Mechatronics educational systems in vibration field

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Abstract. This paper presents a set of didactic applications in order to highlight the phenomena of mechanical vibration, the modal forms of the vibration, the resonance frequency and the methods of vibration damping, passive and semi-active or active. The approach is optimized for the field of mechatronics, describing the behavior of the systems using a mathematical formalism that allows integration into complex mechatronic models. Dynamic response control solutions are also developed from a mechatronic point of view, optimizing the transfer and the response functions. First example of an educational mechatronic system is a steel beam with a diameter of 6mm, with one end fixed on an aluminum plate. When an electrical signal control is applied, an electromagnet stretches a steel wire fixed between the free end of the beam and the movable part of the electromagnet. Another example of an application in the field of mechanical vibration is a metal beam fixed in several ways (fixed at one end, fixed at both ends and with hinged fastening, on a fixed support). The beam oscillates due to a couple of actuators (each connected to a signal generator and to an amplifier), one used to produce a primary vibration in the beam and the other one to produce a secondary vibration that must interfere with the first one in order to reduce it. The aim of this application is to present to the students the phenomenon of mechanical vibration, the role played by the frequency of the vibration wave, the vibration modes existing for this given situation, but also to highlight the excitation of the metallic structures in the magnetic field, used especially for vibrations control applications. The stand also demonstrates the possibility of active vibration control of distributed systems.

1. Introduction

Vibration represents the oscillating motion of a mechanical system and it is characterized by the following parameters: frequency (frequencies) and amplitude. Generally, the term vibration refers either to the periodic movement of an object or structure, or to an oscillating force applied to a mechanical system. Vibrations are widespread in the environment, which is due to the operation of various existing devices and equipment, from vehicles participating in traffic to industrial machinery [1].

Vibration is most often a side effect that occurs during the operation of mechatronic systems and it is generally harmful to mechanical structures. A system that is not intended to use vibration as an input signal or which is designed without taking into account possible vibrations that may occur during the working tasks, will have a much shorter lifetime, compared to properly designed systems, or, in the worst scenario, will not be able to perform the task that it was designed for (e.g. very precise positioning systems). There are also cases where vibration can be beneficial, such as energy harvesting systems using materials that produce an electrical voltage when they are mechanically excited [2]. Another
example of a system that uses vibration is the sorting machine (sorting different pieces of different shapes) that uses the oscillating motion of a structure to differentiate the parts.

Currently, the study of vibration is of particular importance when we talk about devices with a small and light gauge, systems with high velocity and acceleration and very precise positioning, but also for the protection of the environment, especially the safety of the people.

Mechatronics is the field that encompasses most of the engineering sciences and it studies the links between the component parts of a system. The oscillation of a component included in a mechatronic system, for example an electric motor, if analyzed as an isolated part from the rest of the system components, it is not a factor that endangers the functioning. Considering the structure as a whole, the vibration generated by that component (electric motor), during operation, is likely to bring the system to the resonant frequency threshold. This is the less favorable case, when the integrity of the system is compromised by the command at its resonant frequency, a frequency at which the system optimally absorbs the external energy and there is a strong increase in the amplitude of the vibrations, significantly changing the behavior of the system. To avoid this situation, mechatronics can contribute to the vibration field by developing semi-active and active control systems. In order to highlight this ubiquitous phenomenon of vibration of mechanical structures and their different forms of variation and different methods of damping using external actuators, two experimental stands were developed for educational purposes.

2. Educational stand for vibration control of beams using tensioned cables with electromagnetic actuators

One of the methods to control mechanical vibration is to change the resonant frequency by actively “stiffening” the structure using external actuators. [3] The first educational stand is intended to highlight the modification of the mechanical impedance of a circular section beam, with a length of 1000 mm and a diameter of 6 mm, fixed at one end in an aluminum plate with a thickness of 20 mm. At the free end, the beam is rigidly fastened with 2 tensioned steel cables. One of the cables is connected to the movable flap of the electromagnetic actuator and the other cable is rigidly attached to the aluminum plate, in order to balance the bending tension induced into the beam by the first cable.

![Figure 1. The experimental stand](image1)

![Figure 2. Schematic diagram of the stand](image2)

Also, at the free end of the beam there is an accelerometer attached that monitors the changes that occur in the two cases: the impulse-type signal excitation of the beam when the electromagnet is not
actuated and the excitation with the same impulse signal when the electromagnet is powered. For a good understanding of the educational stand, figure 1 presents the physical stand and figure 2 presents the schematic representation of the stand. The elements found on the two figures are numbered as follows: 1-Aluminum plate, 2-Electromagnet, 3-Steel cable, 4-Beam, 5-Accelerometer.

2.1. Case I - inactive electromagnet
An excitation with a force impulse signal is applied using an impact hammer that acts on the free end of the bar (on the opposite side with the accelerometer (5)) or on the flap of the electromagnet. The accelerometer that is fixed at the free end of the beam is connected to an amplifier, with a 10 times voltage gain. The output of the amplifier is connected to the oscilloscope for signal processing. The free damping of the system is observed on the oscilloscope. A Hamming window filter is applied to the signal received and the Fourier transform is calculated to determine the signal spectrum, respectively the resonance frequency of the system. For the force impulse applied to the flap, the result obtained is shown in figure 3. The analysis of the signal in time allows the evaluation of the amplitude logarithmic decrease and the damping of the structure. The frequency of the resonance can be determined from the signal in time or from its spectrum. When using a two-channel acquisition system, the force signal from the impulse hammer can be also acquired to determine the response function of the system.

![Figure 3](image_url)

**Figure 3.** System response to impulse-type excitation with an inactive electromagnet

On the graph at the bottom of figure 3 we observe the acquired, unfiltered signal (pale green) and the filtered signal (bright green) which represents the free damping vibration of the system after the impulse type excitation. The Fourier Transform is calculated for the acquired signal resulting the graph at the top of figure 3. After processing, the maximum amplitude of the signal is obtained at the frequency of 9.76Hz which can be seen on the graph. This value represents the natural frequency of the fixed beam tensioned with the two cables, without the action of the electromagnet.

2.2 Case II - active electromagnet
When the electromagnet receives a 12-volt control signal from the source, it develops a magnetic field that attracts the flap that the steel cable is attached to. The stroke of the movable element is about 2 mm. After supplying the electromagnet, an impulse excitation signal is applied to the system (force impulse using an impact hammer acting on the flap) and the response signal is acquired and processed in the same way as in case I. The resulting graphs are shown in the figure 4.
A change in the resonance frequency of the system from 9.67Hz to 16.9Hz is observed on the graph at the top of figure 4 meaning that the system becomes more rigid. Another parameter that changes when the electromagnet is active is the internal damping of the system. It can be seen, on the graph at the bottom of figure 4, that the system response has less oscillations than in case I which means that it has a higher damping factor. This phenomenon occurs due to the magnetic field that pulls the metallic flap connected to the steel cable. The energy of the vibration wave is dissipated more in the magnetic field and less through the mechanical structure.

2.3 Conclusions
For mechanical structures with tensioned cables, it is possible to translate the resonance frequency of the system by introducing an external actuator that receives a control signal when the system approaches the resonance. If the external actuator introduced into the system is an electromagnet, when it is supplied, the generated magnetic field influences both the rigidity of the system, by introducing an additional tension into the cable, and its damping. The mechanical oscillations are dissipated faster due to the magnetic field.

3. Educational stand for understanding modal shapes of metallic beam structures
The stand presented in this chapter aims to highlight the phenomenon of mechanical vibration, more precisely, to highlight the number of natural frequencies and modes of vibration (modal shapes) for a structure with a fixed metallic beam (fixed at one end or at both ends). The metallic beam chosen for the educational stand is a structure with distributed internal parameters (mass, damping and elasticity) which means that, in theory, it has an infinite number of natural frequencies.

However, it will vibrate, with remarkable amplitude, only at a finite number of frequencies because the energy required for any deformations of the beam at higher natural frequencies will exceed the energy generated by the source. It is important to mention that when the beam vibrates in a single vibration mode, each structural element of the beam executes a simple harmonic oscillation movement around the equilibrium position, except for the supports. [1] For the existing stand, the first 5 modes of vibration of the natural frequencies were determined for the cantilever beam case. For the case where the beam is constrained at both ends there were determined only the first 4 natural frequencies.

An important aspect of vibration is represented by the lines, points or nodal surfaces, which locate the region that remains motionless to a certain mode of vibration. Only one case of vibration makes an exception, the first natural frequency, the fundamental mode, which does not have any nodal points, except the points where the beam is fixed. The positions of the nodal points vary according to the type
of fixing of the beam structure. These points represent important areas for the engineers who develop the design of the mechanical structures because vibration sensitive equipment will be placed there.[4]

3.1. Cantilever beam

The educational stand made to highlight all these behaviors mentioned above (natural frequencies, vibration modes and nodal points) is presented in figure 5. The excitation of the mechanical structure is realized using an electromagnet (2) connected to an amplifier that receives a sinusoidal signal from the signal generator (1). The amplifier is powered from the bank source (5) with 12V. The metal beam (3) (15 x 1000 x 1.5 mm) is fixed at one end on the rigid support (4). The electromagnet is located vertically at a distance of 100 mm from the fixed end of the beam.

It can be observed on the two figures (5.A and 5.B) that there is a difference in shape between the two modes of vibration of the beam; the modal form at the third natural frequency in figure 5.A and the modal form at the second natural frequency in figure 5.B. Visualizing the modal forms of the mechanical structures on experimental stands is a very efficient way, especially for students, to observe this behavior and understand the effects that the vibration has over the structure. The first 5 modal shapes of the beam with the corresponding resonance frequencies were determined experimentally (figure 6, extracted from photos of movements). Due to the fact that the mechanical excitation on the structure is applied with a single electromagnet, the force applied to the beam will be exerted in only one direction represented by the attraction of the beam. Figure 6 shows the first 5 modes of vibration marked with letters from A to E, respectively the nodal points which are marked with red dots. The natural frequencies of the beam are presented in table 1 for each modal shape of vibration.

| Modal shape | A             | B             | C             | D             | E             |
|-------------|---------------|---------------|---------------|---------------|---------------|
| Resonance frequency | f_0=1,2 Hz | f_1=5,6 Hz | f_2=16,2 Hz | f_3=31,6 Hz | f_4=46,8 Hz |

Table 1. The resonance frequencies of the metal beam with one fixed end
3.2. Fixed beam at both ends

The experimental stand also allows the beam to be fully constrained at both ends to highlight the modifications of the modal shapes and their frequencies depending on the rigidity of the system. If the rigidity of the system increases, the resonance frequencies of the structure have higher values. Figure 7 shows schematically the modal shapes of the new structure extracted from photos of movements, marked with letters from A to D, and table 2 shows their resonance frequencies.

![Modal shapes of a fixed metal beam at both ends](image)

**Table 2. Resonance frequencies of a beam fixed at both ends**

| Modal shape | A              | B              | C              | D              |
|-------------|----------------|----------------|----------------|----------------|
| Resonance frequency | $f_0=3.2 \text{ Hz}$ | $f_1=16.4 \text{ Hz}$ | $f_2=37.4 \text{ Hz}$ | $f_3=65.8 \text{ Hz}$ |
By analyzing both tables with frequencies (table 1 and table 2) it can be observed that there is an increase of the fundamental frequency, which is justified by the rigid grip on both ends. By analyzing both pictures, a decrease of the oscillations amplitude from one vibration mode to another can be observed, due to the higher energy consumption necessary to deform the metallic structure (the input energy generated by the external source is considered constant and it is entirely distributed to the structure only as bending energy in the position of maximum amplitude of the deformation; the energy deformation increases with the number of nodes that appear on the beam).

3.3. Modal analysis of a fixed beam using an accelerometer
There are cases where the modal shapes of a structure are not well defined or not perceptible to the human eye. In these conditions, an analysis of the dynamic vibrational behavior of the mechanical systems is necessary to determine the areas with high amplitude of deformation at certain frequencies. This is achieved by raising the amplitude-frequency characteristic of the system. This can be realized by placing transducers on the structure and processing the acquired signals. The current tendency is to use active materials, which can be used both as sensors and as actuators [5], for the selective control of vibration and not only for measurements. In addition, when introducing a controlled and measured force, a complete modal analysis is possible, which allows not only the estimation of the modal shapes and frequencies but also the subsequent evaluation of the behavior under a given external excitation, based on the experimentally determined models of the structure behavior under known external force conditions, at any point of the structure. [6][7]

The educational stand presented in subchapters 3.2 and 3.3 can be modified in such a way that an electrodynamic actuator can be installed, which will introduce a harmonic force into the system, at a certain frequency. The electrodynamic actuating force is measured with a force transducer mounted exactly where the excitation occurs. An accelerometer will be attached to the metal beam in various areas to measure acceleration. The correlation of the two signals, the input signal of the force transducer and the output signal of the accelerometer allow us to raise the response function characteristic (amplitude-frequency characteristic and phase-frequency characteristic) of the given structure.

![Figure 8. Educational stand for frequency analysis of fixed beams](image)

The educational stand developed to perform modal analysis is shown in figure 8, where the electrodynamic actuator (1) generates a sinusoidal oscillation of the structure according to the command received from the signal generator (5). The actuator (1) transmits the vibration to the beam (4) through a steel wire caught between two metal collets, which allows the transmission of the excitation even at
small rotations of the beam during the vibration. The grip of second collet on the beam, where the force transducer (2) is located, is realized using screw-nut assembly. When the beam starts to vibrate, the accelerometer (3) will generate an output signal depending on the vibration mode of the beam at the frequency at which the excitation is performed. The accelerometer is placed in several points to raise the modal shape curve at the respective frequency.

3.4. Modal analysis of a fixed-ends beam with flexible piezo films sensors
Flexible piezo films sensors (figure 9) can be used to raise the frequency response of mechanical structures. These are low cost sensors (compared with the classical accelerometers which are more expensive) and measure the deformation of the material in dynamic regime.[5] In addition, they can also be used as actuators. In order to reduce the time required for dynamic vibration analysis, a large number of sensors is mounted on the beam and thus the deformation of the structure is measured simultaneously at several points.

![Figure 9. Flexible piezo films sensor](image)
![Figure 10. Sensor assembly](image)

Figure 10 presents the placement of one sensor on the beam. It is attached to the beam, and it is connected to the electronic amplification board. Its location was determined experimentally because it must be positioned in the area of maximum bending. The disadvantage of using these types of flexible piezo sensors is the low signal-to-noise ratio. The signal generated by the sensor has a very low voltage and requires a consistent amplification, which inevitably amplifies the noise. Flexible piezo films sensors can also be used to close the feedback control loop in the case of active control systems.

Using only the signals from sensors, without measuring the excitation force, allows the identification of modal shapes and natural frequencies but will not be enough to raise a mathematical model which will permit to foretell the response of the system under a given excitation. This type of analysis is called Operational Deflection Shape Method and it can be used for vibration diagnostics of the structure. [4][5]

3.5. Modal analysis of a double hinged beam using piezoceramic sensors
The experimental stand presented in figure 11 represents another approach for modal analysis and raising frequency response of the various metal structures. Figure 11 presents the distribution of sensors S1, S2, S3, S4, S5 and S6 on the aluminum beam that is hinged in points A and B. The sensors used to measure the deformation at the points marked in the figure are made of piezoceramic material. This material has a more rigid structure than the flexible piezo sensors and generates a stronger signal when deformed, which means a higher accuracy of the acquired signal and an electronic circuit with a smaller amplification factor than in the flexible piezo film sensors.

To vibrate the mechanical structure of the system, two piezoceramic actuators were placed in the middle of the hinged beam. When receiving a control signal, the beam vibrates with a given frequency and the sensors measure the deformations at the given points. Thus, the frequency response of the system can be determined.
3.6. Active vibration control for a fixed beam structure

Active vibration control can be achieved by several methods with variations of sensory components (accelerometers or piezo sensors) and actuators (electromagnets, electrodynamic actuator or piezo actuators). [8][9]

Regarding the vibration damping, the electromagnetic actuator, that can perform an active damping of the vibration by introducing a force signal of the same frequency with the vibration, but with an optimal phase shift, can be added to the educational stand presented in figure 8. Possible structural ways of an active vibration control system are shown in figure 12. The fixed-ends metallic beam is excited by a sinusoidal signal through the electrodynamic actuator (Ed). In order to measure the response of the beam, either an accelerometer (Acc) or a piezo (Sp) sensor can be used.

The acquired signal from the sensor passes through a signal amplifier and it is transmitted further to a digital signal processor (DSP). Here, the feedback control program is loaded, which analyzes the input signal and generates a signal that must dampen the vibration of the beam. This control signal can be sent either to an electromagnet or to a voltage amplifier and piezoceramic actuator. The actuator will generate a counter-excitation in the metal structure with the role of damping the vibration and its effects.[10][11]

The chosen configuration for the experimental stand is shown in figure 13. An electrodynamic actuator has been used to generate a vibration of the structure at a given frequency. The accelerometer was placed at the maximum amplitude vibration point for a strong and precise signal that represents the feedback for the active control program. A MYRIO-1900 controller from National Instruments has been
used for a digital signal analysis and signal generation. It is capable of processing the signal received from the accelerometer in real time and performing the system correction. It has two field-programmable gate arrays (FPGAs) that can be programmed to perform signal generation tasks in parallel with the main processor. The vibration correction of the system is made using an electromagnet, connected to a power amplifier and powered at 12V.

Figure 14. The signal from the accelerometer without active damping control

The accelerometer signal was acquired in parallel by the Fluke bench oscilloscope. In the first case, the system reacts only under the vibration of the electrodynamic actuator at a frequency of 25.3 Hz. A filter was applied to the acquired signal to eliminate any noise. Figure 14 presents the signal from the accelerometer on Channel 2, without active damping control. The peak-to-peak amplitude of the signal is about 800mV. It is desired that when the active vibration control system is active, the amplitude of the signal will decrease meaning a damping of the vibration wave in the beam.

Figure 15. The signal from the accelerometer using the active damping control

Due to the fact that the electromagnetic actuator can influence the beam only in one direction (it creates an electromagnetic field that attracts the beam), the signal correction should take action only when the beam goes away from the actuator. This means that the waveform of the control signal should
be a square shape signal (a PWM signal with low pulse width). The MYRIO controller should adjust three parameters during the active damping control: amplitude of the correction signal, pulse width and phase shift. Amplitude and width of the pulse control the electromagnetic force. A certain signal amplitude will maintain the system stable and a certain pulse width should lower as much as possible the amplitude of vibration. The phase shift depends on the location of the components (actuator and sensor) on the system.

The signal acquired after activating the damping control system (Channel 2) and the signal generated with the controller (Channel 1) are shown in figure 15. The results show a decrease in the vibration amplitude, especially under the pulse. After the active control started, the peak-to-peak amplitude of the signal went down to approximately 500 mV, which means a damping effect of 37.5%, using only one electromagnetic actuator. The damping effect is limited by the electromagnetic force direction generated by the actuator and, also, by the non-linear characteristics of this actuator type. Currently, piezoelectric actuators are tested for active damping control.

4. Conclusions
The developed educational stands facilitate the highlight of multiple aspects of the vibrations field, such as visualization of the modal shapes and the nodal points, the possibility of using several types of actuators for desired oscillating wave generation (electrodynamic, electromagnetic, piezoceramic) and, also, the possibility of using different measurement solutions for vibration (accelerometers, piezoceramic or flexible piezo films sensors). By combining this knowledge and skills to develop a vibration measurement system, students can move on to the next challenge, respectively, applying modern methods in structure analysis such as Modal Analysis and Operational Deflection Shape analysis. The students learn that they can use both traditional method and updated method, based on piezo sensors and active materials, for applications in vibration monitoring and predictive maintenance. Finally, adding the knowledge from microcontroller programming courses and from automatic control systems, students can configure semi-active and active vibration control applications. The semi-active control is applied by using the conversion of mechanical energy into electrical energy using piezoelectric materials connecting them to circuits with controllable impedances (efficient, robust and stable but poorly adaptive solution). The active vibration control is also based on the use of mechanical energy conversion into electrical energy using piezoelectric materials, using the electrical signal as feedback for the close loop control. Vibration correction for active damping control is performed with different actuators as piezoelectric, electromagnetic or electrodynamic. The students learn the importance of the place where the sensor and the actuator are situated, in terms of optimal phase shift between signals and optimal parameters for the correction loop.

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