM, GF, BC, n)

    UnF = zeros(n, 1); G_K = GM; G_K = G_K;
    G_K(BC, :) = 1; G_M(:, BC) = 1; GRS = sparse(G_K);
    G_K[BC, :) = 1; G_K[1, BC] = 1; GRS = sparse(G_K);
    EV=eigs(GRS, GMS, n, 'SM'); F=sort(real(sqrt(EV)));
    for i = 1:n
        UnF(i) = F(i) / (2*pi);
    end
end
```

Figure A.8: Function used to calculate the stiffness matrix of the cracked beam element. It takes as an input the flexibility matrix of the cracked and uncracked beam element, the crack depth and the transformation matrix, and outputs the cracked elemental stiffness matrix.

```matlab
function [Kc] = CrackedStiffnessMatrix(C, s, Co, Tt, T)

    Fun1 = @(s) s.*sqrt(tan(s*pi/0.5)/(s*pi/0.5)).^2;
    Fun2 = @(s) [pi/2 - 0.561.*s + 0.85.*s.^2 + 0.016.*s.^3].^2;
    Fun1 = integral(Fun1, 0, s);
    Fun2 = integral(Fun2, 0, s);
    C11 = C(1) * (C(1) * (Fun1 + Fun2));
    C12 = C(2) * (Fun1 + Fun2);
    C21 = C12;
    C22 = C(3) * (Fun1);
    FM = Co(:, 1) + C11 * Co(:, 2) + C12;
    Co(3) + C21 * Co(:, 3) + C22);
    Kc = [Tt * FM] * T;
end
```

Figure A.9: Function used to calculate the global stiffness matrix of the damaged beam. The function takes as an input the global stiffness matrix of the undamaged beam, the stiffness matrix of the damaged beam element, and outputs the global stiffness matrix of the damaged beam.

```matlab
function [GK]=CrackedGlobalStiffnessMatrix(GK, TX, Flag)

    Fit=[2*Flag-1 2*Flag 2*Flag+1 2*Flag+2];
    GK(Fit, Fit) = GK(Fit, Fit) * TX;
end
```
Figure B.1: Ratios of normalized frequencies of the simulated bending modes versus normalized crack location and depth: a) $f_{12}$, b) $f_{13}$, c) $f_{14}$, d) $f_{15}$, e) $f_{16}$, f) $f_{17}$, g) $f_{18}$, h) $f_{23}$. 
Figure B.2: Ratios of normalized frequencies of the simulated bending modes versus normalized crack location and depth: a) $f_{24}$, b) $f_{25}$, c) $f_{26}$, d) $f_{27}$, e) $f_{28}$, f) $f_{34}$, g) $f_{35}$, h) $f_{36}$. 
Figure B.3: Ratios of normalized frequencies of the simulated bending modes versus normalized crack location and depth: a) \( f_{37} \), b) \( f_{38} \), c) \( f_{45} \), d) \( f_{46} \), e) \( f_{47} \), f) \( f_{48} \), g) \( f_{56} \), h) \( f_{57} \).
Figure B.4: Ratios of normalized frequencies of the simulated bending modes versus normalized crack location and depth: a) $f_{58}$, b) $f_{67}$, c) $f_{68}$, d) $f_{78}$. 
Appendix C

Published work in international scientific journals as part of the preparation of this thesis
Experimental detection by magnetoelastic sensors and computational analysis with finite elements, of the bending modes of a cantilever beam with minor damage

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ARTICLE INFO

Article history:
Received 11 January 2018
Received in revised form 25 March 2018
Accepted 20 April 2018
Available online 22 April 2018

Keywords:
Magnetoelastic sensors
Metglas
Cantilever beam
Natural frequencies
Damage detection
Finite element

ABSTRACT

This work introduces for first time the use of the magnetoelastic sensors as vibration probes for damage detection in mechanical structures such as cantilever beams. The purpose is to show some of the advantages of these materials as vibration detectors, as well as the accuracy of them in detecting the natural frequencies of mechanical structures. The sensor used is a ribbon which is composed of an amorphous metallic alloy known as “Metglas 2826MB3”. Various long aluminum alloy beams of the same dimensions but with a single transverse crack at different positions and depths were tested, fixed at one end by using a hydraulic press so as to have consistent boundary conditions. The beams were excited by a single short and intense mechanical contact pulse and then left free to vibrate. The vibrations were forcing the magnetoelastic sensors to change their magnetic state dynamically and thus produce a voltage signal at a close-by external coil. The Fourier analysis reveals seven dominant peaks which lay very close (most of the error values are between 0.5–1.5 %) to the first seven bending mode peaks predicted by the finite-element method (FEM) commercial software “ANSYS”. Thus the current work is a proof-of-principle that the magnetoelastic sensors can be used for damage detection of mechanical structures.

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1. Introduction

The term cantilever beam refers to any rigid structural limb projecting horizontally from a vertical support, especially one in which the projected dimension (length) is much greater than the other two dimensions (width and depth). In the field of engineering many structures can be treated as cantilever beams such as, projected parts of bridges and buildings, airplane wings, rotating blades of turbines, etc. These structures are subjected to various adverse conditions during their lifetime which cause damages and increase the risk of failure. When a structure suffers from damages its dynamic properties change and its natural frequencies shift, and by exploiting this information using suitable sensors, damage detection is possible. It is therefore important to develop techniques that can monitor the mechanical state of such structures and be able to determine their health at all times.

Over the years many researchers have tried to develop efficient methods that could detect changes in the dynamic behavior of a structure through the modal parameters such as mode shapes and natural frequencies. Pandey et al. [1] have investigated two different configurations, a cantilever and a simply supported beam, and have shown that changes in the curvature mode shapes (second derivative of mode shape) are localized in the region of damage, and increase with increasing the size of damage. Abdou and Hori [2] examined the rotation of the mode shapes of a steel plate model versus damage detection. They found that changes in the derivative (rotation or slope) of the mode shapes are more sensitive than the changes in the displacement mode shapes and that the rotation of a mode is localized in the region of the damage. Wahab and Roeck [3] investigated the change in modal curvatures towards damage in a prestressed concrete bridge. They introduced a damage indicator called “curvature damage factor” (CDF) in which the difference in curvature mode shape for all modes can be summarized in one number for each measured point. Yazdanpanah et al. [4] proposed a damage detection method by introducing a new mode shape data base indicator (MSDBI) which includes the mode shape, the mode shape slope and the mode shape curvature of an uncracked and cracked beam. Shi et al. [5] developed a sensitive statistical method to localize structural damage by direct use of incomplete mode shapes. Their damage detection strategy was to localize the damage sites first by using incomplete measured mode shapes, and then to detect the damage site and extent again by using
measured natural frequencies. Wang and McFadden [6] are among the first who tried to use wavelets transform for damage detection. They used orthogonal wavelet transform to detect abnormal transients generated by early damage from a gearbox casing vibration signal. Liew and Wang [7] examined the wavelet theory for crack identification of structures. The solved simply supported cracked beam using both the eigentheory and the wavelet method of analysis, and showed that crack identification via wavelet analysis is accomplished easily whereas it can hardly be detected by the traditional eigenvalue analysis. Chang and Chen [8] presented a technique for structure damage detection based on spatial wavelet analysis. They used the technique to analyze the mode shape of a Timoshenko beam and observed that distributions of the wavelet coefficients can identify the crack position and that the position can be identified even when there are measurement errors. Authors in refs. [9–12] also used wavelet transform method for structure crack detection, through the decomposition of the mode shapes.

An important parameter in the experimental study of the dynamic behavior of a structure is the detection method of the eigenvalues characteristics, such as mode shapes and natural frequencies. Srim et al. [13] applied a time-domain sorting algorithm to demonstrate the use of a scanning LDV (Laser Doppler Vibrometer) to simulate multiple discrete sensors distributed over the test structure. They illustrated the technique by measuring the second mode shape of a light-weight cantilever beam through the processing of the LDV output signal in the frequency domain. Okafor and Dutta [14] recorded and analyzed with wavelet transform, the first six mode shapes of a damaged and undamaged aluminum cantilever beam using scanning laser vibrometer. A finite-element model of the beams showed a close correlation to the corresponding experimental beam results. Seeley and Chattopadhyay [15] developed a multiobjective optimization technique which includes actuator locations, vibration reduction, power consumption, minimization of dissipated energy and maximization of the natural frequency as design objectives by using piezoelectric sensors. The technique was demonstrated through a cantilever beam problem and showed that performance control can be obtained with only a few optimally placed actuators. Wang and Wang [16] presented a theoretical analysis of the application of piezoelectric transducers to cantilever beam modal testing by considering four pairs of sensors and actuators including accelerometer-point force, accelerometer-PZT, PVDF-point force and PVDF-PZT. Results showed that any sensor-actuator pair can successfully determine natural frequencies and damping ratios. Abramovich and Pletner [17] proposed a piezo-laminated sandwich type structure for active control of sound radiated by harmonically excited thin walled structures. The numerical results are compared with experimental ones obtained during a test series on a cantilever sandwich beam equipped with piezoceramic sensors and actuators, and constructed according to the proposed concept. Sundaresan et al. [18] investigated the concept of a continuous sensor for detecting vibration and stress waves in bars. The sensing material used was PZT fibers patched inside an active fiber composite (AFC) which was bonded to the center of an aluminum panel. Strain, vibration, and wave propagation responses were simulated and results indicate that damage to the bar can be detected by recognizable changes in the sensor output as the wave propagates along the bar and passes over each sensor node.

In this paper an effort has been made to study the dynamic behavior of a cantilever beam using magnetostrictive-magnetoelastic sensors made of an amorphous magnetic alloy. Magnetostriction is the property of some ferromagnetic materials to deform continuously when they are subjected to an external applied magnetic field. The reverse effect, that is the change in the magnetic properties of a material caused by the application of mechanical stress, is known as the magnetoelastic effect. According to Hernado et al. [19] the parameter associated with the energy transfer between the elastic and magnetic subsystems is known as magnetoelastic coupling coefficient k (0 < k < 1). By far the best-known materials for mechanical stress sensing are metallic glasses. Authors in refs. [20,21] investigated the magnetoelastic coefficient factor of some metallic glasses and found very high values of it (0.70 < k < 0.97). They compared the metallic glasses with a conventional strain gage on static stress measurements and calculated sensitivity three to five orders of magnitude. Ausanio et al. [22] examined the influence of stress on the amplitude of the resonant mechanical waves inside a Fe_{62.5}Co_{6}Ni_{7.5}Zr_{6}Cu_{1}Nb_{15} ribbon for strain and/or 2 stress real-time monitoring in civil buildings. The results exhibited good reliability and stability as well as, better sensitivity [up to 200 times higher in proper conditions using resistive and vibrating wire strain gauges].

In the current work Metglas ribbons are used for the first time, at least to our knowledge, to detect the natural frequencies of a cantilever beam with a single traverse crack, and the results were found to have an excellent agreement by a corresponding FEM model made with ANSYS software. The advantage of this new sensing

APPENDIX C. PUBLISHED WORK IN INTERNATIONAL SCIENTIFIC JOURNALS AS PART OF THE PREPARATION OF THIS THESIS

Fig. 1. Experimental setup.
Table 1
CARVER hydraulic press specifications.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Clamping force</td>
<td>12 tons</td>
</tr>
<tr>
<td>2</td>
<td>Platens</td>
<td>10.2 cm round</td>
</tr>
<tr>
<td>3</td>
<td>Ram stroke</td>
<td>12.9 cm</td>
</tr>
<tr>
<td>4</td>
<td>Height</td>
<td>62 cm</td>
</tr>
<tr>
<td>5</td>
<td>Weight</td>
<td>52.2 kg</td>
</tr>
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</table>

Table 2
Waveform generator specifications.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Pulse repetition range</td>
<td>100 ns to 100 s</td>
</tr>
<tr>
<td>2</td>
<td>Pulse resolution</td>
<td>4 digits</td>
</tr>
<tr>
<td>3</td>
<td>Main output amplitude</td>
<td>5 mV to 20 V pk-pk</td>
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<tr>
<td>4</td>
<td>Main output resolution</td>
<td>3 digits</td>
</tr>
<tr>
<td>5</td>
<td>Trigger generator</td>
<td>3-digit resolution</td>
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</tbody>
</table>

2. Experiment

2.1. Experimental setup and procedure

Shown in Fig. 1 is the experimental setup which was used for the sensing of the natural frequencies of a cracked cantilever beam by the use of magnetoelastic ribbons. The setup includes, among other things, a CARVER hydraulic press which was used to clamp firmly the beam on one side so as to make it a cantilever beam, a detection coil connected to the sound card of a laptop, an arbitrary waveform generator, a linear power amplifier and a home-made mechanical stimulator to excite the beam. The specifications of the hydraulic press and the waveform generator are shown in Tables 1 and 2 respectively. The beam stimulator consists of a sharp metallic edge which is driven by a magnetic actuator to impinge on the top surface of the beam when the beam is at rest. The actuator is excited once by a 500 ms pulse sent by the generator and the amplifier. The stimulation sets the beam to bending oscillations and the two Metglas ribbons which are attached to the beam, as shown in Fig. 2b, follow the beam motion and their magnetic state changes accordingly. The detection coil detects this change by means of the electromagnetic induction (Faraday’s law) and an electric signal is produced which is recorded at the laptop via the sound card.

In the current work a total number of 31 beams were used (Fig. 2a) out of which, 1 was crack free and the other 30 beams had cracks at fixed positions and with fixed depths. The beam material is aluminum alloy 6063 and all beams were cut at same dimensions by CNC machinery. The mechanical and geometric properties of the beams are given in Table 4. As it was mentioned above each beam includes two equal-length magnetoelastic ribbons of material Metglas 2826MB3 (physical-geometrical properties are shown in Table 4) (Fig. 2b) attached on it by double-sided tape (physical properties are shown in Table 3), the excess of which is carefully removed using a sharp plastic blade to avoid scratching the beam. The gluing process is shown schematically in Fig. 3. The reason of using two ribbons instead of one is that the detection signal is much stronger than by using one ribbon, without the need for further amplification, and their position is symmetrical in order to avoid any anti-phase effects which could cause a reduction to the recorded signal. Some very good features of Metglas as a vibration sensor are:

- High stress sensitivity due to high magnetoelastic coupling.
- Easy treatment due to its flexible metallic nature.
- High corrosion resistance.
- No wire connections between the sensor and the recording devices are required.
- Low cost.

The clamping of the beam process is as follows: the carved plates, which are shown in Fig. 4a, are composed of aluminum alloys 7075 and used to fix the one end of the beam, as shown in Fig. 4b. The geometric details of the plates are shown in Fig. 4c with the length L equal to 50 mm, leaving out a free length of 350 mm of the beam to vibrate. The total internal opening of the carving inside the joined plates is 7 mm (3.5 mm on each side) so when a 8.18 mm thick beam is inserted between the two carved plates, a gap of 1.18 mm appears between them which allows the application of stress by the press. The clamping process takes place in 3 steps:

1. Centering the beam plates on the cylindrical platens of the press.
2. Closing the expansion valve of the press.
3. Clamping the beam at the desired pressure using the compression lever.

Once the beam is clamped on one of its ends, a detection ring coil is placed at the other end of it (Fig. 4d) using a stand. Table 5 shows the physical and geometrical properties of the ring coil. The position of the coil becomes optimum close to the ends (lengthwise) of the Metglas ribbons so as to trap the maximum of the magnetic flux.

Table 3
Double sided tape properties.

<table>
<thead>
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<th>Sr. No.</th>
<th>Property</th>
<th>Value</th>
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<td>Adhesive type</td>
<td>Modified acrylic adhesive</td>
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<tr>
<td>2</td>
<td>Tape thickness</td>
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<tr>
<td>3</td>
<td>Peel adhesion</td>
<td>300 N/mm</td>
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<tr>
<td>4</td>
<td>Normal tensile</td>
<td>620 kPa</td>
</tr>
<tr>
<td>5</td>
<td>Dynamic shear</td>
<td>620 kPa</td>
</tr>
<tr>
<td>6</td>
<td>Static shear</td>
<td>1000 g (22 °C)</td>
</tr>
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Table 4
Physical and geometrical properties of Metglas2826MB3 ribbons and aluminum alloy 6063 test-beam.

<table>
<thead>
<tr>
<th>Property no.</th>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>Length</td>
<td>100 mm</td>
</tr>
<tr>
<td>2</td>
<td>Width</td>
<td>6 mm</td>
</tr>
<tr>
<td>3</td>
<td>Thickness</td>
<td>29 μm</td>
</tr>
<tr>
<td>4</td>
<td>Density</td>
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<td>5</td>
<td>Modulus of elasticity</td>
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</tr>
<tr>
<td>6</td>
<td>Stoichiometry</td>
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</tr>
<tr>
<td>7</td>
<td>Saturation magnetostriction λₚ</td>
<td>12 ppm</td>
</tr>
<tr>
<td>8</td>
<td>Curie temperature</td>
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</tr>
<tr>
<td>9</td>
<td>Saturation induction</td>
<td>0.88 T</td>
</tr>
</tbody>
</table>

Table 5
Physical and geometrical properties of ring coil.

<table>
<thead>
<tr>
<th>Property no.</th>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Coil diameter</td>
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<tr>
<td>2</td>
<td>Wire diameter</td>
<td>0.16 mm</td>
</tr>
<tr>
<td>3</td>
<td>Tread number</td>
<td>1475</td>
</tr>
<tr>
<td>4</td>
<td>Wire material</td>
<td>Copper</td>
</tr>
<tr>
<td>5</td>
<td>Electrical resistance</td>
<td>232 Ω</td>
</tr>
<tr>
<td>6</td>
<td>Coefficient of inductance</td>
<td>0.215 H</td>
</tr>
</tbody>
</table>
which emerges from the beam ends (dipole distribution). The beam vibrations cause changes to the stress state of the ribbons and due to their magnetoelasticity, their magnetic flux changes. In turn, the flux changes induce voltage changes at the coil’s ends by means of Faraday’s law of induction. An AUX mono cable transmits the sensing signal from the coil to the laptop’s sound card through the microphone socket.

Data collection and processing was done through coding in the Matlab software. Fig. 5a shows the recording signal, when the beam is excited, narrowed to a specific range of time for a better visual-
3. Beam modeling

3.1. Configuration of cracks on the beam

In beam theory the mode frequencies of an undamaged beam are easily calculated by applying the differential equations of elasticity with the appropriate boundary conditions. However, when defects are introduced into the beam, no matter how small, the differential equations become intractable. In such a case only numerical solutions are possible through appropriate modelling and finite element analysis. In our case the modeling was done using an open-source parametric 3D modeler software named FreeCAD. Totally a number of 31 designs of different beams were made which correspond to 30 different beams with single regular faults (rectangular cracks) at prescribed positions, plus one undamaged beam. Fig. 7a shows our beam model with the crack locations pointed out. Cracks were chosen to be at locations 50 mm, 100 mm, 150 mm, 200 mm, 250 mm and 300 mm from the fixed end. Recalling that the free length of the beam is 350 mm, the crack positions correspond to the ratios of 1/7, 2/7, . . . , 6/7 of the beam’s length from the fixed end. The crack depth for each location is varied from 1 mm to 5 mm with a step of 1 mm (Fig. 7b). The two Metglas ribbons are attached to the opposite side of the beam where the cracks are located so as to keep them clear.

3.2. FEM analysis of the beam

In order to determine the natural frequencies of the modeled beams, the method of finite element analysis (FEM) was carried out using ANSYS 2016 Workbench software program. The design models from the FreeCAD program were imported into ANSYS Workbench in the form of ‘.step’ files and simulations were performed in pre-stress custom system which includes static and modal analyses. Three steps were followed in the Workbench environment to analyze the beam model and extract the natural frequencies of it. The first step involved the selection of the material, the contact parts, the meshing, the boundary conditions and the environment conditions (such as standard earth gravity direction). The second step included the solution of the problem with these settings. The last step was the extraction of the mode shapes. Fig. 8a shows the meshed model of the undamaged beam in top view and Fig. 8b–h show 3D graphs of the deformation of the same beam for the first seven bending modes along the easy axis (length-axis). Even though ANSYS predicts other kind of motions like torsion and compression, experimentally we did not detect any such modes because they are harder to excite with the kind of stimulus that was used in the present work.

4. Results and discussion

4.1. Results

The first seven bending modes (lengthwise) of a cracked and uncracked cantilever beam were studied experimentally using rib-
bons of magnetoelastic sensor Metglas2826MB3. While in ANSYS simulations fixing one end of the beam is easy to do, experimentally we had to use a hydraulic press in order to approach this boundary condition as much as possible. Fig. 9 shows the dependence of each natural frequency of the uncracked cantilever beam on the pressure load at the fixed end. The applied load range was 1–10 kN with a step of 0.5 kN, and for each step 5 different frequency measurements were made to derive the mean value of each bending mode. The frequencies show saturation at high values of pressure, as expected, and we will consider that the boundary condition of the fixed end is satisfied at these high pressures and take the saturation frequencies as the mode measured frequencies. For each

Fig. 7. (a) Designed beam in FreeCAD, (b) the different crack depths that were tested in mm, transverse to the beam’s width.

Fig. 8. (a) Top view of the mesh used to model the beam in FEM and different bending modes produced by ANSYS simulations: (b) 1st, (c) 2nd, (d) 3rd, (e) 4th, (f) 5th, (g) 6th, (h) 7th.
Fig. 9. Graphs of frequency versus pressure load for the first seven bending modes of undamaged beam: (a) 1st, (b) 2nd, (c) 3rd, (d) 4th, (e) 5th, (f) 6th, (g) 7th.

The second bending mode (Fig. 10b) along with the 3rd, 4th, 5th and 6th modes (Fig. 10c–f) also have most of the error values below 0.5%, showing thus a very good agreement between experiment and theory. For the 2nd bending mode the maximum error is at point \((L = 200 \text{ mm}, \alpha = 5 \text{ mm})\) with a value of 2.0%, while for the 3rd, 4th, 5th and 6th bending modes the maximum errors are at the points \((L = 250 \text{ mm}, \alpha = 5 \text{ mm}), (L = 300 \text{ mm}, \alpha = 2 \text{ mm}), (L = 300 \text{ mm}, \alpha = 2 \text{ mm}), (L = 300 \text{ mm}, \alpha = 3 \text{ mm})\) with values of 2%, 2%, 4% and 2% respectively. The 7th bending mode has the highest error value in relation to the other bending modes as most of the error points are between [0.5–1.5]%. The maximum error in this mode is at the point \((L = 300 \text{ mm}, \alpha = 3 \text{ mm})\) and its value is about 2%.

Some useful information can be extracted from the graphs of Fig. 11. As it was mentioned above, the normalized frequency of the 1st mode in Fig. 11a can give information about the distance of the crack from the fixed end. Also, it appears that for crack locations near to the fixed end the normalized frequency is changing rapidly versus crack depth. For example, for crack locations near the fixed end the normalized frequency is quite low (high in the vertical axis means low value). On the other hand, when the crack location is away from the fixed end the 1st harmonic resumes its “undamaged” value \(f_0\). The plot also shows strong dependence on the crack depth for almost all crack locations. The 2nd bending mode (Fig. 11b) shows marked changes mainly in the region between [100–300] mm from the fixed end and intense changes in the region [150–250] mm. The quite opposite behavior appears in the 6th bending mode (Fig. 11f), which appears to be more active at regions [50–100] mm and [250–300] mm and less to almost no-active in the region [150–250] mm. These two modes seem to be complementary to each other and can give information about the activity of the modes mostly in the middle region of the beam. The same complementary behavior appears between the 3rd mode (Fig. 11c) and the 5th mode (Fig. 11e). It is apparent that the 3rd mode shows intense maxima at the locations 90 mm and 250 mm from the fixed end, while the 5th mode shows deep minima there. The 4th bending mode presents activity in the region [250–300] mm and it keeps quite stable behavior on the rest regions. The last bending mode (Fig. 11g) shows a decrease of the normalized frequency at the region [50–200] mm and increase at [200–300] mm.

Fig. 12 is a graphic illustration of the beam deformation for each resonant mode except the 1st one as given by ANSYS for the 5 mm test-beam seven of these graphs are made, one for each mode, and the saturation error is extracted by fitting the data with exponential saturated functions (OriginPro 2015 software).

Table 6 shows the saturated frequencies (Mean), the calculated frequencies (Ansys) and the absolute value of the percent error between them for a beam with a crack located 50 mm from the fixed end. Values for crack depth \(\alpha = 0 \text{ mm}\) (undamaged), 1 mm, 2 mm, 3 mm, 4 mm and 5 mm are shown. It can be seen in this table that the error values are everywhere below 1% for both the damaged and the undamaged case, showing thus an excellent agreement between ANSYS and experiment, with only one exception for \(\alpha = 5 \text{ mm} at the 7th harmonic.

At this point we would like to mention that, the extra mass of the tape on the beam \((M_{tape} = 1.16 \pm 0.01)\text{gr}\), which could cause frequency shiftings, did not affect the error values significantly. In fact, in most of the cases the error is below 0.5%, indicating that the extra mass of the tape did not affect the natural frequencies of the beam. This can also be explained by comparing the mass and elasticity of the tape with that of the beam. Due to the very small mass of the tape, in relation to the beam, and its high elasticity, the beam almost defies the effect of the tape on it.

Shown in Figs. 10 and 11 is a summary of all our data in 3D graphs with each graph a, b, ..., g corresponding to a different bending mode 1, 2, ..., 7 respectively. The insets to the lower right corner of these two Figures show clearly the axes numbers and units of the plots, since it is hard to read them directly on the plots. The vertical axes in Fig. 10 correspond to percent errors (similar to the errors in Table 6) while the vertical axes in Fig. 11 correspond to the normalized measured frequencies \(f_{\text{meas}}/f_0\), where \(f_0\) is the frequency of the undamaged beam (note that the axis is reversed starting from 1, where \(f_{\text{meas}} = f_0\) and going up to 0.80). The other two axes of Figs. 10 and 11 are common and correspond to the location of the crack (from the beams fixed end) and its depth. Beginning with the 1st mode we note from (Fig. 10a) that, the error values are lower than 0.5% with the only exception being the case with crack location \(L = 50 \text{ mm}\) and crack depth \(\alpha = 3 \text{ mm}\), where the error value is between [0.5–1]%]. These results prove the accuracy of our experimental method. We note also from Fig. 11a that the \(f_{\text{meas}}/f_0\) ratio varies monotonically with respect to the distance of the crack from the fixed end (it is maximum close to this end and it drops as the crack location moves further from it).
Table 6
Comparison of the theoretical and experimental values of the natural bending frequencies for a crack location \(L = 50\) mm from the fixed end.

<table>
<thead>
<tr>
<th>(\alpha) (mm)</th>
<th>(f_1) (Hz)</th>
<th>(f_2)</th>
<th>(f_3)</th>
<th>(f_4)</th>
<th>(f_5)</th>
<th>(f_6)</th>
<th>(f_7)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>54.0</td>
<td>338.0</td>
<td>943.5</td>
<td>1839.4</td>
<td>3020.3</td>
<td>4475.1</td>
<td>6187.6</td>
</tr>
<tr>
<td>Ansys</td>
<td>54.1</td>
<td>337.7</td>
<td>943.9</td>
<td>1840.9</td>
<td>3026.2</td>
<td>4493.5</td>
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<tr>
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<td>0.09</td>
<td>0.05</td>
<td>0.08</td>
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</tr>
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<td>Error(%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>1</td>
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<td>943.8</td>
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<td>3014.3</td>
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<td>0.02</td>
<td>0.01</td>
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<tr>
<td>Error(%)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>2</td>
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<td>1829.3</td>
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<td>0.06</td>
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<td>0.18</td>
<td>0.21</td>
<td>0.54</td>
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<td>Error(%)</td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td>3</td>
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<td>333.4</td>
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Fig. 10. Node frequency error values between experiment and theory versus crack depth and crack location for different bending modes: (a) 1st, (b) 2nd, (c) 3rd, (d) 4th, (e) 5th, (f) 6th, (g) 7th. The inset to the right indicates the axes numbers and units (common to all plots) and the error color scale. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

crack (little white bar) at different beam locations (indicated by numbers to the right of graph a). Please note that the top illustration in each set corresponds to the undamaged case and the fixed end of the bar is on the right side (where the distance of the crack is measured from). Basically, the blue color corresponds to no deformation which indicates a standing wave node, while the red color corresponds to exactly the opposite, a large deformation which is a local maximum of the standing wave. We see that the location of the 5 mm crack has a major effect on the shape of the harmonics. We probably expect that as the total bar thickness is close to 8 mm and thus the 5 mm crack is quite deep one. In particular, we note that when the crack is in the vicinity of a node (blue color in the undamaged case), no much is happening. On the contrary, when the crack is midway between two nodes it creates there a local maximum (red colors around the crack) which means that the harmonic has been strongly altered. Indeed this is in accordance with the data shown in Fig. 11. For example, lets examine the second mode deformation illustration shown in Fig. 12a. The undamaged illustration exhibits two nodes, one at the fixed end at the right and one around 275 mm. When the crack is located at 150 mm which is about half way between 0 and 275 mm, some red color appears around the crack where it was yellow before at the undamaged case. Taking a closer look at Fig. 11b, which corresponds to the frequency changes of the 2nd harmonic, we can confirm that indeed the highest change happens for crack locations between 150 and 200 mm. Similarly in Fig. 12b, which shows the deformation map of the 3rd mode, the crack at 100 mm which is midway between two nodes (blue colors) acts as a local maximum with some red color around it. Of course there are also some exceptions to this rule. If we take a look at the 200 mm case of the same graph we notice an extra red region to the right, even though the crack is relatively close to a node. However, the exceptions are rather low in numbers.

APPENDIX C. PUBLISHED WORK IN INTERNATIONAL SCIENTIFIC JOURNALS AS PART OF THE PREPARATION OF THIS THESIS
and the above rule is quite valid. For example in Fig. 12f, which represents the 7th bending mode shape, there are quite a few nodes. When the crack location is at 50 mm, it is close to a node but when it is at 200 mm, it is situated almost symmetrical between two successive nodes and thus we expect a major change in the resonance frequency. Indeed, in Fig. 11g we see that at crack location 200 mm the normalized frequency appears to have it most intense change. The fact that the frequency changes are minimal, when the crack is close to a node, can be explained physically: a standing wave does not carry energy and thus at the nodes the energy is very low which means that the crack cannot absorb energy from the beam. On the contrary, when the crack is midway between two nodes, there is plenty of energy available which is absorbed by the crack in order to behave like an almost free-end. Another conclusion that can be drawn from Fig. 11, which can probably explained by Fig. 12, is that on the average the frequency changes are less intense as we move to higher harmonics in Fig. 11. As expected, there are more nodes in higher harmonics in Fig. 12 which means that the beam energy is divided in more portions and thus the beam locally can absorb less energy and have a smaller effect.

5. Conclusions

A new experimental method for detecting the natural frequencies of an uncracked and cracked cantilever beam is proposed. This method uses as sensors ribbons which are composed of metallic glass alloy 2826MB3. The method detects the beam’s natural frequencies by means of Faraday’s law of induction and offers several advantages such as (a) no wire connections between the sensor and the detector, (b) flexible sensors so as to not alter the
beam's stiffness, (c) low cost sensor material and (d) high sensitivity. A total of 31 beam specimens were tested by our new method and the measured natural frequencies had small errors 0.5–1.5% compared to the analytical values predicted by ANSYS software. Special care was taken in order to have consistent boundary conditions by the use of a hydraulic press and by taking the saturation resonant values through data fitting with Origin software. 3D graphs of the normalized frequency of each bending mode were constructed versus crack depth and location, which show a great variety and thus make the current sensing technique an ideal candidate for damage detection. Whenever a crack location is near to a node the changes on normalized frequencies are minimal, while when a crack happens to be located midway between two nodes the corresponding resonance frequencies show a major change.

Acknowledgements

Special thanks to the University of Patras machine shop operator Gerasimos Diamantis for providing us technical support and useful discussion about the experimental setup, and Ioannis Kapsalis for his contribution on the initial setup of the experiment.

References


Biographies

Prof. Dimitris Kouzoudis received his Bachelor Degree in Physics from the University of Ioannina, Greece in 1990, his M.S. in Physics and Material Science and his Ph.D. in Physics from Iowa State University in 1994 and 1999 correspondingly. Between 2000 and 2001 he was employed as a research scientist at the Dept. of Electrical Engineering, University of Kentucky, U.S.A. and between 2001 and 2002 as a consultant at the headquarters of the "Kindred Healthcare" company. At 2003 he received a Lecturer position at the Material Science Department of University of Patras, Greece and at 2005 he received a tenure-track Assistant Professor position at the Department of Engineering Sciences at the same University. Between 2012-2013, he took a sabbatical leave at the HCT Engineering School in the United Arab Emirates. At 2014 he was received a tenure Assistant Professor appointment at the Chemical Engineering Department of University of Patras, Greece and at 2017 he became an Associate Professor at the same department. He was the editor-in-chief of the scientific peer-review journal "Sensor Letters" between 2003 and 2012 with impact factor 0.7-1.0, and he is the owner of two patents No. 6,688,162 and 6,393,921 in the United States Patent and Trademark Office related to magnetoelastic sensors. His publication list includes 38 papers with 1093 citations and an h-index of 16 (Scopus) and focus on the application of different sensor platforms towards monitoring environmental/chem/bio parameters, such as gas concentration, small mass loads, pressure, flow velocity, humidity, and precipitation of biological salts in aqueous solutions, blood coagulation time, and glucose concentration, the use of zeolite films as active sensor layers for the detection of volatile organic compounds.

A pattern matching identification method of cracks on cantilever beams through their bending modes measured by magnetoelastic sensors

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ARTICLE INFO

Keywords:
Crack detection
Magnetoelastic sensors
Bending modes
Metglas
Damage detection
Modal analysis

ABSTRACT

This work introduces a simple method on identification of single transverse cracks on cantilever beams, having as an input the first 8 bending modes of the beam. The beam modeling part of the method was developed based on a finite element method and fracture mechanics theory, and implemented using Matlab programming. Bending mode normalized frequencies were extracted from the model, versus crack location (CL) and crack depth (CD), and all possible normalized frequency ratios (NFRs) were calculated in order to be stored in a database. This database information was used to find the optimum CL and CD values from input data, through a pattern matching process. Two different approaches were presented to validate the method, one experimental and one numerical. In the case of the experimental approach a number of 19 beam specimens of aluminum alloy 6063 were used, with various fixed crack locations and depths, and the extracted CL and CD errors were 1.7% and 11.4%, respectively. The measurement of the 8 bending modes was accomplished non-invasively by the use of magnetoelastic vibration sensors, which were composed of 29 μm thin ribbons of magnetoelastic material Metglas 2826MB3. On the other hand, the numerical approach was performed using ANSYS 2016 Workbench software on beam designs with the same physical and geometrical characteristics as the beam specimens used in the experiment. The extracted CL and CD errors are found to be 0.7% and 8.5%, respectively. The above errors show that the proposed method is extremely capable of predicting the crack location and quite capable of predicting the crack depth. This work sets the basic prospects for the design of a new Structural Health Monitoring (SHM) technique combining the efficiency of the magnetoelastic sensors together with the proposed method.

1. Introduction

The evolution of material engineering in recent decades has greatly improved the way in which researchers deal with their various issues. In the field of sensing and health monitoring of mechanical structures, more and more research is being conducted towards the exploitation of specific material properties for the purpose of increasing the lifetime of these structures. Mechanical structures are subject to various adverse conditions during their lifetime which causes damages that can affect their performance, as well as their function. Consequently, timely detection of these damages can prevent the failure of the structure and ensure proper functioning, both for safety and financial terms.

Several studies have been presented in recent years on the subject of health monitoring of engineering structures using a variety of materials as sensors and sensing devices. The authors in [1–4] developed several synthetic detection devices, incorporating optical fibers, to monitor the structural health of concrete and non-concrete structures, by measuring the static and dynamic loads in the structures. Another category of materials that have exhibit high sensitivity to the dynamic response of mechanical structures, are the piezoelectric materials. In Refs. [5–7] the authors investigated the concept of a continuous, grid-type sensor array composed of lead-zirconium-titanium oxide (PZT) sensors, for detecting vibrations and stress waves in mechanical structures for cracks detection. In our recent work [8] we studied the possibility of using magnetoelastic sensors as vibration probes in order to detect the bending modes of a mechanical structure, such as cantilever beams, for damage detection. As we have shown, there are certain advantages of this method due to the magnetic nature of the detection process and the high sensitivity of the sensor.

Many studies have been carried out on magnetoelastic materials to exploit the magneto-elastic property for sensing purposes [9]. Magnetoelasticity is the property of some ferromagnetic materials to change their magnetic state (magnetization) by the application of mechanical stresses. The reverse effect, which is the continuous deformation of magnetoelastic material under the application of a magnetic field, is known as magnetostriction. An important parameter in these materials is
the magnetoelastic coupling coefficient $k$ which is equal to the ratio of conversion of the magnetic to the elastic energy ($0 < k < 1$). Authors in Refs. [10,11] have measured this coefficient of some ferromagnetic metallic glasses ribbons and found values between 0.70 < $k$ < 0.97, with the latter one being the highest among the known magnetoelastic materials. Hison et al. [12] examined the influence of stress on the amplitude of the resonant mechanical waves inside a Fe$_{82.5}$Co$_{6}$Ni$_{5}$Zr$_{2}$Cu$_{8}$Nb$_{3}$B$_{2}$ magnetoelastic ribbon for strain-stress real-time monitoring in civil buildings, obtaining sensitivity results of 0.1 mV/μm.

The sensor output itself, in a damage detection experiment of a mechanical structure, is not enough to detect defects without the use of a suitable theoretical model. Over the years many researchers have tried to develop efficient methods that could detect damages on mechanical structures, focusing mostly on the changes of the dynamic characteristics of the structure such as its mode shapes and natural frequencies. Starting with mode shapes, Ratcliffe [13] developed a technique for identifying the location of structural damage in a beam, by applying a finite difference approximation of Laplace’s differential operator to the mode shape of the fundamental natural frequency. Homaei et al. [14] introduced a criterion named Multiple Damage Localization Index Based on Mode Shapes (MDLUMS criterion) to localize the damage by combining the mode shapes of both damaged and undamaged structures. The authors in [15–18] proposed various methods on exploiting the derivatives of mode shapes (slope and curvature) for damage detection, due to their higher sensitivity on changes compared to displacement (mode shape). However, Cao et al. [19] indicated that the disadvantage in deriving the mode shapes is the fact that they amplify the noise, and proposed a technique which combined a wavelet transform (WT) and a Teager Energy Operator (TEO), and was able to identify multiple cracks in cantilever beams, especially in noisy conditions. Jaiswal et al. [20] presented a numerical study for damage detection in beam structures with mode shape curvatures and spatial wavelet transforms. As they observed, the decomposition of the spatial signal into wavelet details can identify the damage position in beam-like structures by showing relatively larger peaks at the position of the damage. Kim et al. [21] investigated the use of two different methods, one based on mode shapes and one on frequencies, in order to compare them in damage detection process in beam-like structures. As they concluded, both methods can localize the damage and accurately estimate the sizes of the cracks.

Although the above methods, based on the mode shapes, can accurately predict the location and size of the damage, the utility of them in practical applications is limited due to the requirement of large number of sensors in every point of the structure. This limitation is avoided in the case of natural frequencies. The process of measuring the natural frequencies of a mechanical structure is simpler and easy, and can be accomplished using only one sensor. For example, Lifshitz et al. [22] were the first who developed an experimental technique on damage detection using natural frequencies. They studied the shifting of natural frequencies of composite materials, through the changes in their Young modulus. The authors in [23–26] presented damage detection methods based on natural frequencies on various mechanical structures that are quite common in structural engineering. Due to the dependence of natural frequencies on the position and depth of the crack, some researchers investigated the method of using frequency contours [27–29]. Rhiem and Lien [30] proposed a robust methodology on multi-crack detection using natural frequencies, by presenting a procedure which solves a constrained non-linear optimization problem of a rotational spring modeled crack. Recently, a new method has been proposed by Dahak et al. [31] to detect the location of cracks by using natural frequencies on cantilever beam structures. The authors developed an analytical method that discretize the beam into a number of zones, and they derived a unique classification of the normalized bending modes for each zone.

The present work is a continuation of our previous study on the use of magnetoelastic sensors as sensing devices for detecting the mechanical health of a structure. In [8] we introduced for the first time the use of the magnetoelastic sensors as vibration probes, and examined the proof of principle of the technique by comparing the experimental data with analytical data predicted by finite element analysis (FEM) software ANSYS. Now we make a step further and study the possibility of evaluating the location and the size of the damages on cantilever beams through the their bending modes measured by magnetoelastic sensors. The innovation of this work lies in the exploitation of a specific category of magnetic materials which can be used as a vibration sensor to measure high natural frequency orders, and the way in which this information can be used in order to identify with better precision the health state of a mechanical structure.

2. Beam modeling

The modeling of the dynamic behavior of the cantilever beam through its bending modes with the presence of damages on it was accomplished by using a finite element method and fracture mechanics theory [32,33]. The task was carried out in three stages which were later combined in order to be programmed using Matlab software. The first stage involves the analytical extraction of the stiffness and mass matrices of the undamaged beam element using the Euler–Bernoulli beam finite element method. The second stage involves the extraction of the stiffness matrix of the damaged beam element using fracture mechanics theory. The final stage involves the assembling of all beam elements and the extraction of the global stiffness and mass matrices, considering the boundary conditions, and solving the eigenvalue problem for the bending modes of the cantilever beam.

2.1. The undamaged beam element

Fig. 1 illustrates a schematic diagram of a beam element under bending. The uniform element of length $L_e$ consists of two nodes at its two ends (red dots).

Each node has two degrees of freedom (DOFs), one for translational ($u_i(0), u_i(0)$) and one for rotational displacements ($\theta_i(0), \theta_i(0)$). Following typical approaches of FEM analysis, the equation of motion of the element in Fig. 1 can be derived in the form:

$$[m_i][\ddot{u}(t)] + [k_i][u(t)] = [f(t)]$$

where $u(t)$ is the column matrix with the four displacements $u_i(0), \theta_i(0)$ is the column matrix of node’s force vectors and $[k_i], [m_i]$ are the elemental stiffness and mass matrices, respectively. Eq. (1) constitutes a different form of the Lagrange’s equation of motion, which integrates the analytical expressions of the kinetic and the dynamic energies of each element (a detailed analysis on the derivation of the Lagrange’s equation of motion and extraction of the mass and stiffness matrices can be found in chapter 8 of book [32]). The corresponding elemental mass and stiffness matrices can be expressed as follows:

$$m_i = \frac{\rho A L_e}{420}$$

$$\begin{bmatrix}
156 & 22L_e & 54 & -13L_e \\
22L_e & 4L_e^2 & 13L_e & -3L_e^2 \\
54 & 13L_e & 156 & -22L_e \\
-13L_e & -3L_e^2 & -22L_e & 4L_e^2
\end{bmatrix},$$

$$k_i = \frac{EI}{L_e^3}$$

$$\begin{bmatrix}
12 & 6L_e & -12 & 6L_e \\
6L_e & 4L_e^2 & -6L_e & 2L_e^2 \\
-12 & -6L_e & 12 & -6L_e \\
6L_e & 2L_e^2 & -6L_e & 4L_e^2
\end{bmatrix}$$

$$\text{where } \rho = \text{material density, } A = \text{element’s cross section area, } L_e = \text{length of the element, } E = \text{Young’s modulus and } I = \text{moment of inertia}.$$

1 For interpretation of color in Figs. 1 and 7, the reader is referred to the web version of this article.
The matrices in Eq. (2) refer to the intact state of the beam element and consist the basic information for studying the vibration behavior of the beam without damage.

### 2.2. The damaged beam element

The presence of a damage in the form of a crack in a structural element affects significantly the vibration behavior by reducing its local stiffness. Although the crack alters both the elemental mass and stiffness matrices, the changes on the mass matrix can be neglected due to the insignificant mass removal. Fig. 2 shows the schematic diagram of the beam element with only bending (M1, M2) and shearing (T1, T2) forces acting on it, under the presence of a transverse crack with depth s, halfway along the element’s length.

The static equilibrium condition of the element in Fig. 2 is expressed as follows:

\[
\begin{align*}
\sum_{i=1}^{2} T_i &= 0 \Leftrightarrow T_1 + T_2 = 0 \Leftrightarrow T_1 = -T_2 \\
\sum_{i=1}^{2} M_i &= 0 \Leftrightarrow M_1 + M_2 + T_1 L_s = 0 \Leftrightarrow M_1 = -T_1 L_s - M_2
\end{align*}
\]

where \( T_i \) and \( M_i \) are the crack-tip stress intensity factors for each mode with the indexes indicating the respective mode, and \( L_s \) is the crack cross section. The presence of the crack on the element affects significantly its stiffness and accordingly its elemental stiffness matrix, and the equation of stiffness matrix for the element of Fig. 2 can be defined as:

\[
[k_s] = [A]^T [C]^{-1} [A] \tag{7}
\]

where \([A]\) is the transformation matrix of the equilibrium state of the cracked element and \([C]\) is the flexibility matrix. The form of this last equation is derived using the energy method and it can be applied to both the damaged and the undamaged state of the beam element, which makes it easy to use for further analysis. Using Eqs. (5)–(7) the stiffness matrix is obtained as:

\[
\begin{bmatrix}
1 & C_{11} C_{22} - C_{12} C_{21} \\
C_{12} & L_s C_{22} - C_{12} & L_s C_{22} - C_{12} & C_{12} \\
L_s C_{22} - C_{12} & L_s C_{22} - C_{12} & L_s C_{22} - C_{12} & C_{12} \\
C_{21} & L_s C_{21} - C_{12} & C_{22} & -C_{12} \\
C_{21} & L_s C_{21} - C_{12} & C_{22} & -C_{12} \\
C_{21} & L_s C_{21} - C_{12} & C_{22} & -C_{12} \\
\end{bmatrix}
\tag{8}
\]

The strain energy for a crack free beam element subjected to total bending moment \( M_0 = TL + M \) is given by:

\[
U_0 = \frac{1}{2} \int_0^{L_s} \frac{M_0^2}{EI} dL = \frac{1}{2} \int_0^{L_s} \left( \frac{TL + M}{EI} \right)^2 dL = \frac{L_s}{2EI} \left( \frac{(TL_s)^2}{3} + M^2 + 3ML_s \right) \tag{9}
\]

Using Eqs. (6) and (9) to calculate the coefficients of the flexibility matrix and introducing them into Eq. (8), we get the stiffness matrix of Eq. (2) of the undamaged beam element.

When cracks are introduced, the strain energy in Eq. (9) must be written with an additional term that is related with the relative modes of crack surface displacements. There are three basic modes of fracture damages, as shown in Fig. 3. Mode I (Fig. 3a) is the “opening” or “tensile” mode where the crack surfaces are displaced normal to themselves. Mode II (Fig. 3b) is the “sliding” or “in-plane shear” mode where the crack surfaces slide over one another in a direction perpendicular to the leading edge of the crack. Finally, Mode III (Fig. 3c) is the “tearing and antiplane shear” mode where the crack surfaces move relative to one another and parallel to the leading edge of the crack. According to Tada et al. [33], the additional term (due to the crack) that comes into the expression of strain energy is given by:

\[
U_s = \int_s \frac{1}{E} \left[ K_{ii}^2 + K_{ii}^2 + \frac{E' K_{iii}^2}{2\mu} \right] dS \tag{10}
\]

where \( E' = E(1 - \nu^2) \) is the plane strain Young’s modulus, \( \mu = E/(2(1 + \nu)) \) is the shear modulus, \( E \) and \( \nu \) the Young’s modulus and the Poisson’s ratio, respectively, \( K_i \) and \( K_{iii} \) are the crack-tip stress intensity factors for each mode with the indexes indicating the respective mode, and \( S \) is the crack cross section. The \( K_i \) and \( K_{iii} \) factors represent the strength of the stress field surrounding the crack tip and physically may be regarded as the intensity of transmitted load though the crack-tip region. In our case, the cracked beam element of

---

**Fig. 1.** Schematic diagram of the undamaged beam element under bending.

**Fig. 2.** Schematic diagram of the damaged beam element under bending and shearing forces.

**Fig. 3.** Relative modes of crack surface displacements: (a) Mode I, (b) Mode II, (c) Mode III.
Fig. 2 has width w, height h and crack depth s. When only shear and bending forces are applied, as is the case of a beam under bending, these factors can be expressed as:

\[ K_s = K_{IT} + K_{SM}, \quad K_b = K_{IT} + K_{III}, \quad K_{III} = 0 \]  
(11)

The above equations express the fact that there are two contributing components to the K factors, one due to shear forces (T) and one due to bending moments (M). Because the K factors depend linearly on the stresses, the superposition principle applies and these two components are added together. In particular, in Mode I both components are present, in Mode II only the shear component is present and in Mode III none of them contributes to the crack formation. By applying Eqs. (11) to Eq. (10) and taking into account that the crack cross-section area is equal to \( S = w \times s \), we get:

\[ U_1 = \int_0^w \frac{W}{E} \left[ (K_{IT} + K_{SM})^2 + K_{III} \right] ds \]  
(12)

The factors \( K_{IT}, K_{SM} \) and \( K_{III} \) in Eq. (12) are functions of both the external forces and the size of the crack, and according to Tada et al. [33] they are given as:

\[ K_{IT} = \sigma_{IT} \sqrt{\pi} F_1(\lambda), \quad K_{SM} = \sigma_{SM} \sqrt{\pi} F_1(\lambda), \quad K_{III} = \sigma_{III} \sqrt{\pi} F_1(\lambda) \]  
(13)

where

\[ F_1(\lambda) = \frac{2}{\sqrt{\pi}} \tan \frac{\pi}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi}{2})^4}{\cos \frac{\pi}{2}} \right) \]  
(14)

\[ F_2(\lambda) = \frac{1.122 - 0.561 + 0.085(2^2 + 0.18^3)}{\sqrt{1 - \lambda}} \]  
(15)

\[ \sigma_{IT} = 3T_L \frac{h}{wh}, \quad \sigma_{SM} = 6M \frac{w}{wh}, \quad \sigma_{III} = \frac{T}{wh} \]  
(16)

The \( \sigma_{IT}, \sigma_{SM} \) are the bending stresses applied by shear and bending forces, respectively, in order to open the crack, the \( \sigma_{III} \) is the shear stress applied by shear forces in order to slide the crack, the \( F_1(\lambda), F_2(\lambda) \) are the correction functions of the intensity factors of each respective mode and \( \lambda = s/h \) is the normalized crack depth. By substituting the above two equations into Eq. (13), the explicit forms of the K factors are:

\[ K_{IT} = \left[ \frac{3T_L h}{wh} \right] \sqrt{\pi} \lambda \theta \left( \frac{2}{\sqrt{\pi}} \tan \frac{\pi}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi}{2})^4}{\cos \frac{\pi}{2}} \right) \right) \]  
(17)

\[ K_{SM} = \left[ \frac{6M w}{wh} \right] \sqrt{\pi} \lambda \theta \left( \frac{2}{\sqrt{\pi}} \tan \frac{\pi}{2} \left( \frac{0.923 + 0.199(1 - \sin \frac{\pi}{2})^4}{\cos \frac{\pi}{2}} \right) \right) \]  
(18)

\[ K_{III} = \left[ \frac{T}{wh} \right] \sqrt{\pi} \lambda \theta \left( \frac{1.122 - 0.561 + 0.085(2^2 + 0.18^3)}{\sqrt{1 - \lambda}} \right) \]  
(19)

From Eqs. (9), (12) and (16) the total strain energy \( U = U_1 + U_2 \) of the cracked beam element is given as:

\[ U = \frac{L_2}{2EI} \left[ \frac{T(L_2)^2}{3} + M^2 + TML_2 \right] + \frac{\pi(1 - \nu^2)}{Ew} \int_0^s \left[ \int_0^r \frac{\partial F_1(\lambda)}{\partial \lambda} d\lambda \right] \]  
(20)

Using Eq. (7) and the above equation, the flexibility coefficients can be calculated. The result is:

\[ C_{11} = \frac{L_2}{2EI} \left[ \frac{T(L_2)^2}{3} + M^2 + \frac{\pi(1 - \nu^2)}{Ew} \int_0^s \frac{\partial F_1(\lambda)}{\partial \lambda} d\lambda \right] \]  
(21)

\[ C_{12} = C_{21} = \frac{L_2}{2EI} \left[ \frac{T(L_2)^2}{3} + \frac{\pi(1 - \nu^2)}{Ew} \int_0^s \frac{\partial F_2(\lambda)}{\partial \lambda} d\lambda \right] \]  
(22)

\[ C_{22} = \frac{L_2}{2EI} \left[ \frac{T(L_2)^2}{3} + \frac{\pi(1 - \nu^2)}{Ew} \int_0^s \frac{\partial F_1(\lambda)}{\partial \lambda} d\lambda \right] \]  
(23)

The next step is to introduce these coefficients into Eq. (8) and calculate the elemental stiffness matrix. As it has been mentioned above, the changes on the mass matrix of the beam element with the presence of the crack can be neglected, leaving thus the matrix \( [m] \) from Eq. (2) unchanged. The final step of the beam modeling involves the assembling of the whole beam, considering the boundary conditions, and the extraction of the bending modes through the solution of the eigenvalue problem (in chapter 8 of this book [32] there is an example of the assembling procedure of a discretized cantilever beam, as well as an example of the solution of the eigenvalue problem).

3. Beam model simulation results and discussion

3.1. Matlab simulation

The simulation of the model presented in the previous sections was carried out using the Matlab software program. A cantilever beam with \( NE = 2000 \) elements was considered, the geometric and physical characteristics of which are shown in Table 1. These characteristics were selected based on the experimental specimens that were used for the evaluation of this method. As it is known, the number of elements used to simulate the beam is related to the accuracy of the method because it discriminates the beam in smaller and smaller pieces, and thus the detection of the crack can be done with less error. Also, a sparse matrix method was used to reduce significantly the elapsed time of the algorithm due to the high number of elements.

### Table 1

<table>
<thead>
<tr>
<th>Property No.</th>
<th>Property</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>Length</td>
<td>350 mm</td>
</tr>
<tr>
<td>2</td>
<td>Width</td>
<td>30.25 mm</td>
</tr>
<tr>
<td>3</td>
<td>Height</td>
<td>8.18 mm</td>
</tr>
<tr>
<td>4</td>
<td>Material density</td>
<td>2.69 g/cm³</td>
</tr>
<tr>
<td>5</td>
<td>Cross section</td>
<td>247.45 mm²</td>
</tr>
<tr>
<td>6</td>
<td>Moment of inertia (easy axis)</td>
<td>1379.8 mm⁴</td>
</tr>
<tr>
<td>7</td>
<td>Modulus of elasticity</td>
<td>68.3 GPa</td>
</tr>
<tr>
<td>8</td>
<td>Poisson’s ratio</td>
<td>0.33</td>
</tr>
</tbody>
</table>

3.2. Normalized frequency variation versus crack depth and location

Fig. 4 shows the variation of the first 8 bending modes of the simulated cantilever beam versus crack location (CL) and crack depth (CD). The vertical axis corresponds to the normalized frequency (NF) values of the beam, which is equal to the ratio \( \lambda_{damaged}/\lambda_{undamaged} \), where \( \lambda_{damaged} \) and \( \lambda_{undamaged} \) are the bending modes of the cracked and cracked-free beam, respectively. The horizontal axis corresponds to the normalized values of the crack location (NCL) with values between 0 and 1 (with 0 corresponding to the fixed end and 1 to the free end), and \( step = 1/NE = 0.0005 \), where \( NE \) is the number of elements.

There are a total number of 14 curves in each graph of Fig. 4, each corresponding to different CD ratios (the caption in Fig. 4a shows the variation of the CDs in percentages). A common feature of all the curves in Fig. 4 is that none of them contains values greater than 1. This indicates that, according to the model, the presence of a transverse crack on the beam always tends to reduce the frequency values of the bending modes, since the ratio \( \lambda_{damaged}/\lambda_{undamaged} \) is always below 1, regardless of the CL and CD. Another common feature is that there is at least one point where all curves converge to unity. The magnification of the converging points shown in Fig. 4b reveal that the converge to unity is only approximate and not exact. The same is true for all graphs. For the first bending mode (Fig. 4a) the converging point is the free end of the beam, while for the rest bending modes (Fig. 4b–h) there are more than one converging points along the beam, including the free end. Since at the converging points \( \lambda_{damaged}/\lambda_{undamaged} \approx 1 \), the corresponding bending...
mode frequency is almost the same as the one in the undamaged beam. This is happening because the converging points of each mode are identical to the beam nodes, where no displacements take place. In our recent work [8] we have observed experimentally such a behavior. Additionally it was observed that when the CL was symmetrical in between two nodes (highly deformation point), the changes in the corresponding frequency were maximal. Although these rules are valid for all the nodes along the beam, there are two points on the cantilever
beam which are exceptions and the opposite rule holds, and these points are the ends of the beam.

3.3. Ratio of normalized frequencies of different modes versus crack depth and location

The graphs in Fig. 4 show the behavior of each bending mode individually with the presence of the crack. Even though in principle these graphs could have been used to extract the crack location and depth, in practice this approach leads to large errors due to the small sample size (8 NFs values), and since the proposed methodology (described in Section 4) is based on the evaluation of the variance between the predicted and experimental values, a larger number of sample values can increase the precision of the method by reducing the variance of the sample mean (standard error). Thus, this number can be greatly increased to 28 by taking all possible combinations of ratios of normalized frequencies between the different NFs (from now the ratios \( f_m / f_n \), where \( f_n \) and \( f_m \) are the NF of the bending modes \( n \) and \( m \) respectively, will be referred as NFR for simplicity). For example Fig. 5 shows some of the NFR graphs which have the same horizontal axis as the graphs in Fig. 4. Due to the high number of graphical representations, Fig. 5 shows only the NFR graphs of the combinations \( f_1 / f_2 \), \( f_1 / f_3 \), \( f_2 / f_3 \), and \( f_2 / f_4 \).

Taking now the graph of Fig. 5a, it can be said that it is divided into two non-symmetric regions. For these regions the ratio \( f_1 / f_2 \) varies depending on whether the crack appears on the left or the right side of cross-one point. On the left region the ratio \( f_1 / f_2 \) is always below the unity \( (f_1 / f_2 < 1) \), while on the right region is always above the unity \( (f_1 / f_2 > 1) \), creating thus a condition which determines the location of the crack. The same conclusion is obtained on the curves of Fig. 5b–d, but with four regions where the ratio \( f_2 / f_n \) is switching alternatively \( (f_2 / f_n > 1 \) and \( f_2 / f_n < 1 \)). Similarly in Fig. 6 (NFR \( f_7 / f_8 \)) there are 14 such regions. From a physical point of view, the cross-one points of the NFR curves sets the boundaries between different regions on the beam and determines the relative behavior among the NFs with the presence of the crack.

4. Crack localization and quantification methodology

According to the simulation results of the beam model described above, for each CL and CD on the beam there is a set of 28 NFR values, which can be stored as a database in the form of patterns. These patterns can be described as holes in the puzzle whose shape is determined by the set of NFR values, and for which there is ideally a single match. Thus, the idea of this methodology is to extract the optimum match through a pattern matching process, given as an input a set of 28 NFR values. The steps consists this pattern matching process are the follows:

1. Collection of the set of the first 8 bending mode frequencies for the
undamaged cantilever beam \( f_{undamaged} \).

2. Repetition of the step 1 for the case of the damaged beam \( f_{damaged} \).

3. Calculation of the NF values using \( f_{damaged}/f_{undamaged} \).

4. Determination of all NFRs by using all possible combinations between the NF values in step 3.

5. Evaluation of the variance between the input NFRs (NFRinput) and database NFRs (NFRmodel) for each CL and CD value in the database, using the following equation:

\[
V = \frac{1}{N} \sum_{i=1}^{N} (\text{NFRmodel}_i - \text{NFRinput}_i)^2
\]

where \( N = 28 \).

6. Extraction of the optimum (CL, CD) values through the minimization process of \( V \).

In order to give an example and test the validity of this damage detection methodology, two different test methods were used as described in the next two subsections. In Section 4.1 the eight bending modes were produced numerically by ANSYS 2016 Workbench simulation software program, while in Section 4.2 they were measured experimentally using magnetoelastic sensors.

### 4.1. ANSYS numerical verification

To make a thorough test on the validity of the method a series of numerical simulations in ANSYS modal analysis were carried out using the physical and geometrical characteristics of Table 1. In particular, 19 designs of different cantilever beams were made which correspond to 18 beams with a single crack at prescribed positions, plus one undamaged beam. Fig. 7 shows the designed undamaged beam model as it prepared for modal analysis and the 3D deformation graphs of the first eight bending modes of the same beam, as they visualized in ANSYS. 

CLs were chosen to be at the points 50 mm, 100 mm, 150 mm, 200 mm, 250 mm and 300 mm from the fixed end, and the CD for each location was varied from 1 mm to 3 mm with step of 1 mm. It should be mentioned that the fixed end of the beam in (Fig. 7e,j) is colored with deep blue and it is the upper left part in each 3D graph. Table 2 shows the numerical frequencies calculated by ANSYS for all CLs and CDs. The next step was to calculate the NFRs from these values and fed them as inputs to the variance method described above, using Matlab software program.

Tables 3 and 4 show respectively the CL and CD results of our method as they are predicted by the model, and the percentage deviations from the fixed values used in ANSYS. It can be seen that in the CL case most of the error values are below 1%, thus indicating an excellent prediction ability of the method concerning the crack location. Also, the error values seem to drop as we move away from the fixed end of the beam, with the highest values being mainly at the point CL = 50 mm. On the other hand, the predicted CD values appear to have error values higher than 1%, with most of the them being from 5 to 10%. Although these deviations are higher compared to the CL ones, they are still within an acceptable range showing thus very good prediction ability of the method concerning the CD.

### 4.2. Experimental verification

The next step is to test the efficiency of the method on predicting the CL and CD values on real beam specimens through their bending modes measured by magnetoelastic sensors. In order to do that, a total number of 19 beam specimens were used (Fig. 8a) out of which, 1 was damage free and the other 18 had damages at fixed locations and depths, same as in the previous subsection. The beam material is aluminum alloy 6063 (Table 5) and all beams were cut at same dimensions by CNC machinery. The type of the damage inserted to the beam specimens can be seen in Fig. 8b and its a transverse rectangular shape damage with a sharp V-notch on top of it (Notch Tip Diameter = 0.1 mm).

Two equal-length ribbons of magnetoelastic material Metglas 2826MB3 (a commercial soft magnetic alloy) were used as vibration sensors and they were attached on each beam specimen (Fig. 8a) using double sided tape. The physical and geometrical properties of the beams and the ribbons are shown in Table 5. The experimental setup conducted in this work is shown in Fig. 9a and includes, among other things, a CARVER hydraulic press which clamps the beam one side with the help of specific curved plates, a detection coil connected to the side, and an arbitrary waveform generator, a linear power amplifier and a mechanical stimulator to excite the beam into bending oscillations (for more details about the experimental setup please refer to our previous work [8]).

Fig. 9b shows the spectrum analysis of the free vibrating undamaged beam and reveals 8 different peaks within a frequency range of 0–9000 Hz, which correspond to the first bending modes. The mechanically excitation of the cantilever beam (CB) with the stimulator forces it into free vibrations, and due to the geometry of the CB only the bending oscillations to the easy axis can be highly excited. The metglas ribbons attached to the free end of the CB follow the bending oscillations and due to their magnetoelastic character change their magnetic state (magnetization) continuously. The change of the magnetic state of the beam produces an alternative magnetic field around it which can induce an alternative electric field (voltage) to any nearby coil through the Faraday’s law of induction. Using the sound card of a laptop through the mic input, this voltage can be recorded. The data recording and processing in this work was done through coding in Matlab software program. The recording time was settled to be 10 s and the sampling rate 48 kHz in order to have the frequency resolution value at 0.1 Hz.

Table 6 contains the experimentally frequency values of the bending modes for all the beam specimens. Making a quick comparison of these frequency values with those predicted in ANSYS software program (Tables 2), it can be seen that they are very close to each other. For example, comparing the frequency values for the undamaged beam the deviation between those is calculated to be: \( \text{Dev} - f_1 = 0.18\% \), \( \text{Dev} - f_2 = 0.09\% \), \( \text{Dev} - f_3 = 0.05\% \), \( \text{Dev} - f_4 = 0.08\% \), \( \text{Dev} - f_5 = 0.19\% \), \( \text{Dev} - f_6 = 0.41\% \), \( \text{Dev} - f_7 = 0.62\% \), and \( \text{Dev} - f_8 = 0.44\% \).

Tables 7 and 8 show respectively the CL and the CD results as they are predicted by the model, and the percentage deviations from the fixed values used on the experiment. The deviations in these tables seem to be a bit higher compared to the ones in Tables 3 and 4. For example, on Table 3 the maximum CL error of 3.2% appears at the values (CL = 50 mm, CD = 3 mm), while the maximum CL error of Table 7 appears at the same pair (CL,CD) but with the value of 6.0%. At the same place, the maximum CD error values on Tables 4 and 8 appears at the values (CL = 50 mm, CD = 1 mm), (CL = 300 mm, CD = 1 mm) and (CL = 50 mm, CD = 3 mm), and is equal to 14% and 20%, respectively. The higher deviations in the case of the experimental results are mainly due to the extra error factors that come into the environment of the experiment, such as, not all the beams specimens are at the perfect shape and dimensions, the temperature of the specimens, the boundary conditions, the efficiency factor of the magnetoelastic sensor, etc. However, the predicted CL and CD from the method are in a very good agreement with the actual values, thus indicating the efficiency of the method in being able to detect the location and the intensity of the crack, as well as the capability of the magnetoelastic sensor to provide the vital information needed for the damage detection.

### 5. Conclusions

A methodology on localization and quantification of crack damages on cantilever beams with use of magnetoelastic sensors is proposed in this study. To develop the methodology, the modeling of the dynamic
The behavior of a damaged and undamaged cantilever beam was accomplished using finite element method and fracture mechanics theory. While the changes on the mass matrix can be neglected with the presence of the crack on the beam element, the stiffness matrix is significantly affected, altering the vibrating behavior of the beam. The simulation of the beam model was carried out using the Matlab.

Fig. 7. (a) The designed beam model that was used, (b) A rectangular meshing geometry was used to discretize the beam into elements. The 3D representation of first 8 bending modes of the undamaged cantilever beam: (c) 1st, (d) 2nd, (e) 3rd, (f) 4th, (g) 5th, (h) 6th, (i) 7th, (j) 8th.
software program, and the normalized frequency (NF) values of the first 8 bending modes were calculated for every crack location (CL) lengthwise in steps of 1/2000 and every crack depth (CD) in the range of 0–65% with a step of 1%. A graphical representation of the calculated NFs is included in this work.

The validity of the method was tested both with numerical and experimental data from damaged beams with a variety of fixed CLs and CDs, and the predicted model results showed a very good agreement with them. Specifically, the average CL and CD error values were 0.7% and 8.5%, respectively, using numerical data and 1.7% and 11.4%,

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Cantilever beam bending modes frequency values as calculated using ANSYS modal analysis for all fixed CLs and CDs, including the undamaged beam.</th>
</tr>
</thead>
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<td>CL (mm)</td>
<td>CD (mm)</td>
</tr>
<tr>
<td>---------</td>
<td>---------</td>
</tr>
<tr>
<td>Undamaged</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>2</td>
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<table>
<thead>
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<th>Table 3</th>
<th>Comparison of the predicted CLs by the model using the ANSYS bending mode frequencies.</th>
</tr>
</thead>
<tbody>
<tr>
<td>CD (mm)</td>
<td>ANSYS CL (mm)</td>
</tr>
<tr>
<td>---------</td>
<td>----------------</td>
</tr>
<tr>
<td>1 Model (mm)</td>
<td>51.1</td>
</tr>
<tr>
<td>Error(%)</td>
<td>2.2</td>
</tr>
<tr>
<td>2 Model (mm)</td>
<td>51.1</td>
</tr>
<tr>
<td>Error(%)</td>
<td>2.2</td>
</tr>
<tr>
<td>3 Model (mm)</td>
<td>51.6</td>
</tr>
<tr>
<td>Error(%)</td>
<td>3.2</td>
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<table>
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<tr>
<th>Table 4</th>
<th>Comparison of the predicted CDs by the model using the ANSYS bending mode frequencies.</th>
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<td>CD (mm)</td>
<td>ANSYS CL (mm)</td>
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<td>----------------</td>
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<tr>
<td>1 Model (mm)</td>
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<td>3 Model (mm)</td>
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<td>Error(%)</td>
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Table 5

<p>| Physical and geometrical properties of Metglas2826MB3 ribbons and aluminum alloy 6063 test-beam. |</p>
<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Property</th>
<th>Metglas2826MB3</th>
<th>Al Alloy 6063</th>
</tr>
</thead>
<tbody>
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<td>1</td>
<td>Length</td>
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<td>350 mm fixed</td>
</tr>
<tr>
<td>2</td>
<td>Width</td>
<td>6 mm</td>
<td>30.25 mm</td>
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<tr>
<td>3</td>
<td>Thickness</td>
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<td>8.18 mm</td>
</tr>
<tr>
<td>4</td>
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<td>2.69 g/cm³</td>
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<td>Modulus of Elasticity</td>
<td>100–110 GPa</td>
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<td>–</td>
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<td>7</td>
<td>Curie Temperature</td>
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<td>–</td>
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<tr>
<td>8</td>
<td>Saturation Induction</td>
<td>0.88 T</td>
<td>–</td>
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</table>

Fig. 8. (a) The beam specimens as they were prepared for experiment. The single beam at the bottom of the figure is the free damage beam, (b) Shape of the damage inserted to the beam specimens (Width $V_w = 1$ mm, V-notch height $V_h = 0.5$ mm, CD varies from [0 to 3] mm with step 1 mm).
respectively, using experimental data. The numerical bending mode frequencies were produced using ANSYS modal analysis and the simulation of a total 19 cantilever beams (1 undamaged beam and 18 damaged beams in fixed locations and depths) were made. In correspondence, the experimental bending modes frequencies were measured using ribbons of magnetoelastic material Metglas 2826MB3 as vibration sensors. The geometry and the material of the beam specimens were the same as with the ANSYS model beams.

Acknowledgments

The present work was financially supported by the "Andreas Mentzelopoulos Scholarships for the University of Patras". Also, the authors would like to thank Prof. G.C. Papanicolaou for valuable conversations and comments on this work.

References


[4] S. Krishnamurthy, R.A. Badcock, V.R. Machavaram, G.F. Fernando, Monitoring pre-

Table 6
Cantilever beam bending modes frequency values as measured experimentally for all CLs and CDs, including the undamaged beam.

<table>
<thead>
<tr>
<th>CL (mm)</th>
<th>CD (mm)</th>
<th>$f_1$ (Hz)</th>
<th>$f_2$ (Hz)</th>
<th>$f_3$ (Hz)</th>
<th>$f_4$ (Hz)</th>
<th>$f_5$ (Hz)</th>
<th>$f_6$ (Hz)</th>
<th>$f_7$ (Hz)</th>
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</thead>
<tbody>
<tr>
<td>Undamaged</td>
<td>54.1</td>
<td>337.7</td>
<td>943.2</td>
<td>1840.9</td>
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<td>942.8</td>
<td>1836.8</td>
<td>3015.9</td>
<td>4472.7</td>
<td>6221.9</td>
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<tr>
<td>2</td>
<td>54.1</td>
<td>337.5</td>
<td>938.9</td>
<td>1818.3</td>
<td>2974.0</td>
<td>4420.7</td>
<td>6181.9</td>
<td>8182.5</td>
</tr>
<tr>
<td>3</td>
<td>54.1</td>
<td>336.6</td>
<td>930.5</td>
<td>1785.0</td>
<td>2848.2</td>
<td>4329.2</td>
<td>6074.8</td>
<td>8161.6</td>
</tr>
</tbody>
</table>

Table 7
Comparison of the predicted CLs by the model, using the experimental bending mode frequencies.

<table>
<thead>
<tr>
<th>CD (mm)</th>
<th>Experimental CL (mm)</th>
<th>50</th>
<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Model (mm)</td>
<td>51.6</td>
<td>102.2</td>
<td>150.5</td>
<td>199.1</td>
<td>247.1</td>
<td>294.2</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>3.2</td>
<td>2.2</td>
<td>0.3</td>
<td>0.4</td>
<td>1.2</td>
<td>1.9</td>
<td></td>
</tr>
<tr>
<td>2 Model (mm)</td>
<td>52.3</td>
<td>102.0</td>
<td>153.6</td>
<td>200.4</td>
<td>246.8</td>
<td>298.9</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>4.6</td>
<td>2.0</td>
<td>2.4</td>
<td>0.2</td>
<td>1.3</td>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>3 Model (mm)</td>
<td>53.0</td>
<td>101.8</td>
<td>151.4</td>
<td>201.2</td>
<td>250.4</td>
<td>298.2</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>6.0</td>
<td>1.8</td>
<td>0.9</td>
<td>0.6</td>
<td>0.2</td>
<td>0.6</td>
<td></td>
</tr>
</tbody>
</table>

Table 8
Comparison of the predicted CDs by the model, using the experimental bending mode frequencies.

<table>
<thead>
<tr>
<th>CD (mm)</th>
<th>Experimental CD (mm)</th>
<th>50</th>
<th>100</th>
<th>150</th>
<th>200</th>
<th>250</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Model (mm)</td>
<td>1.11</td>
<td>1.19</td>
<td>0.90</td>
<td>1.06</td>
<td>1.17</td>
<td>1.06</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>11.0</td>
<td>19.0</td>
<td>10.0</td>
<td>6.0</td>
<td>17.0</td>
<td>6.0</td>
<td></td>
</tr>
<tr>
<td>2 Model (mm)</td>
<td>2.25</td>
<td>2.37</td>
<td>1.96</td>
<td>2.13</td>
<td>2.29</td>
<td>2.13</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>12.5</td>
<td>18.5</td>
<td>10.0</td>
<td>6.5</td>
<td>14.5</td>
<td>6.5</td>
<td></td>
</tr>
<tr>
<td>3 Model (mm)</td>
<td>3.60</td>
<td>3.27</td>
<td>3.35</td>
<td>3.35</td>
<td>3.44</td>
<td>3.27</td>
<td></td>
</tr>
<tr>
<td>Error (%)</td>
<td>20.0</td>
<td>9.0</td>
<td>11.7</td>
<td>11.7</td>
<td>14.7</td>
<td>9.0</td>
<td></td>
</tr>
</tbody>
</table>
Characterization of magnetoelastic ribbons as vibration sensors based on the measured natural frequencies of a cantilever beam

Georgios Samourgkanidis, Dimitris Kouzoudis*

Department of Chemical Engineering, University of Patras, GR 26504 Patras, Greece

**ABSTRACT**

In the current work, magnetoelastic ribbons of metallic glass alloy known as Metglas 2826MB are fully characterized as vibration sensors. The characterization involves seven different sensor parameters such as the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. Two experimental setups where used for the characterization process, one for the frequency response parameter (FR setup) and one for the rest of the parameters (NFR setup). The frequency response parameter was examined for two different states of the ribbon, the non-annealed and the annealed states, and better characteristics were revealed for the annealed state. In the NFR setup, a cantilever beam (CB) was used as a vibrating platform, with two annealed Metglas ribbons attached on its free end using a double-sided tape. The 2nd, 4th and 6th bending modes of the CB were used for the characterization process. Concerning linearity, the ribbons showed an extremely linear behavior, with an average value for the adjusted R-square being $R^2 = 0.99995$. The SNR and quality factor parameters were studied versus the DC magnetization field of the Metglas ribbons (bias field), and the results showed that the implementation of a DC magnetic field increased the strength of the detectable signal without reducing its quality. In particular, the improvement on the detectable signal for the 2nd, 4th and 6th bending modes was 151%, 41% and 27%, respectively. The ribbons stability was examined within a time period of 2h, with the average percentage deviation from the mean frequency being as small as 0.005% and the average change of the frequency with time as small as $f_0 = (1.1 \pm 0.4) \times 10^{-6}$ Hz/min. As for the repeatability parameter, the ribbons were subjected to alternating biasing and showed an excellent performance during repetition cycles, with the average percentage deviation from the reference state being 0.004%, 0.002% and 0.003% for the 1st, 2nd and 3rd cycle, respectively. The last parameter studied was the sensitivity of the ribbons in detecting the shift of the natural frequencies versus CB stiffness, when a crack is introduced. Each bending mode revealed a different value of sensitivity, with the 2nd mode having the lowest and the 6th mode having the highest. The average sensitivity value among the three modes was calculated to be $S_n = (38 \pm 1) \times 10^{-3}$ Hz/(N m$^{-1}$).

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1. Introduction

Sensors are an integrable part of modern technology and even common people are using indirectly 5–10 sensors daily on average through their devices like, smart phones, air-conditions, televisions, refrigerators etc. This is the reason why there is an intense research for the design and implementation of various types of sensors, with focus mainly towards the miniaturization and the low cost, without compromising important sensor qualities. These qualities are typically the linearity, sensitivity, repeatability, stability, selectivity, frequency response etc. Thus, it is imperative for each new sensor design to have the sensors fully characterized so as to know if they fulfill certain specifications.

Several studies have been conducted around the development of vibration sensors, with greater emphasis being placed on their characterization. Abas et al. [1] characterized a vibration sensor made of piezoelectric paper by attaching it on an aluminum cantilever and measuring its frequency response function when it was subjected to impulse loading and sinusoidal excitation. They compared their measurements with a piezoelectric polymer sensor which was placed at the same cantilever and found that the two spectra were very similar with the natural frequencies differing by no more than 1%. Jeng and Chang [2] have used a piezofilm sensor on both a Glass-fiber-reinforced-plastic cantilever and an aluminum cantilever, and characterized it with respect to its frequency response. They found that the sensor behaves as expected

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https://doi.org/10.1016/j.sna.2019.111711
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above 3 kHz, while it underestimates the structural response below that limit. Kim et al. [3] have designed a shear-mode piezoelectric accelerometer for high temperature vibration sensing applications. They measured the frequency response of their sensor as sensitivity versus temperature from 1 Hz to 3 kHz and found that the functional frequency range was 1–350 Hz. Also they showed that the sensor exhibited linearity with the g values and its resolution was quite stable in the temperature range 0–1000 °C. Lin et al. [4] have implemented an embedded microstrain sensor inside a polyurethane (PU) thin film to measure its stress/strain in situ. They only presented data for the linearity of the sensor. Dai et al. [5] have designed a strain sensor using carbon nanotube-based non-woven composites. They did a very detailed sensor characterization as well as mechanical characterization of their composite material. The sensor characterization involved linearity, reversibility and stability. Ge et al. [6] have put together a nice review on flexible and wearable strain sensors for human-motion monitoring and in this review they presented data for linearity and repeatability of various sensors.

In recent years, more and more interest has grown around magnetoelastic materials, and lots of studies have been carried out on the exploitation of their magnetostrictive-magnetoelastic property for sensing purposes [7–15]. Grimes et al. [16] reviewed 68 articles on magnetoelastic resonance sensors, and presented a comprehensive review on the theory, operating principles, instrumentation and key applications of magnetoelastic sensing technology. In general, the magnetoelastic effect (or Villari effect) appears in certain ferromagnetic materials, where their magnetic susceptibility changes with the applied of mechanical stress. By far the best known magnetoelastic materials are the metallic glasses [17,18]. One of the most common used material is this category is the Metglas 2826MB series. It is an iron-nickel based alloy material, with a medium saturation induction and lower magnetostriction, and is usually made in the forms of thin foils or ribbons. Authors in Refs. [19–21] used these series of metallic glasses to develop contact-less and wireless strain sensors. Tan et al. [22] developed a wireless and passive stress sensor by measuring changes in the induced magnetic field of a Metglas 2826MB series strip attached to a solid body in a form a bridge-like structure. Their measurements were the frequency change of the 2nd-order harmonic of the strip, and showed that there is a very good correlation between the sensor’s signal and the applied lateral compressive stress on the structure. Ausanio et al. [23] proposed a elastomagnetic material made of Sm2Co17 particles as a magnetoelastic sensor. They detected vibrations amplitudes from 0.1 to 1 mm with a sensitivity of 3.5 mV/mm, in the frequency range from 5 up to 50 Hz, and showed the usefulness of the sensor in low vibration frequency detection. Zhang et al. [24] made a different approach on developing a magnetic stress detection system, by presenting a non-contact stress detection device based on the magnetoelastic effect, which does not include metallic glasses as transmitters. The device setup included two coils, one excitation and one detection, winded on a magnetic core out of soft magnetic ferrite, and placed orthogonally without contact on a ferromagnetic steel plate. They measured the induced voltage on the detection coil with respect to the applied stress on the plate, and showed the linear dependence between those variables, both experimentally and theoretically. Wichmann et al. [25] presented an advanced measurement device, using magnetoelastic coil sensors, for the direct stress monitoring in prestressing steel elements used in sensitive infrastructure buildings and other building structures subjected to high demands, as e.g., bridges.

In our earlier work [26] we exploited the magnetoelastic property of thin ribbons of the Metglas 2826MB series to develop a vibration sensor for structural health monitoring (SHM) applications. In particular, we used these ribbons to experimentally measure the bending frequencies of cracked cantilever beams and compared the measured frequencies with those analytically predicted by ANSYS to give the method’s proof of principle. The main purpose of the current work is to make a step further and characterize these materials as vibration sensors for structural health monitoring (SHM) applications. In the course of years many researchers have attempted to create techniques that are effective to detect damages on mechanical structures by means of modal parameters like the natural frequencies. Lifshitz et al. [27] were among the first to introduce an experimentation method capable of identifying damages in composite materials, through the shifting of their natural frequencies. Salawu [28] discussed in his review the use of natural frequency as a diagnostic parameter in vibration monitoring structural assessment procedures. Owolabi et al. [29] investigated the effect of crack depth and location on the first three modes of vibrating beams (simply supported and fixed beams). They studied both the frequency and the amplitude of each mode, in order to identify the damage. Recently [30], we presented a new identification methodology of cracks on cantilever beams, where we were comparing the results of a beam model with the input data, by means of a pattern matching process. The validation process was carried out in two different ways, one experimental and one numerical, and the results revealed the very good capability of the method to identify both the crack location and depth. Since the reduction of the natural frequencies of a mechanical structure is a function of the damage location and depth, several researchers developed damage detection methods using frequency contours [31–35]. Sinou [36] made a step further and improved the contour method by proposing a damage identification methodology which can predict not only the location and the size of the damage, but also the orientation, in circular cross section beams.

The current work is an innovation in the field of magnetoelastic sensors, as a characterization process is carried out for these materials as vibration sensors, which includes seven overall sensor parameters, and are the frequency response, linearity, signal to noise ratio (SNR), quality factor, stability, repeatability and sensitivity. The above parameters are very useful for structural health monitoring (SHM) applications, where the monitoring is carried out by measuring the vibration frequencies of the structure.

### 2. Materials and methods

#### 2.1. Materials

The magnetoelastic material used in this study was an Iron-Nickel based commercial ribbon known as Metglas 2826MB. Table 1 shows the ribbon physical and geometrical properties, and Fig. 1a is the room temperature hysteresis loops, respectively. “As-cast” refers to the state of the ribbons as they are produced using a spin melting–quenching method without any further process, while “annealed” refers to the same ribbons after the application of a heat treatment process. Both, annealed and as-cast, hysteresis loops

<table>
<thead>
<tr>
<th>Property</th>
<th>Metglas 2826MB</th>
<th>Cantilever Beam</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>100 mm</td>
<td>700 mm free + 50 mm fixed</td>
</tr>
<tr>
<td>Width</td>
<td>6 mm</td>
<td>30 mm</td>
</tr>
<tr>
<td>Thickness</td>
<td>29 μm</td>
<td>10 mm</td>
</tr>
<tr>
<td>Material Density</td>
<td>7.90 g/cm³</td>
<td>2.69 g/cm³</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>100–110 GPa</td>
<td>68.3 GPa</td>
</tr>
<tr>
<td>Stoichiometry</td>
<td>Fe₃₅Ni₅₀Mo₁₅</td>
<td></td>
</tr>
<tr>
<td>Saturation magnetostriiction λ</td>
<td>12 ppm</td>
<td>–</td>
</tr>
<tr>
<td>Annealed saturation induction</td>
<td>0.88 T</td>
<td>–</td>
</tr>
<tr>
<td>Cure temperature</td>
<td>303 °C</td>
<td>–</td>
</tr>
<tr>
<td>Crystallization temperature</td>
<td>410 °C</td>
<td>–</td>
</tr>
</tbody>
</table>
have been measured along the longitudinal direction of the ribbon. It can be seen from Fig. 1a that the heat treatment process improves the magnetic properties of the Metglas to a significant extent. For example, the saturation magnetic induction $\mu_0M_t$ is increased by 131.6% (from $\mu_0M_t^{\text{as-cast}} \approx 0.38T$ to $\mu_0M_t^{\text{Annealed}} \approx 0.88T$).

Fig. 1b shows a ribbon of Metglas 2826MB, as it has been prepared to be characterized. The preparation method was carried out in the following steps:

1. Two Metglas ribbons were cut at the same dimensions and cleaned up using ethanol. In order to achieve the precise same length on both ribbons, the cutting process was accomplished using a Dahle Guillotine Paper Cutter.
2. Next, the annealing process took place. The ribbons were inserted into an atmospheric furnace at $T = 300\, ^\circ C$ for 1 h. The value $T = 300\, ^\circ C$ was picked as a safe limit in order not to get close to the Curie and crystallization temperatures.
3. Finally, both ribbons were let to cool off slowly for 10 minutes and were cleaned up again using ethanol.

Since the purpose of this study was to characterize the Metglas ribbons as a structural vibration sensor, an aluminum alloy 6063 material cantilever beam (CB) was picked as the mediator for the characterization process (Fig. 2a) and the prepared ribbons were attached on the free end of it using double-sided tape (Fig. 2b). The dimensions of the beam were set using a CNC (Computer Numerical Control) machinery and Table 1 shows the physical and geometrical characteristics of it. Three in total natural frequencies of CB were chosen for the process and are the 2nd, 4th and 6th bending mode of the CB (Fig. 3 shows the 3D visualization of these modes using ANSYS program). The reason for choosing bending modes rather than torsional is because the torsional modes are very difficult to excite and detect in CBs, due to their very small oscillation amplitude. Also, the reason for choosing these specific bending modes is because they cover a decent range in the frequency response of the ribbons. At this point it would be reasonable to mention that the detectable signal strength was sufficiently strong, without the need for further amplification, by using two ribbons instead of one, because both ribbons were oscillated simultaneously and in the same way, due to the shape of the bending oscillations.

2.2. Experimental setup description

In the present work two different experimental setups were used for the characterization of the Metglas ribbons (Fig. 4). One setup was to characterize the frequency response parameter (Fig. 4a), while the other setup was to characterize the rest of the parameters (Fig. 4b-d). Totally seven parameters were characterized and are listed below:

- Frequency response
- Linearity
- Signal to noise ratio (SNR)
- Quality factor
- Stability
- Reversibility
- Sensitivity

From now on, the experimental setup for measuring the frequency response will be referred as the FR (Frequency Response) setup, whereas the setup for measuring the rest of the parameters will be referred as the NFR (Non-Frequency Response) setup. Fig. 4a shows the devices that were part of the FR setup. A homemade single layer solenoid detection coil was placed in the center of a pair of homemade Helmholtz coils. Table 2 contains the physical and geometrical properties of the coils. Fig. 5a shows a schematic diagram of the setup. The electrodes of the detection coil were connected to a digital oscilloscope which monitors the output voltage of the coil, while the electrodes of the Helmholtz coils were connected to a waveform generator and parallel to it a multimeter, in order to monitor the voltage of the Helmholtz coils.

For the case of the NFR setup, Fig. 5b shows the schematic diagram of the connections between the devices, and it can be seen that more devices are involved in this setup (compared to the FR setup). On this setup, the waveform generator was connected to a mechanical vibrator which is attached to the cantilever beam close to its fixed end. Parallel to the mechanical vibrator a multimeter was connected to monitor the vibrator excitation voltage. Two carved plates, consisting of aluminum alloy 7075, are used to fix the one end of the beam, leaving out a free length of $L = 700\, mm$ to vibrate. The free end of the beam was, along with the attached Metglas ribbons, located inside a homemade solenoid detection coil, which properties are shown in Table 2. The electrodes of the detection coil were connected both to the multimeter and the DAQ (Data Acquisition System) of a laptop. Finally, a DC power supplier was connected to the Helmholtz coils in order to create the bias field (DC magnetic field) for the magnetization of the Metglas ribbons (in the next subsection, the effect of the bias field will be discussed briefly). In particular, the following devices were used in the current work (for both setups):

- Waveform Generator: KEITHLEY 3390 50 MHz Arbitrary Waveform Generator
Fig. 2. (a) Aluminum alloy 6063 material beam specimen, (b) attached Metglas ribbons on the one end of the beam specimen.

Fig. 3. 3D visualization of the cantilever beam’s bending mode shapes using ANSYS program: (a) 2nd, (b) 4th and (c) 6th bending modes. The fixed end of the beam is colored with deep blue and it is the upper left part in each 3D graph. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

Fig. 4. (a) Experimental setup for measuring the ribbon’s frequency response, (b) a panoramic view of the non-frequency response experimental setup, (c) a detailed view of the left side of the non-frequency response experimental setup, (d) a detailed view of the right side of the non-frequency response experimental setup.

Table 2
Physical and geometrical properties of the detection and Helmholtz coils.

<table>
<thead>
<tr>
<th>Property</th>
<th>FR detection coil</th>
<th>NFR detection coil</th>
<th>Helmholtz coils</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coi diameter</td>
<td>29.74 mm</td>
<td>60 mm</td>
<td>390 mm</td>
</tr>
<tr>
<td>Wire diameter</td>
<td>0.16 mm</td>
<td>0.16 mm</td>
<td>1.05 mm</td>
</tr>
<tr>
<td>Tread number</td>
<td>625</td>
<td>1600</td>
<td>48 (each coil)</td>
</tr>
<tr>
<td>Wire material</td>
<td>Copper (Cu)</td>
<td>Copper (Cu)</td>
<td>Copper (Cu)</td>
</tr>
<tr>
<td>Electrical resistance</td>
<td>34.0 Ω</td>
<td>250.3 Ω</td>
<td>3.1 Ω (total)</td>
</tr>
<tr>
<td>Coefficient of inductance</td>
<td>3.2 mH</td>
<td>139.3 mH</td>
<td>4.4 mH (total)</td>
</tr>
</tbody>
</table>
Fig. 5. Schematic diagram of the connections between the devices in the experimental setups: (a) FR setup and (b) NFR setup.

- Digital Oscilloscope: Tektronix TDS 1002 Digital Storage Oscilloscope (60 MHz–1GS/s)
- Multimeter: KEITHLEY 2000 Multimeter
- DC Power Supplier: Tti QL355P Power Supply
- Mechanical Vibration Generator: Frederiksen Vibration Generator no. 2185.00

### 2.3. Experimental setup optimization

During the FR experimental procedure, the Helmholz coils were generating AC magnetic field and the procedure was divided in two parts. The first part was to find the optimum operating parameters of the Helmholz coils, while the second part was the measurement of the ribbons frequency response based on these parameters, and will be discussed in the next section (Section 3.1). As far as the first part is concerned, it is necessary to determine the resonant frequency of the complex RL circuit (resistor–inductor circuit) which is formed because of the connections of the Helmholz coils to the waveform generator. This is done in order to define a suitable frequency range away from the resonance. Fig. 6a presents the results of this part in a logarithmic scale. The magnitude was calculated using the equation:

\[
\text{Magnitude (dBs)} = 20 \log \left( \frac{V_p}{V_i} \right)
\]

where \(V_p\) is the inductive voltage on the detection coil due to the Faraday’s law of induction and \(V_i\) is the voltage of the Helmholtz coils. It is clear that there are three distinct areas in this plot, the low-frequency area (below \(f_1 = 50\) Hz), the mid-frequency area (between \(f_1\) and \(f_2 = 50\) kHz) and the high-frequency area where the resonance peak is at \(f_t = 313\) kHz. Since in this work we are interested to use the Metglas ribbons as a structural vibration sensors, all frequency response measurements were performed in the range of [0.01–50] kHz, so as to have a clear sensor signal free of the coil resonance. Another important optimization procedure that was performed was to find the homogeneity domain of the magnetic field within the coils. Fig. 6b shows the change of the magnetic field along the Helmholz coils axis, where the position \(z = 0\) mm corresponds to the center of one of the two coils (the center of the other coil is located at position \(z = 210\) mm). The magnetic field was produced by a DC current of \(I = 1\) A and it was measured with the help of gaussmeter (FW Bell 7010 Single Channel Gauss/Tesla Meter, 50 kHz). It is clear from Fig. 6b that the field reaches the maximum point and starts to stabilize about \(L = 50\) mm from the center of one of the two coils, and remains stable up to the value \(L = 150\) mm, where it starts to decrease. In all experiments, the detection coils shown in Fig. 5a and b were located entirely within the homogeneous region of the Helmholtz coils.

Similarly to the FR setup procedure, optimization was performed also in the NFR setup but with different parameters, such as the boundary conditions of the CB and the bias field (DC magnetic field). Concerning the first parameter, Fig. 7a presents the dependence of the 2nd bending mode frequency with the pressure load applied at the fixed end of the CB. It can be seen that as the applied load increases, the frequency value comes to a saturation \((f_t = 102.10\) Hz). So, it can be considered that the boundary conditions of the fixed end are optimum under high applied loads, such as 10kN. Concerning the second parameter, Fig. 7b presents the dependence of the signal voltage of the 2nd bending mode on the bias field for an applied load of 10kN. Here, the signal voltage shows an increase with the applied field, up to a certain point, and then a gradual decrease. The peak of the curve occurs at \(B = H_k = 0.64\) Gauss, where \(H_k\) is an important property of magnetic materials known as “anisotropy field”, and is defined as the magnetic field required to disorient all the magnetic domains from the anisotropy direction. This optimum field was used to study the characterization parameters of linearity, stability and reversibility, as it was important for these parameters to have a maximum signal strength.

### 2.4. Experimental methods

For each characterization parameter studied in this work a specific experimental method was followed in order to acquire the results. Starting with the frequency response parameter, the procedure of measuring was similar to the aforementioned frequency response measurement of the Helmholtz coils, with the only difference being that one of the ribbons is placed alone inside the detection coil. The measurements were taken within the frequency range of [10 Hz–50 kHz] and the intensity of the applied AC magnetic field was kept fixed at 0.045 Gauss for all frequencies, by feeding the Helmholtz coil with a constant AC current of 20 mA. This intensity value was chosen because the ribbon at this value exhibits linearity with respect to the AC field.

One of the most important sensor characteristics is the linearity, which means how linear is the sensor output signal with respect to changes to its input. Here, as an input was used the vibrator excitation voltage, which is proportional to the excitation amplitude of the beam, and as an output the voltage developed on the detection coil. The principle of operation of this linearity characterization experiment is the following: The mechanical vibrator sets the beam in oscillatory motion and the Metglas ribbons, attached to the free end of the CB, follow the oscillation, which alters their magnetization due to their magnetoelastic nature. The change on the magnetic state of the ribbons can be detected using a detection coil, through the Faraday’s Law of induction.
APPENDIX C. PUBLISHED WORK IN INTERNATIONAL SCIENTIFIC JOURNALS AS PART OF THE PREPARATION OF THIS THESIS

Fig. 6. (a) The frequency response of the Helmholtz coils – waveform generator circuit, (b) magnetic field along the axis crossing the center of the Helmholtz coils.

Fig. 7. (a) Frequency dependence of the 2nd bending mode of the cantilever beam with the applied load at the fixed end, (b) change of the signal voltage of the 2nd bending mode versus the bias field.

The signal to noise ratio (SNR) is another important sensor parameter that shows how high is the signal with respect to the noise which is always present in every measurement. It is a clear number and it is defined as the ratio of the sensor signal (voltage $V_s$) under a given stimulus, divided by the same signal ($V_n$) when the stimulus is absent. As it has been shown above, the magnetic state of the Metglas ribbons, and thus their sensor signal, changes with the application of a DC magnetic field (Fig. 6b), which is known as “bias field”. So, here the SNR is studied versus the bias field for a certain dynamic stress. All measurements were taken with the mechanical vibrator voltage fixed at $V_{mm} = 172$ mV and the SNR was calculated in dBs using Eq. (1), where the voltage $V_{ij}$ was replaced with the background noise voltage $V_n$. The background noise was measured under the same operational conditions of the experiment but without the Metglas ribbons attached on the CB. This was accomplished by using a second beam specimen with the same physical and geometrical properties.

Another parameter that was studied versus the bias field is the quality factor of the resonance peaks of the bending modes. In this case the measurements were carried out using the Matlab software program, where a simple algorithm was created to record and calculate the FFT (fast Fourier transform) spectrum of the signal from the detection coil. The total record time was settled to be 50 s and the sampling rate 5 kHz. Based on these settings the frequency resolution was calculated to be 0.02 Hz, which defines the frequency error value of each bending mode.

The time stability of the Metglas ribbons was studied with respect to the frequency, for each of the three bending modes of the CB. The mechanical vibrator was oscillating the CB at a constant frequency for a total time of 120 min and a measurement was taken from the detection coil, through the laptop’s sound card, every 2 min. The signal was FFT analyzed with the help of the Matlab software in order to extract the frequency and the recording settings of the algorithm were the same as those used to in quality factor parameter.

In order to study the repeatability of the sensor, the boundary conditions of the CB were cycled between two states for a total of three cycles, and every cycle was consisted of 20 min intervals, with each state sharing an equal time of 10 min per cycle (frequency measurements were taken every minute). The reason of picking the applied load as the changing variable in this experiment is because, as it has been shown in Fig. 6a the frequency values of the bending modes are directly dependent on it, and also it is easy to cycle it in order to have the precise same conditions on each cycle.

The last parameter studied was the sensitivity of the ribbons in detecting the shift $\Delta f$ of the natural frequencies versus CB stiffness change $\Delta k$, when a crack is introduced. Special attention was given to the crack location because if it happens to be close or right on a mode nodal point, then it has little or no effect to the mode resonance frequency. So, the crack location for all three bending modes that were tested was picked to be close to the middle of the CB, area which was free of nodal points for all the three modes. For the calculation of the stiffness $k$, two different variables were measured,
changes its linear character and keeps increasing gradually, till the point where the saturation begins (500 Hz). The deviation on the magnitude between the lower limit of this range (100 Hz) and the saturation is 2.54 dB for the single coil, 3.67 dB for the non-annealed Metglas ribbon and 3.51 dB for the annealed Metglas ribbon. Finally, saturation occurs at the high-frequency range, where the extra gain on the signal with the insertion of the ribbons within the coil is 2.10 dB for the non-annealed ribbon and 5.39 dB for the annealed. The gain difference between the non-annealed and annealed state is equal to 3.29 dB, and corresponds approximately to 157% increase on the magnitude between those states. Based on these results, the annealed state of the ribbon presents better frequency response characteristics compared to the non-annealed state, and especially in the extra gain on the signal. For the rest of the current work, all experiments were done with the use of annealed ribbons.

3.2. Linearity

Fig. 9a–c presents the sensor’s linearity plots for the 2nd, 4th and 6th bending modes of the CB. From these data plots it is apparent that there is a strong linearity, with $R^2$ values being very close to unity. This linearity is a combination of the vibrator linearity (oscillating amplitude versus input voltage) and the sensor linearity (output signal versus oscillating amplitude). As the vibrators are known to be very linear devices, it is concluded that the sensor shows very high linearity as well, which reveals the high coupling between the magnetic and the elastic properties of the ribbons.

3.3. Signal to noise ratio (SNR)

Fig. 10a–c shows the SNR results for the 2nd, 4th and 6th bending mode of the CB, correspondingly. It can be seen that SNR increases rapidly with the magnetic field up to a certain maximum, and then decreases gradually with the increasing magnetic field. The magnetic field values on the maximum point seem to not change drastically between the bending modes, and their values varies within the range of [0.8–0.9] Gauss. The percentage increase between the minimum and the maximum SNR value for the 2nd, 4th and 6th bending modes is 151%, 41% and 27%, respectively.

3.4. Quality factor

Fig. 11a to c shows the resonance data and the fitted peaks for the case $B_{bias} = 0$ Gauss. The shape of the peaks corresponds to a Cauchy–Lorentz distribution for which the quality factor is given by the relationship $Q = f_p / BW$, where $f_p$ is the frequency corresponding to the maximum value of the distribution and BW is the bandwidth which corresponds to the full width at half maximum (FWHM) of the peak. Fig. 11d–f presents graphically the calculated values of the $Q$ factor versus the bias field for each bending mode. The range of the applied field was picked to be the same with this of the SNR ([0–1.4] Gauss). As it can be seen, the changes of the $Q$ factor are mostly stable for the 2nd mode, while in the case of the 4th and 6th mode there is a slight increase and saturation. In particularly, the mean value for the 2nd mode is

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**APPENDIX C. PUBLISHED WORK IN INTERNATIONAL SCIENTIFIC JOURNALS AS PART OF THE PREPARATION OF THIS THESIS**

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Fig. 9. Metglas ribbons linearity graphs through the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam. (d) Linearity comparison plot for all bending modes.

$Q_{2\text{nd}}^{\text{mean}} = (4355 \pm 0.1\%)$, and the saturation values for the 4th and 6th modes are $Q_{4\text{th}}^{\text{sat}} = (21335 \pm 0.01\%)$ and $Q_{6\text{th}}^{\text{sat}} = (76510 \pm 0.004\%)$, respectively. Comparing these results with those of the SNR it can be seen that, within the range [0.8–0.9] Gauss where the SNR is optimum for all modes, the quality factor is almost stable for the 2nd mode and slightly increasing for the 4th and 6th modes. Thus, it can be concluded that the implementation of a DC magnetic field for the magnetization of the Metglas ribbons improves the characteristics of the detectable signal, by increasing its strength without reducing its quality.

3.5. Stability

Fig. 12 shows the stability results for all three bending modes. For the 2nd mode (Fig. 12a) the mean value along with the percentage deviation is $Q_{2\text{nd}}^{\text{mean}} = (102.10 \pm 0.01\%)$, and for the 4th and 6th modes (Fig. 12b and c, respectively) is $Q_{4\text{th}}^{\text{mean}} = (550.14 \pm 0.004\%)$ and $Q_{6\text{th}}^{\text{mean}} = (1322.08 \pm 0.002\%)$, respectively. Also, a linear fitting on the data of Fig. 12 shows very small changes on the frequency values with time. Specifically, the slope of the linear fit for the 2nd, 4th and 6th modes is $F_{2\text{nd}} = (1.4 \pm 0.3) \times 10^{-4}$ Hz/min, $F_{4\text{th}} = (1.1 \pm 0.8) \times 10^{-4}$ Hz/min and $F_{6\text{th}} = (0.7 \pm 0.9) \times 10^{-4}$ Hz/min, respectively. From these results it is clear that the Metglas ribbons exhibit excellent stability with time.

3.6. Repeatability

Repeatability results are shown in Fig. 13 where the red line represents schematically the cycling of the applied load between 2 and 9 kN, and the blue circles the measured frequency data. Table 3 contains the calculated mean values and percentage deviations of the plotted data. The mean values are calculated in each state of the cycles and the percentage deviation gives the % difference of each cycle with respect to the reference state, for both load values. As these percentages are very small, it can be concluded that the sensor exhibits high repeatability.

3.7. Sensitivity

Shown in Table 4 are the experimental results of the procedure described in Section 2.4 for the sensitivity parameter. As it was expected, the presence of the crack alters the stiffness of the beam by reducing it, and also affects the beam’s dynamic behavior through the frequency change of each bending mode. The frequency change data for the three modes versus CD is plotted in Fig. 14a–c and the stiffness change data versus the CD in Fig. 14d. As can be seen both, frequency and stiffness, values are decreasing non-linearly versus CD, with the rate of change exponentially increased. On the other hand, the plots of $\Delta f$ data versus $\Delta k$ presented in Fig. 15, for all three modes, show a much better linearity, and their slope can be used as the definition of the sensor sensitivity. The vertical axis corresponds to the $\Delta f$ change of the frequency between the undamaged and damaged state of the beam, while the horizontal axis corresponds to the $\Delta k$ change of the stiffness, for the same states of the beam. The slopes (sensitivity) of the linear fits for the 2nd, 4th and 6th bending modes are calculated to be $S_{2\text{nd}} = (7.9 \pm 0.2) \times 10^{-3}$ Hz/[N m$^{-1}$], $S_{4\text{th}} = (34.2 \pm 0.6) \times 10^{-3}$ Hz/[N m$^{-1}$] and
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Fig. 10. Metglas ribbons SNR graphs for the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam. (d) SNR comparison plot for all bending modes.

Fig. 11. Resonance data and fitted peaks for the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam. Quality factor parameter versus bias field for the (d) 2nd, (e) 4th and (f) 6th bending modes.
Fig. 12. Metglas ribbons stability graphs versus time for the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam.

Fig. 13. Metglas ribbons repeatability graphs versus time and applied load, for the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam.

Table 3
Calculated mean values and percentage deviations for each cycle and bending mode.

<table>
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<tr>
<th>Mode no.</th>
<th>Cycle no.</th>
<th>Load = 2 kN</th>
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<th>Load = 9 kN</th>
<th>Deviation (%)</th>
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<td>Mean value (Hz)</td>
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<td>1303.269</td>
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Fig. 14. Frequency changes with respect to the crack depth for the (a) 2nd, (b) 4th and (c) 6th bending modes of the cantilever beam. (d) Reduction of the CB stiffness with the crack depth.
The experimentally measured sensitivities of different crack depths within the cantilever beam are shown in Fig. 15. The sensitivities were determined for crack depths ranging from 0% to 7%.

<table>
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<tr>
<th>Crack depth (mm)</th>
<th>CB stiffness (N/m)</th>
<th>Bending modes (Hz)</th>
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<tr>
<td></td>
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<td>2nd</td>
</tr>
<tr>
<td>0 (0%)</td>
<td>13911</td>
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</tr>
<tr>
<td>1 (10%)</td>
<td>13908</td>
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</tr>
<tr>
<td>2 (20%)</td>
<td>13895</td>
<td>101.56</td>
</tr>
<tr>
<td>3 (30%)</td>
<td>13788</td>
<td>100.98</td>
</tr>
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<td>4 (40%)</td>
<td>13670</td>
<td>99.78</td>
</tr>
<tr>
<td>5 (50%)</td>
<td>13372</td>
<td>97.86</td>
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<td>6 (60%)</td>
<td>12968</td>
<td>94.10</td>
</tr>
<tr>
<td>7 (70%)</td>
<td>12267</td>
<td>88.82</td>
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</table>

$S_{6th} = (77 \pm 3) \times 10^{-3}$ Hz/(N·m$^{-1}$), respectively. It is clear from these results that each mode presents its own sensitivity to stiffness, with the 2nd having the smallest and the 6th the largest sensitivity.

4. Conclusions

A full characterization of magnetoelastic ribbons as vibration sensors is given in this work, with the ribbons composed of metallic glass alloy Metglas 2826MB. Seven different sensor parameters were studied using two different experimental setups. The first setup was used to examine the frequency response of the ribbons, while the second setup, which contained a vibration platform (cantilever beam), was used to examine the rest of the parameters (linearity, signal to noise ratio SNR, quality factor, stability, repeatability and sensitivity), through the measured bending modes frequencies of the vibrating platform (2nd, 4th and 6th bending modes).

In the case of the frequency response, two different states of the ribbon were examined, the non-annealed and the annealed state. The annealed state showed better frequency response characteristics in relation to the non-annealed, especially on the extra gain of the signal magnitude ($\approx 157\%$ increase), and for the rest of the characterization process only the annealed ribbons were used. Concerning linearity, the ribbons showed an extremely linear behavior with respect to the oscillation amplitude of the cantilever beam (CB), with the average value of the adjusted R-square from all three bending modes being $R^2 = 0.99995$. The SNR parameter of the ribbons was examined with respect to the DC magnetization field (bias field), and it was observed that it reaches a maximum within the range [0.8–0.9] Gauss for all three bending modes. The improvement on the detectable signal, upon the application of the bias field, was about 151%, 41% and 27% for the 2nd, 4th and 6th bending modes, respectively. The resonance peak’s quality factor of each bending mode was also examined versus the bias field. It was observed that within the range [0.8–0.9] Gauss where the SNR was optimum for all modes, the quality factor was stable for the 2nd mode and slightly increased for the 4th and 6th modes, concluding that the implementation of a DC magnetic field for the magnetization of the ribbons improves the characteristics of the detectable signal, by increasing its strength without reducing its quality.

Sensor’s stability parameter was examined within a time period of 2 hours with all the experimental setup settings (beam’s boundary conditions, vibrating voltage, ribbon’s magnetization field, etc.) fixed and stable, and showed an excellent behavior, with the average percentage deviation from the mean frequency for all bending modes being as small as 0.005%. The changes on the frequency with time for each bending mode were also small, and they calculated to be $F_{2nd} = (1.4 \pm 0.3) \times 10^{-4}$ Hz/min, $F_{4th} = (1.1 \pm 0.8) \times 10^{-4}$ Hz/min and $F_{6th} = (0.7 \pm 0.9) \times 10^{-4}$ Hz/min. On the other hand, the repeatability of the sensor was examined by following two applied load values cyclically at the fixed end of the CB (2 and 9 kN). The average percentage deviation from the reference state was 0.004%, 0.002% and 0.003% for the 1st, 2nd and 3rd cycle, respectively. Finally, the sensor’s sensitivity parameter was examined versus the stiffness change of the CB, when a transverse crack was introduced. The results showed sensitivity values of $S_{2nd} = (7.9 \pm 0.2) \times 10^{-3}$ Hz/(N·m$^{-1}$), $S_{4th} = (34.2 \pm 0.6) \times 10^{-3}$ Hz/(N·m$^{-1}$) and $S_{6th} = (77 \pm 3) \times 10^{-3}$ Hz/(N·m$^{-1}$) for the 2nd, 4th and 6th bending modes, respectively, indicating the increase of it in higher modes.

Conflict of interest

None declared.

Acknowledgements

The current work has been supported financially by the “Andreas Mentzelopoulos Scholarships for the University of Patras”.

References

Biographies

Prof. Dimitris Kouzoudis received his Bachelor Degree in Physics from the University of Ioannina, Greece in 1990, his M.S. in Physics and Material Science and his Ph.D. in Physics from the University of Kentucky, U.S.A., and between 2001 and 2002 as a consultant at the headquarters of the "Kindred Healthcare" company. At 2003 he received a Lecturer position at the Material Science Department of University of Patras, Greece and at 2005 he received a tenure-track Assistant Professor position at the Department of Engineering Sciences at the same University. Between 2012 and 2013, he took a sabbatical leave at the HCT Engineering School in the United Arab Emirates. At 2014 he received a tenure Assistant Professor appointment at the Chemical Engineering Department of University of Patras, Greece and at 2017 he became an Associate Professor at the same department. He is the editor-in-chief of the scientific peer-review journal “Sensor Letters” between 2003 and 2012 and focus on the application of different sensor platforms towards monitoring environmental/chem/bio parameters, as gas concentration, small mass loads, pressure flow, velocity, humidity, and precipitation of biological salts in aqueous solutions, coagulation time, and glucose concentration, the use of zeolite films as active sensor layers for the detection of volatile organic compounds.

Selected Publications


Georgios Samourgkanidis received his Bachelor Degree in Physics from the University of Patras, Greece in March 2015, his M.S. in Chemical Engineering of University of Patras in March 2017 and from April 2017 he continues as a Ph.D. candidate at the same department. Between 2013 and 2014 he completed his graduate diploma thesis on “Study and analysis of deformations in static and rotating aluminium blades using magneto-elastic sensors” in the department of chemical engineering. He joined in 2011 and during his studies in Physics, he was awarded with the “AlexandrosTheodouso” scholarship, which is a scholarship from the department of physics of University of Patras. Also, at 2017 he was awarded with the LIMMAT scholarship from the department of chemical engineering of University of Patras.

Selected Publications:


ΠΡΟΣΩΠΙΚΕΣ ΠΛΗΡΟΦΟΡΙΕΣ

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ΕΠΑΓΓΕΛΜΑΤΙΚΗ ΕΜΠΕΙΡΙΑ

Σεπτέμβριος 2017 – Μάιος 2020
- Προετοιμασία εβδομαδιαίων εργασιών Φυσικής I & II
- Επιπηρητής και βοηθός συντονιστή σε εργαστήρια φυσικής (10 ώρες/ Εβδομάδα)
- Φροντιστηριακά μαθήματα φυσικής (4 ώρες/ Εβδομάδα)

Σεπτέμβριος 2015 – Μάρτιος 2017
- Φροντιστηριακά μαθήματα φυσικής (4 ώρες/ Εβδομάδα) στο τμήμα χημικών μηχανικών (Επικουρικό έργο)

Σεπτέμβριος 2009 – Μάιος 2015
- Βοηθός τεχνικού συντηρητή δικτύων ίντερνετ
- Επισκευές και συντήρηση σταθερών και φορητών υπολογιστών
- Ελεγκτής φώτων και ήχου σε θεατρικές παραστάσεις
- Ιδιαίτερα μαθήματα φυσικής και μαθηματικών

Αύγουστος 2006 – Γενάρης 2009
- Καθαριστής πολυκατοικιών και γραφείων
- Βοηθός Ηλεκτρολόγου μηχανικού σε ηλεκτρολογικές συντήρεσεις πολυκατοικιών
- Οικοδόμος – Εργάτης

ΕΚΠΑΙΔΕΥΣΗ ΚΑΙ ΚΑΤΑΡΤΙΣΗ

Διδακτορικό Δίπλωμα Ειδίκευσης (PhD) – Τομέας: << Επιστήμη και Τεχνολογία Υλικών >> – Τίτλος Θέματος: << Experimental study and characterization of magnetoelastic ribbons as vibration sensors, and their application for the identification of cracks in cantilever beams through the dynamic behavior of the beam >>

Τμήμα Χημικών Μηχανικών Πανεπιστημίου Πατρών

Γενικά
- Ανάπτυξη ικανοτήτων συγγραφής δημοσιεύσεων και ερευνητικών προτάσεων
- Ανάπτυξη ικανοτήτων προετοιμασίας posters και ομιλιών σε επιστημονικά συνέδρια
- Μεθοδολογία στη προετοιμασία πειραματικών διατάξεων
- Βελτίωση ικανοτήτων συντηρητικής και ανάλυσης πειραματικών δεδομένων
- Βελτίωση ικανοτήτων συγγραφής και προσαρμογής μηχανικών δομών
- Βελτίωση προγραμματισμικών ικανοτήτων
- Απαίτηση κατανόησης και χρήση αγγλικής γλώσσας
- Ανάλυση ευθυνών και πρωτοβουλιών σε θέματα εργαστηριακής διοίκησης
- Βελτίωση ικανοτήτων ακαδημαϊκού έργου (διαδικασία μαθημάτων και εργαστηριακές ασκήσεις)
Μεταπτυχιακό Δίπλωμα Ειδίκευσης (Master) – Τομέας: << Επιστήμη και Τεχνολογία Υλικών >> – Τίτλος Θέματος: << Σύνθεση Μαγνητοελαστικών Υλικών και η Χρήση τους ως Αισθητήρες για την Καταγραφή των Μηχανικών Τάσεων και Παραμετρώσεων Λόγω Αστοχίων σε Πτερύγια Περιστρεφόμενων Ελικών >>
Τμήμα Χημικών Μηχανικών Πανεπιστημίου Πατρών

Γενικά
- Εκμάθηση τεχνικών εικονικής σχεδίασης και προσομοίωσης μηχανικών δομών
- Ανάπτυξη προγραμματιστικών ικανοτήτων
- Βελτίωση αγγλικής γλώσσας
- Χρήση μηχανογενειακών μηχανημάτων όπως φρέζα και τόρνος
- Ανάπτυξη ικανοτήτων προετοιμασίας και εκτέλεσης πειραμάτων
- Εκμάθηση ηλεκτρονικής μικροσκοπίας σάρωσης (SEM)

Σεπτέμβριος 2009 – Φεβρουάριος 2015
Πτυχίο Φυσικής – Τίτλος Διπλωματικής: << Μελέτη Ηλεκτρικής Αγωγιμότητας και Φωτοαγωγιμότητας σε Διάφορες Θερμοκρασίες του Διοξειδίου του Τιτανίου με Διαδοχική Προσρόφηση Ιόντων Μολύβδου και Θείου >>
Τμήμα Φυσικής Πανεπιστημίου Πατρών

Γενικά
- Εκμάθηση βασικών αρχών εργαστηριακής υγιεινής και ασφάλειας
- Εκμάθηση τρόπου χρήσης εργαστηριακού εξοπλισμού (Πολύμετρα, Θάλαμοι κενού, Φιάλες αερίων, συνδεσμολογία ηλεκτρονικών συσκευών για την διεξαγωγή πειραμάτων κ.α.)
- Εμπάθυνση σε ειδικά θέματα επιστήμης υλικών όπως τεχνικές χαρακτηρισμού υλικών, πειραματικές διατάξεις, συνδυαστικοί τρόποι μελέτης κ.α.

ΑΤΟΜΙΚΕΣ ΔΕΞΙΟΤΗΤΕΣ

Μητρική γλώσσα
Ελληνικά

Λοιπές γλώσσες
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<td>Επικοινωνία</td>
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<td>Ρωσικά</td>
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First Certificate in English (FCE) του Πανεπιστημίου CAMBRIDGE B2

Κοινό Ευρωπαϊκό Πλαίσιο Αναφοράς για Γλώσσες

Επικοινωνιακές δεξιότητες
Ος προπτυχιακός φοιτητής ήμουν για αρκετά χρόνια μέλος θεατρικής ομάδας, και έχω πάρει μέρος σε πολλές θεατρικές παραστάσεις. Επίσης, για 3 χρόνια ήμουν βασικός στη ποδοσφαιρική ομάδα της ΦΕΠ πανεπιστημίου Πατρών.

Οργανωτικές / διαχειριστικές δεξιότητες
- Για αρκετά χρόνια έχω υπάρξει συντονιστής στα φώτα και στον ήχο, σε συναυλίες και θεατρικές παραστάσεις. Επίσης, για 3 χρόνια ήμουν βασικός στη ποδοσφαιρική ομάδα της ΦΕΠ πανεπιστημίου Πατρών.
- Επί του παρόντος είμαι υπεύθυνος για την οργάνωση του πειραματικού μέρους των προπτυχιακών διπλωματικών εργασιών που εκτελούνται στο εργαστήριο του βρίσκομαι.
Ψηφιακές δεξιότητες
- Καλός χειρισμός του Office (επεξεργασία κειμένου, λογιστικά φύλλα, παρουσιάσεις)
- Καλός χειρισμός της Matlab (δημιουργία αλγορίθμων καταγραφής και ανάλυσης πειραματικών δεδομένων)
- Καλός χειρισμός του ANSYS (σχεδίαση και προσομοίωση μηχανικών δομών προς μελέτη)
- Καλός χειρισμός του OriginPro (επεξεργασία και γραφική απεικόνιση πειραματικών δεδομένων)
- Καλός χειρισμός του LaTeX (προετοιμασία κειμένων και δημοσιεύσεων)
- Καλός χειρισμός του Adobe InDesign (Προετοιμασία Posters)

Άλλες δεξιότητες
- Ορειβασία: Μέλος ερευνητικής ορειβατικής ομάδας στο πανεπιστήμιο μου
- Αστροπαρατήρηση: Μέλος ομάδας ερασιτεχνών αστροπαρατηρητών

Δίπλωμα οδήγησης: Β

ΠΡΟΣΘΕΤΕΣ ΠΛΗΡΟΦΟΡΙΕΣ

Συνέδρια
- ΧΧVIII Πανελλήνιο Συνέδριο Φυσικής Στερεάς Κατάστασης και Επιστήμης Υλικών (2012)
- Δεύτερο Διεθνές Πανελλήνιο Συνέδριο Κρυσταλλογραφίας στο Πανεπιστήμιο Πατρών (2013)
- ΧΧΧΙΙΙΙ Πανελλήνιο Συνέδριο Φυσικής Στερεάς Κατάστασης και Επιστήμης Υλικών (2019)

Δημοσιεύσεις

Κριτής σε επιστημονικά περιοδικά (ημερομηνία, περιοδικό, συντάκτης)
- Φεβρουάριος 2018, IOP - Measurement Science and Technology, Charlotte O’Neale

Τιμητικές διακρίσεις και βραβεία
- Υποτροφία “Αλέξανδρος Θεοδοσίου” τμήματος Φυσικής πανεπιστημίου Πατρών (2011)
- Υποτροφία LIMMAT τμήματος Χημικών Μηχανικών πανεπιστημίου Πατρών (2017)
- Υποτροφία “Andreas Mentzelopoulos Scholarships for the University of Patras” (2018)