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IMPLEMENTATION OF AN ACTIVE MASS DAMPER TO CONTROL VIBRATIONS IN A "LIVELY" FOOTBRIDGE

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Abstract. *This paper describes the work involving a real implementation of an active control system to reduce vibrations in a "lively" footbridge located at FEUP. This structure has some natural frequencies around 2Hz which can be easily excited by some pedestrian activities. Although the levels of vibration may not be considered excessive according to some design codes, this footbridge experiments regularly unusual vibrations for this type of structure, which has motivated the implementation of a control system for research purposes. The active control developed is composed by an Active Mass Damper commanded by a controller based on the Velocity Feedback Control law. The efficiency of this control system is evaluated in terms of the reduction of the response when the structure is excited by several pedestrian loads.*

1 INTRODUCTION

Pedestrian bridges are Civil Engineering structures which may be vulnerable to unacceptable levels of vibration motivated by the proximity of the step frequency of pedestrians with regard to some natural frequencies of the structure. This phenomenon can be more frequently observed in footbridges with natural frequencies around 2Hz, for vertical vibrations, and 1Hz for lateral vibrations.

In order to reduce vibrations in this type of structures, several kinds of control systems have been proposed and implemented [1,2], consisting generally in passive systems composed by viscous dampers or Tuned Mass Dampers (TMDs). This type of devices has the objective of adding damping into the system to reduce its dynamic response. This constitutes a good control strategy because, in case of the occurrence of resonance problems, the amplitude of the response is strongly influenced by damping ratios of the system.

However passive control systems like those described before, can have some problems which can compromise the use of such devices. For example, viscous dampers are effectively useful in cases where it is possible to connect different points of the structure characterized by having a significant relative modal displacement for the problematic vibration modes. Even when it is physically possible to implement such elements, in most of the practical cases there are significant architectural restrictions which may limit this solution. TMDs are also vulnerable to some problems, particularly frequency tuning. It is well studied that relatively small frequency variations of the structure may lead to significant reduction of efficiency. Moreover, TMDs are designed to control a specific vibration mode, which means that there will be needed as many TMDs as many vibration modes to be controlled. In most of the cases, it is adopted several TMDs tuned to each specific vibration mode, which results in a relatively high number of units.

On the other hand, it is generally accepted that active control is not an interesting solution for many structural problems, particularly when dealing with large structures. The reason of this is because active control demands more sophisticated technology, higher costs, less robustness and more energy consumption [3]. This is why there are only few real applications of this type of control to Civil Engineering structures. However, as an alternative solution to the passive systems described before, active control can be an attractive solution for small structures like some footbridges. In this case, active systems have some pertinent advantages which should be seriously considered. For instance, active systems using AMDs don’t need to be tuned to any natural frequency of the structure because they work with the measured response of the system. Also AMDs (like TMDs) can be placed in the most suitable positions of the deck in order to actuate on sections characterized by having significant modal components of the critical vibration modes. Combining these advantages it can be concluded that a single device can control several vibrations modes simultaneously. For small structures that require low control forces there is also the possibility of using electrical actuators which may have some advantages in terms of cost, maintenance and noise reduction. Active control systems can be also robust and adaptive to the variations of the dynamic parameters of the structure and can be more effective in controlling small vibrations.

In this context, the aim of this work is to implement an active control system to reduce vibrations in a “lively” stress-ribbon footbridge located at the campus of FEUP. This experience is the result of some research and laboratorial work done involving the implementation of control systems in small physical models [4,5] developed at the Laboratory of Vibrations and Monitoring (www.fe.up.pt/vibest). This paper describes extensively the work developed which includes the analytical study used to design the control system as well as the experimental results obtained.

2 CHARACTERIZATION OF THE FOOTBRIDGE

2.1 General description

The footbridge under analysis provides a pedestrian link between the main buildings of FEUP and the student's canteen and parking areas (Figure 1). This structure was designed by ENCIL [6] and is characterized as a stress-ribbon footbridge formed by two spans of 28m and 30m. The deck is composed by four prestressing cables embedded in a reinforced concrete and it takes a catenary shape over the two spans with a circular curve over the intermediate support. The deck is a rectangular cross-section of external design dimensions of 3.80m×0.15m.

Figure 1. View of the footbridge

2.2 Identification of the modal parameters

The identification of the modal parameters that characterize the dynamic behavior of the footbridge, as well as the corresponding numerical assessment, was extensively studied and developed by Caetano and Cunha [7], recurring to several identification techniques based on ambient and forced vibration tests.

The ambient vibration tests were performed with the help of four seismographs including force-balance accelerometers, duly synchronized by a laptop. The ambient response of the system was recorded according to several setups, using two fixed reference stations located at sections 3 and 18, and eighteen other measurement stations distributed by the total length of the deck (Figure 2).

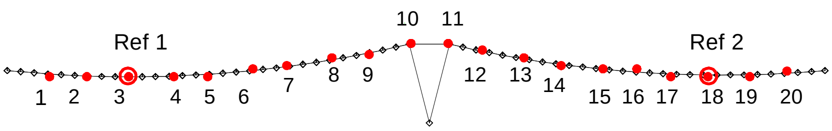

Figure 2. Measurement stations and references

For each pair of measurement points, acceleration-time series with 6 minutes of duration were acquired and subsequently processed in order to obtain estimates of power spectral density functions (PSD) and frequency response functions (FRF) relating the response at each section to the reference station. Figure 3 shows two average power spectral density estimates obtained at the two reference sections, from which the main natural frequencies can be identified.

Using the conventional peak peaking method applied to the set of FRFs, it was possible to identify the natural frequencies and the corresponding vibration mode shapes, some of which are represented in Figure 4. Table 1 lists the identified natural frequencies, as well as the corresponding mode type.

The numerical model developed to represent the dynamic characteristics of the system was achieved by considering the experimental results obtained and by taking into account the different phases of construction process [7]. Figure 4 and Table 1 shows a comparison between analytical and experimental results in terms of natural frequencies and modal shapes.

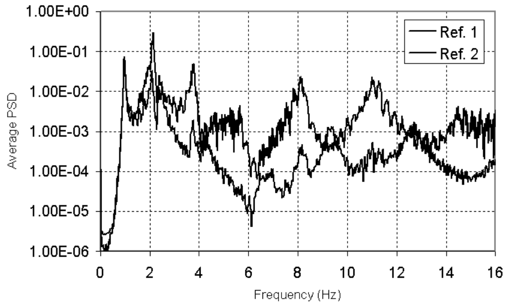

Figure 3. Average normalized power spectral estimates at the two reference sections

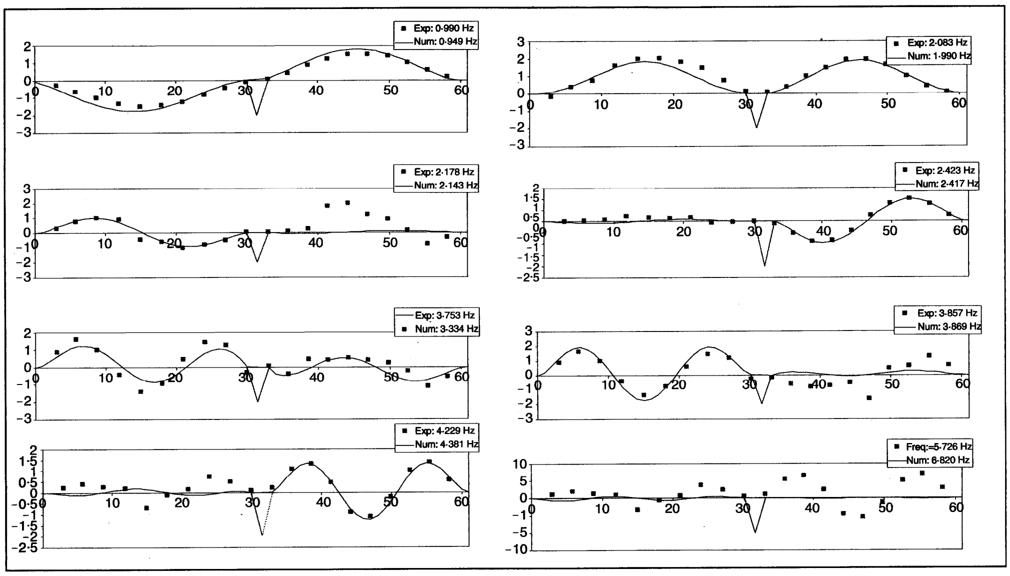

Figure 4. Identified and calculated vibration modes

Order	Measured frequency (Hz)	Calculated frequency (Hz)	Type of mode
1	0.990	0.949	First antisymmetric (two spans, opposite phase)
2	2.083	1.990	First symmetric (two spans, in-phase)
3	2.178	2.143	Second antisymmetric (L=30m)
4	2.423	2.417	Second antisymmetric (L=28m)
5	3.753	3.334	Second symmetric (two spans, opposite phase)
6	3.857	3.869	Second symmetric (L=30m)
7	4.229	4.381	Second symmetric (L=28m)
8	5.726	5.915	Third antisymmetric (L=30m)
9	6.517	6.820	Third antisymmetric (L=28m)
10	8.262	8.271	Fourth symmetric (two spans, opposite phase)

Table 1. Identified and calculated natural frequencies

Damping properties of the footbridge were also evaluated adopting several identification techniques using either ambient or forced vibration tests [8]. An expedite method adopted to estimate the damping factors associated to the first vibration modes consists in exciting the structure with a frequency closed to a natural frequency using a pedestrian skipping at a fixed position. After achieving a resonant response, the excitation stops suddenly and the free motion of the structure is recorded. By analysing the free decay curve, it is possible to estimate the respective damping coefficient using the logarithmic decrement method. Figure 5 shows the results obtained by using the excitation frequencies about 1Hz and 2Hz. It is possible to identify a damping factor about of 1.7% for the 1st vibration mode and 2.6% for the 2nd one. These estimated damping factors are averaged values because, as it is well known, damping in real structures varies with the amplitude of the oscillations.

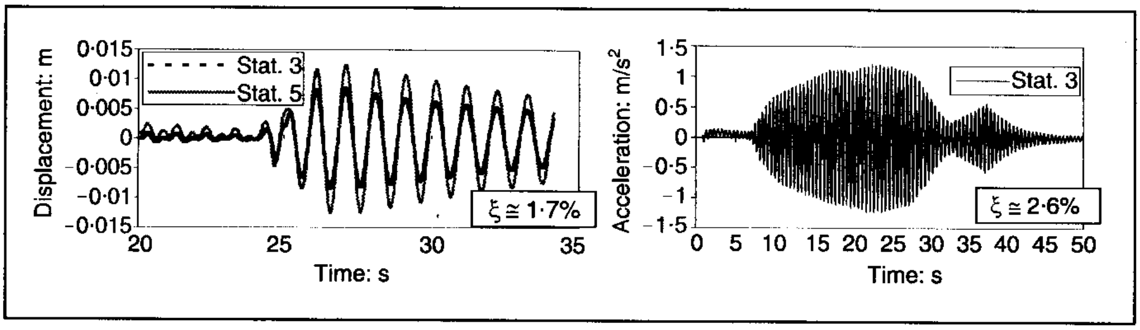

Figure 5. Free vibration response: a) skipping at 1Hz, stations 3 and 5; b) skipping at 2Hz, station 3

3 STUDY OF THE CONTROL SYSTEM

3.1 Number and position of sensors and actuators

The number and position of sensors and actuators adopted to control a dynamic system are naturally associated to an evaluation of the objectives of the control system and the cost correspondent to a specific solution. Furthermore the number and position of actuators is directly related to the concept of controllability and, in the case of the sensors, to the system observability. As a first approach on the choice of a control system to reduce vibrations in the footbridge described before, it can be clearly observed that the structure has local vibration modes which suggests that the global control of the system is possible with the help of several actuators. In fact if only one actuator is adopted to operate in any of the spans of the footbridge, it is inevitable to install the device in a section where some vibration modes have reduced modal components, which would demand a strong control force to damp these modes. On the other hand, it can also be concluded that the adoption of two devices positioned at each span doesn't mean that the system is completely controllable, because it is difficult to select a section where all vibrations modes have important modal contributions.

To consider this problem, it is convenient to plot in a single graph the significant vibration modes of the system. In the case of the footbridge studied in this work, this kind of graph is presented in Figure 6, where the numerical modal configurations of the first five vibration modes are shown. It can be assumed that the 1st vertical vibration mode with a frequency of 0.95Hz is not critical in terms of the human-induced vibrations because it is out of a critical frequency range which is characterized by frequencies that can be easily excited by regular vertical loads of pedestrians. For the same reason, vibration modes with frequencies above 3.33Hz are not here represented.

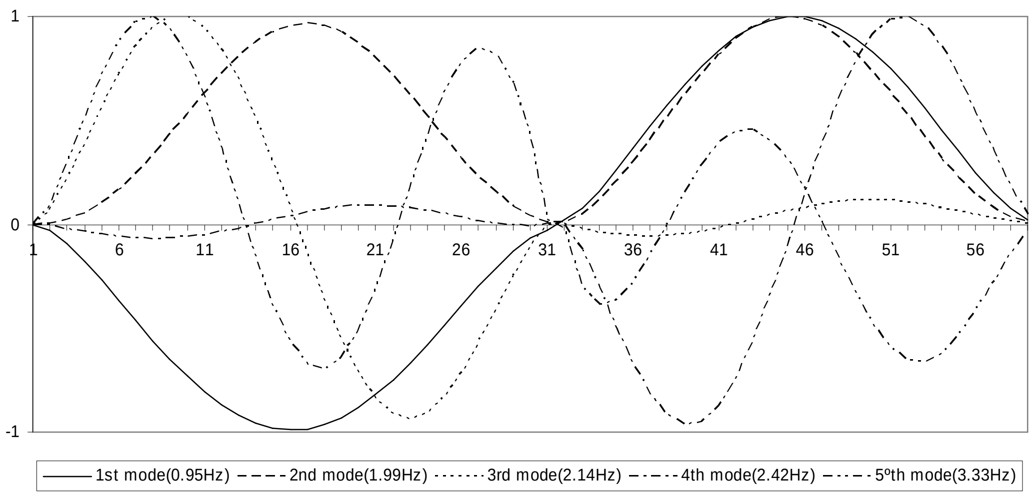

Figure 6. Representation of the first five vibration modes of the system

The analysis of Figure 6 allows the establishment of some scenarios in terms of the positioning of the actuators. At this stage, it can be assumed that the control of the 2nd, 3rd ant 4th vibration modes is a priority when compared to the control of the 1st and 5th modes, for the reasons exposed before. On this basis, it seems reasonable to use one actuator near section 13 of the first span, suitable to the control of the 2nd and 3rd modes, and another actuator near section 49 at the second span, reinforcing the control of the 2nd mode and controlling of the 4th mode.

As far as the number and position of sensors are concerned it must be paid particular attention to the control strategy and the number of modes to control. In fact, if control strategies which require a state space approach are adopted (like optimal control or predictive control), it is desirable the use of a large number of sensors because it benefits the system observability and, if needed, the design of an observer. On the other hand, in case of adopting more simple strategies, like decentralized control schemes, the number of sensors can be reduced.

3.2 Control strategy

The response of a dynamic system subject to periodic loads under resonance conditions is strongly dependent of the damping factor of the excited vibration mode. This means that if the system is vulnerable to the occurrence of resonance phenomena, like the footbridge described previously, the increase of damping is certainly a good strategy. This will reduce the structural response and will keep the vibration levels below some maximum acceptable limits. To estimate the amount of damping needed to reduce the amplitude of the harmonic response of a linear system to a predefined value, it is sufficient to use the coefficient $1/2\zeta$, which represents the dynamic amplification of a system in the described conditions, where ζ is the damping factor of the vibrating mode.

The increasing of damping in a system can be achieved by a control force calculated by $F_C(t) = -cv(t)$, where $v(t)$ is the velocity at the position of the control force and c is a gain correspondent to a determinate damping constant. This control law is the well-known Direct Velocity Feedback (DVF) or a derivative controller. Its effect can be compared to the introduction of a passive viscous damper in the structure because both produce a force proportional to the velocity.

In the case of the studied footbridge, the application of active forces at the nodes 13 and 49, calculated based on the DVF, is equivalent to having two passive dampers attached from these sections to some fixed points, as shown in Figure 7. It is worth noting that if it was possible to adapt these passive devices on the structure as described, such solution would be much more interesting than the application of active forces. The problem is that, as it happens in many practical cases, such solution would interfere dramatically in the architecture of the footbridge. This is the reason why active control could be an attractive solution for this case, particularly when combined with AMDs.

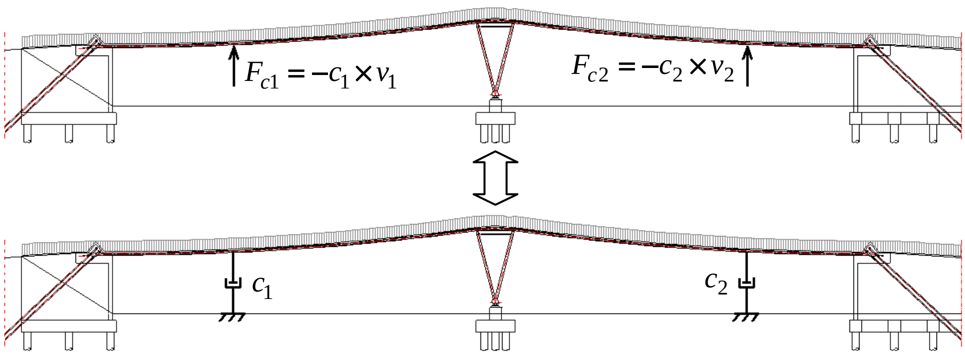

Figure 7. Control effect using Direct Velocity Feedback

The use of AMDs allows the application of concentrated forces at any section of the deck without having any external connections. This solution is often mentioned in the literature as “sky-hook dampers”, because it seems that the dampers are connected from the structure to a fixed point in space.

However the control strategy described based on DVF must be adapted to this kind of devices because the control forces are formed by a pair of forces applied between the system and the active mass instead of isolated forces applied to the structure. This means that the interaction effect between the structure and the actuators should be considered and analysed by some design technique.

4 ASSESSMENT OF EFFICIENCY USING AN AMD EXISTING AT FEUP

4.1 Description of the control system

The control system described before has the objective of reducing vibrations in the structure when excited by several pedestrian activities. The control scheme described to achieve this goal demands the use of two actuators positioned at each span of the footbridge. However only one actuator was available to perform real tests which constituted a limitation in terms of control of the global system. Nevertheless, even when using one device, it is possible to implement a local control system based on the same principles referred in the last section, allowing reducing vibrations in one selected span of the structure. The analytical study of the expected efficiency to obtain with such a scheme is exposed in next sections.

It is assumed that the control system is composed by an AMD which can be adapted in a certain section of the deck, allowing the application of control forces computed by using DVF control law. This means that the response of the system must be measured locally, in the same position of the actuator. Despite the similar positioning of the pair sensor/actuator, this scheme doesn’t constitute a collocated system, because there is a component of the control force applied to the additional mass of the actuator which origins some stability problems that must be properly considered.

4.2 Characterization of the actuator

The identification of the characteristics of the AMD is crucial to evaluate the performance of the proposed control system. The device existing at FEUP is composed by an electromagnetic shaker "Electro-seis Model 400" manufactured by APS Dynamics, which can be modeled as a linear system like the indicated in Figure 8. The dynamic parameters of the actuator, in terms of natural frequency and damping factor, were identified experimentally by introducing a motion according to the resonant frequency of the system and by analysing its free decay response after stopping the excitation. The natural frequency could be measured directly counting the number of cycles occurred in a finite time interval and the damping factor could be extracted by the logarithmic decrement method. Figure 8 plots the free response of the actuator obtained as described, allowing estimating a vibration frequency of 1.33Hz and a damping factor of 7.1%. The total active mass was evaluated as 34kg.

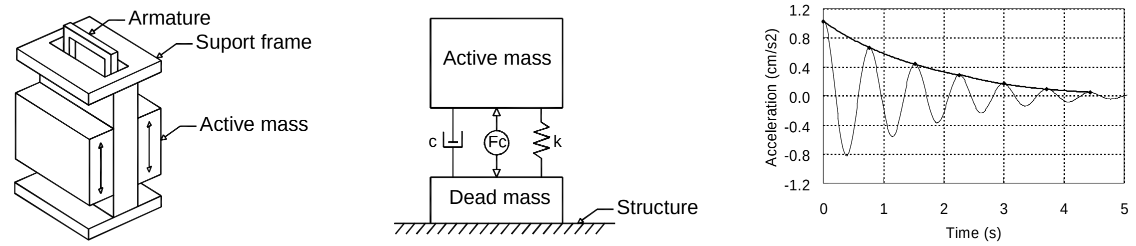

Figure 8. Electromagnetic shaker and respective dynamic model

It is well established that AMDs should have a natural frequency below the first frequency of the structure, in order to avoid pole-zero flipping associated to the intermediate vibration modes, and it should have high damping with the objective to get a large gain margin to increase stability [9]. The device used in this experience has a natural frequency superior to the first system frequency, which means that the control system will not be able to control this vibration mode, as discussed in next section. In addition, the ratio of the active mass to the system mass is relatively small (<2‰) which may limit efficiency due to the maximum stroke of the actuator. In fact, the amplitude of motion of the active mass is related to the corresponding mass ratio. In order to dissipate the same amount of energy, a large moving mass will require a shorter stroke than a shorter mass.

4.3 Root-locus design

The method used to design the control system is based on the classic root-locus analysis. This is because the system may be considered linear in the range of displacements usually observed, and the control action is based on one output of the system, corresponding to the location of the pair sensor/actuator. It was considered the positioning of the actuator alternatively in several relevant sections of the deck. As discussed before, there is an interest in locating the actuator in section 13 to control the 2nd and 3rd modes and in section 49 to control the 2nd and 4th modes. It was also analysed the positioning of the actuator in section 9, suitable to control the 3rd mode, in section 52, suitable to control the 4th mode, and in sections 17 and 45, suitable to control the 2nd mode.

The transfer functions, which relate the system response at each of these sections and the pair of forces applied by the actuator, were calculated using the numerical model developed. It was considered a modal analysis approach using the superposition of the first five vibration modes. The root-locus analysis was performed using the control system toolbox of Matlab software. Figure 9 represents the root-locus plots for three selected cases corresponding to the positioning of the actuator at sections 13, 49 and 52. Figure 10 indicates the evolution of the modal damping factors associated to different levels of gain, as well as the maximum allowable gain. Notice that it was adopted equal initial damping ratios of 1% for all vibration modes, which may be considered a conservative design procedure. In fact, previous studies suggest some variability of this parameter, which indicates estimates superior to this value no matter the method used for its evaluation [8].

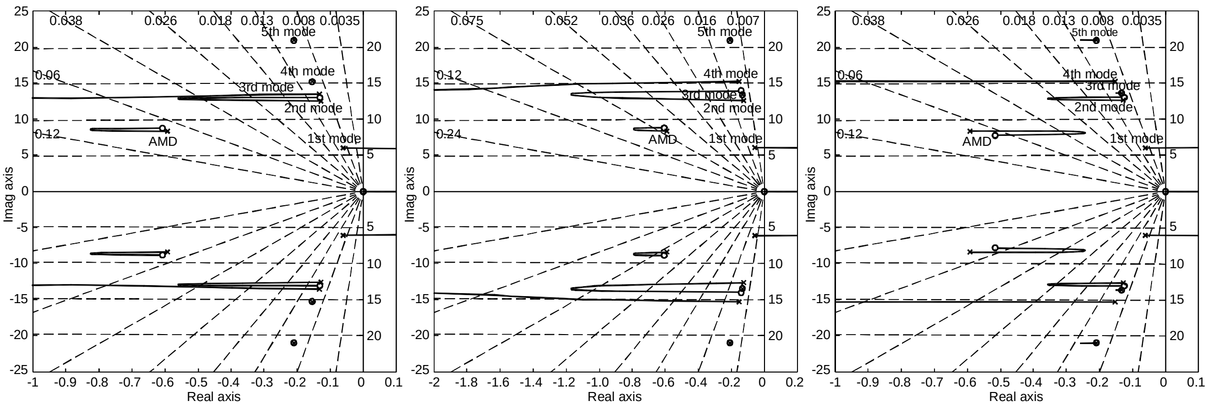

Figure 9. Root-locus plots corresponding to the actuator positioned at sections 13, 49 and 52, respectively

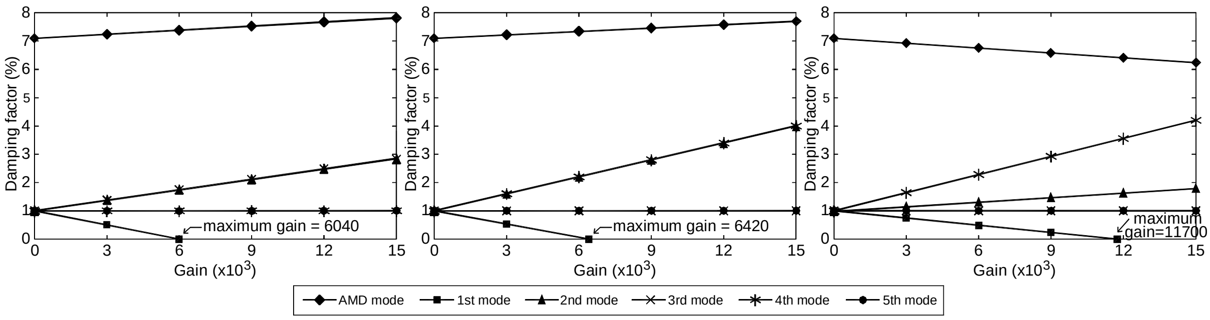

Figure 10. Evaluation of the damping factors and maximum gains

The analysis of the previous figures leads to some interesting conclusions about the performance of the control system. The first relevant issue is about stability which is certainly the most important quality in an active system. Looking to any of the represented root-locus plots, it can be observed that instability may occur by the uncontrolled vibration of the 1^{st} mode of the structure. This fact was clearly assumed before due to the relatively high frequency of the actuator which originates a pole-zero flipping associated to the intermediate vibration modes of the system. This means that as the gain increases, the damping ratio of the 1^{st} mode of the structure decreases, which leads to the system instability for gains superior to the maximum gain admissible. This constitutes a limitation of this actuation system which can be resolved by limiting the maximum control gain. Furthermore, it was mentioned before that the 1^{st} vibration mode is not critical in terms of its vulnerability to the excitation induced by pedestrian loads, which means that decreasing its damping ratio will not affect significantly the global performance of the control system.

It can also be concluded that the positioning of the actuator in any of the mentioned sections will benefit the system behaviour in terms of the damping of some specific vibration modes. This is the case of the actuator located at section 13 which allows the increasing of the damping factor of the 2^{nd} mode from 1% to 1.76% and the damping factor of the 3^{rd} mode from 1% to 1.73%, when using the maximum gain value. It can be noted through the observation of the respective root-locus diagram that, in this case, the control system is inefficient to actuate upon the 4^{th} and 5^{th} modes because, as shown in Figure 6, the actuator is near their respective nodes. When considering the actuator positioned at section 49, the damping factor of the 2^{nd} mode increases from 1% to 1.85% and the damping factor of 4^{th} mode from 1% to 1.84%, for the maximum gain value. The actuator can not control the 3^{rd} and 5^{th} modes, for the same reason pointed in the previous situation.

These results are not very encouraging because, even when using the maximum control gain, it is expected a relatively low efficiency of the control system. However, when analysing the positioning of the actuator at other different sections of the deck, it can be observed an interesting performance by trying to control the structure at section 52. This happens because, at this location, the actuator concentrates its effort to damp mainly one vibration mode instead of two as before. In addition, the 1^{st} vibration mode has a reduced modal component which increases system stability by doubling the allowed gain estimated in previous cases. Observing the respective root-locus plot, it can be expected an increase of the damping factor of the 4^{th} vibration mode from 1% to 3.49%, adopting the maximum gain. This result corresponds to a nice performance of the actuation system located at this section which has motivated the real implementation of this control scheme. Notice that this performance is achieved by using the maximum gain which can be limited by other physical conditions as the maximum stroke available at the actuator.

5 IMPLEMENTATION OF THE CONTROL SYSTEM

5.1 Description of equipments and instrumentation

In order to verify experimentally the efficiency of the described control system, it was implemented a real control experience using the equipments and instrumentation available at Laboratory of Vibrations and Monitoring. The control system was basically composed by an actuator commanded by a controller which calculates the control force based on the structure response measured by sensors.

As shown in Figure 11a, the actuation system used in the tests is formed by the electromagnetic shaker described in section 4.2, which is capable of applying control forces of 445N of amplitude. This actuator was supported in a metallic frame which is responsible to transport the device and to transmit forces to the structure by means of a tripod system. Interaction forces between the actuator and the structure were measured by three load cells aligned with the three foots of the device, as shown in Figure 11b. The actuator was also endowed with an LVDT to measure displacements of the active mass relatively to the structure, with the objective to avoid shocks with the boundaries of the equipment

The footbridge was instrumented with two piezoelectric accelerometers with the objective to obtain the system response at the relevant sections. One of these sensors was positioned in the same location as the actuation system in order to feedback the control system by measuring the correspondent velocity obtained by integrating the signal from the accelerometer. The other sensor was positioned alternately at the sections where the maximum system response was expected which correspond to the location of the maximum modal component of the excited vibration mode (Figure 12a).

The control algorithm was implemented in a Real-Time controller composed by a computer NI PXI, which includes a specific operating system dedicated to ensure determinism in control loops. This controller works as a target computer commanded by a host computer used to develop software and visualizing data from the Real-Time machine (Figure 12b). All software created to operate with this system was developed with Labview package software from National Instruments. When the controller is functioning, it executes several tasks according to some defined levels of priority. Naturally, tasks like calculating the control force and tasks involving security procedures are of priority comparing to tasks like saving or visualizing data.

An important issue about the implementation of active systems is the development of certain verifications to check the correct functioning of the control system. In this particular application, it was given special attention to the possibility of the existence of shocks between the active mass and the frontiers of the device. If the displacement of the active mass exceeds some predefined maximum value (which may occur in the case of the system instability or in the case of the existence of large external loads), the controller shutdown the actuator automatically. To increase security, it was also imposed a limitation to the amplitude of the control force.

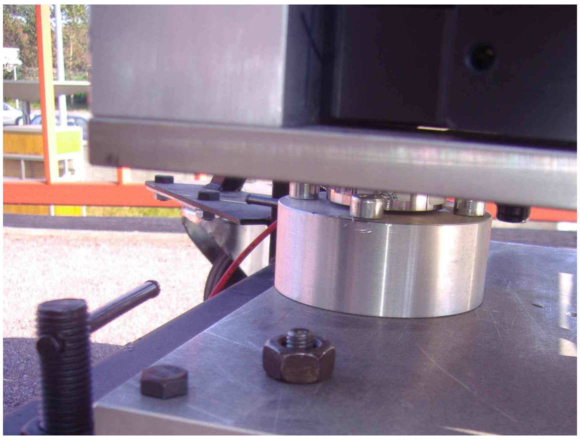

Figure 11. a) View of the actuator; b) Load cells responsible to measure interaction forces

Figure 12. a) Accelerometer used to measure the system response; b) Controller and host computer

Figure 13. a) Pedestrian skipping near the actuator; b) Pedestrian walking along the footbridge

5.2 Description of the tests

Taking into account that only one actuator was available to implement real tests, it was decided to performed control experiences at some selected sections of the deck. To this purpose, it was considered the actuation system positioned at section 45 to control the 2nd mode, at section 49 to control simultaneously the 2nd and 4th mode and at section 52 to control the 4th mode. The external load was induced by a pedestrian with approximately 700N weight, when skipping at a fixed position or walking along the footbridge with an appropriate step frequency (Figure 13a and 13b).

The first experience consisted in exciting the structure at section 52 by a pedestrian skipping at 2.42Hz with the help of a metronome. Initially, when the pedestrian started to skip, the control system was switched off and, a while after, the gain was gradually increased until reaching *g*=3600. Through a sudden stop of the excitation, it was possible to evaluate the damping ratio achieved by analysing the respective free decay envelope. The graphs below represent the results obtained which allow the understanding of the effect of the control action applied to the structure. It can be seen that as the gain increases (Figure 14a) the system response decreases (Figure 14b), which corresponds to an increment of damping from 1.76% (measured before with the control off) to 2.54%. The reduction of the response is accomplished by an increase of the control force (Figure 15a)

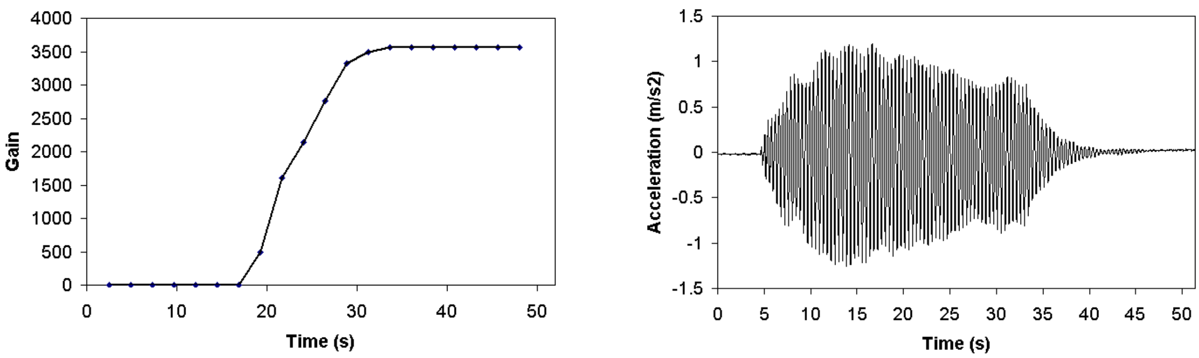

Figure 14. a) Evolution of the control gain; b) System response at section 52

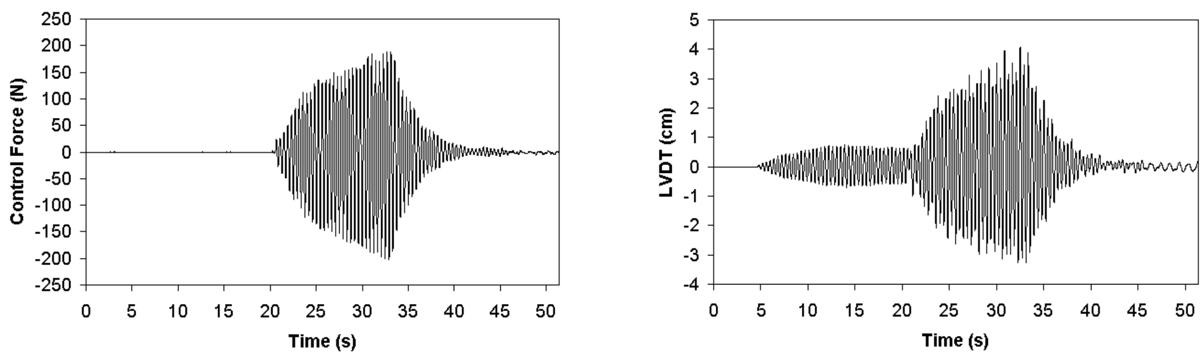

Figure 15. a) Control force measured at the power amplifier; b)Relative displacement of the active mass

which originates the force motion of the active mass. In fact, the actuator acts initially like a passive mass attached to the structure but, when the control force starts to operate, the mass is forced to move, absorbing the energy from the main system (Figure 15b).

Using the actuator positioned in the same section as before, another test was conducted by exciting the footbridge with a pedestrian walking along the deck with a step frequency of 2.42Hz. As in previous case, the structural response was dominated by the harmonic vibration of the 4^{th} mode, but with lower levels of vibration with regard to the results obtained with the skipping tests. For this reason, it was possible to increase the control gain to higher values in order to improve efficiency by introducing more damping into the system. Figure 16a represents the comparison of the structure response measured at section 13 obtained without and with control adopting g=11900. A significant reduction of the system response can be observed which corresponds to a reduction of the maximum acceleration from $0.30m/s^2$ to $0.19m/s^2$. The effect of the control system can be interpreted as an artificial increase of the damping factor from 1.76% to 4.15%. This is a nice result, mainly if taking into account that the ratio between the active mass and the modal mass is just 0.15%. Notice that the reduction of the maximum amplitude observed is not in the inverse proportion of the increase of the damping because this principle only applies to stationary responses, which is not the present case. In fact, vibration modes with lower damping ratios take more time to stabilize than the higher damped modes which leads to an apparent loss of proportionality in transient responses.

Comparing the experimental results obtained in this test to the numerical study described in section 5.3, it can be observed a good approximation between the analytical estimates of the damping ratios of the system and the values effectively measured in the controlled structure. For this particular experience, the analytical damping ratio of the 4^{th} vibration mode expected to get by adopting the maximum gain was 3.49%, assuming an initial value of 1%, which is in line with the measured value of 4.15%, taking into account that the effective initial value was 1.76%. On the other hand, the adoption of high values of gain may introduce stability problems, as it is perceptible by the analysis of the Figure 16b. This situation was clearly predicted in the analytical study by imposing a maximum gain of 11700, which was slightly exceeded in this experience.

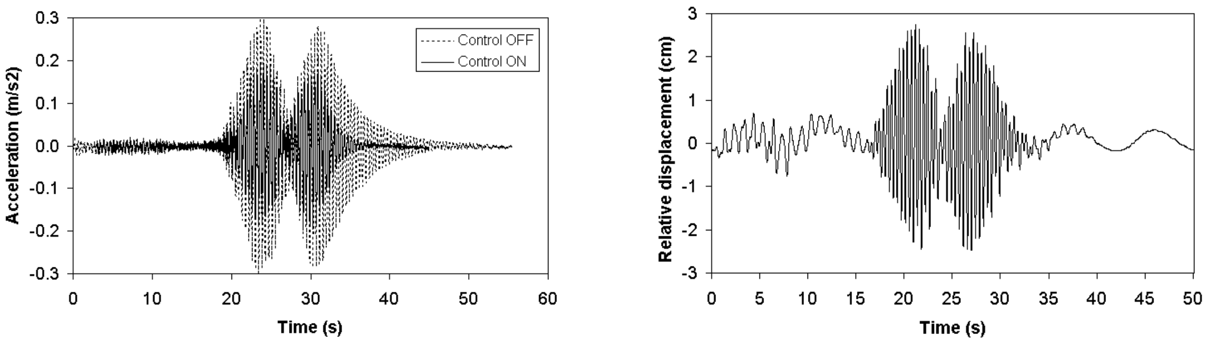

Figure 16. a)Controlled and uncontrolled response at section 52; b)Response of the active mass

The previous experience shows some advantages of using active over passive control systems. It was mentioned before that the mass ratio used was 0.15% which is in fact a small value when compared to the mass ratio of an equivalent TMD. In this case, to achieve the same performance in terms of the structural response, this passive device should have a mass ratio of 0.72%, i.e., a passive mass of 160kg which is about five times greater than the active mass of 34kg of the actuator. This is possible because the active device has a maximum stroke compatible with the observed relative displacements of the moving mass. In theory, it would be possible to use even a smaller active mass as long as the stroke of the device and the space available at the structure allowed it. This advantage, combined with the possibility of controlling several modes with the same device and the inexistence of tuning problems, may constitute an attractive control solution for small structures. In the case of the experience just described, the maximum relative displacement of the active mass was 2.6cm (Figure 16b), which is about three times greater than the displacement expected in an equivalent TMD. However, this amplitude is still considered acceptable.

After the conclusion of the tests on section 52, the actuation system was positioned at two other locations, namely at sections 45 and 49. The footbridge was excited with skipping and walking loads and, in each case, the control gain was adjusted to obtain the possible maximum efficiency. Table 2 summarizes the results obtained in these tests as well as the previous results in section 52. It is worth noting that in each case there is some level of improvement of the structural response. For the reasons already explained, the best performance of the control system corresponds to locating the actuator in section 52, where it was possible to increase 135% the damping factor of the 4th vibration mode which led to a reduction of 37% of the structural response to pedestrians loads induced by walking. It can also be concluded that the control system is generally more efficient when controlling structural responses to walking loads than skipping loads, because the level of excitation is smaller in the former case, allowing exploring higher levels of gain. Notice that in several cases the control gain adopted was superior to the maximum allowed to avoid instability. This was possible because in resonance situations the system response is conditioned by the dominant pole, minimizing the instability originated by the 1st vibration mode.

Position of the actuator	Load type	Without control		With control				
		Damp. fact.(%)	Accel. max.(m/s^2)	Gain $\times 10^3$	Damp. fact.(%)	Accel. max.(m/s^2)	C. force max (N)	Rel.disp. max.(cm)
45	Skipping at 2.08Hz	1.44	0.63	3.7	1.87	0.48	150	-
	Walking at 2.08Hz	1.44	0.28	5.2	2.23	0.19	77	-
49	Skipping at 2.08Hz	1.44	0.82	4.0	1.95	0.60	155	4.8
	Walking at 2.08Hz	1.44	0.31	7.0	2.31	0.21	82	2.9
	Skipping at 2.42Hz	1.76	0.96	6.4	2.43	0.73	269	5.2
	Walking at 2.42Hz	1.76	0.30	10.4	2.75	0.25	130	3.1
52	Skipping at 2.42Hz	1.76	1.30	3.6	2.54	0.86	195	3.7

Walking at 2.42Hz	1.76	0.30	11.9	4.15	0.19	140	2.6

Table 2. Summary of the results obtained in the tests

6 CONCLUSIONS

This document describes the numerical study and the implementation of an active system to control vibrations in a "lively" footbridge located at FEUP. Although the levels of vibration regularly observed may not be considered excessive according to some design codes, this footbridge experiments regularly unusual vibrations for this type of structure, which has motivated the implementation of a control system for research purposes. Due to the existence of local vibration modes and the possibility of the occurrence of resonance phenomena, it was proposed an active control system composed by two actuators positioned at each span of the structure, commanded by Direct Velocity Feedback control law. The real implementation of this control system was restricted to the utilization of only one AMD existing in the Laboratory of Vibrations and Monitoring which has constituted a limitation in terms of a global solution for the control of the footbridge. This device was positioned in several sections of the deck, reducing vibrations originated by pedestrian loads like walking or skipping, exciting the structure in resonance conditions. In spite of not having the adequate characteristics in terms of natural frequency and active mass, it was observed a generalized reduction of the structural response according to several scenarios of the actuator positioning and frequency of excitation. In a particular case, it was possible to increase the damping factor of a specific vibration mode from 1.76% to 4.15% which has conducted to a reduction of dynamic response from 0.30 to 0.19m/s^2.

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