Wire rope springs for passive vibration control of a light steel structure

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Abstract – The restoring force of wire rope springs is generated by the wire ropes deformation that, at the same time, dissipates energy due to the friction forces arising between its wires during the deformation of the rope.
In this paper, the dynamic behavior of a light structure isolated by means of isolators, constituted by wire rope springs and a ball transfer unit, is investigated; the adopted structure simulates objects sensitive to seismic accelerations, like works of arts or cabinets containing electromechanical devices whose functioning must be ensured during seismic events. In the first part of the paper the system dynamic characteristics are evaluated by means of an experimental modal analysis conducted on a shaking table by assigning different laws of motion to its moving platform. The frequency response has been obtained through mono-frequential and multi-frequential excitations; the platform has been also moved with laws deduced from accelerograms recorded during seismic events to evaluate its insulation efficiency. The modal analysis has shown the frequency range where the proposed isolation system produces a beneficial action for the specific passive vibration control application.
Experimental data have been used to validate a numerical model that can be adopted to simulate operating conditions different from those of the experimental tests; the results of several simulation are reported to highlight the influence of some parameters on the system dynamic behavior.

Key-words: wire rope springs, shaking table tests, experimental modal analysis, nonlinear dynamics, vibration control.

1. Introduction

A new type of seismic isolator, consisting of wire rope springs (WRS) and a ball transfer unit (BTU), has been developed at the DII laboratory. The device [1] is constituted (Fig. 1) by two plates between which it is interposed a ball transfer unit which allows horizontal relative displacement between the two plates with low friction; the two plates are connected by wire rope elements whose deformation provide a restoring force and a dissipative action due to friction between the wires of the cables.

The device can thus be considered rigid in the vertical direction while the horizontal stiffness can be suitably calibrated choosing the number and the type of rope elements or defining their length (Fig. 2) and/or diameter.

The isolator horizontal stiffness must be tuned ensuring that the natural frequency of the isolated system is moved away from the frequency interval where is concentrated most of the seismic energy.

In [1] it has been also presented an analytical model of the WRS-BTU behaviour; the model assumes that the restoring force developed by the ropes is characterized by the sum of three contributions: elastic, hysteretic and a nonlinear contribution proportional to the cube of the displacement; furthermore the model takes into account the rolling friction force due to contact between the BTU main ball and the device upper plate. The restoring force description is used to preview the dynamic response of an isolated structure, subjected to seismic ground acceleration and some typical behaviour of nonlinear vibrating systems.

The isolator investigation has been extended; in [2] several tests have been reported in order to estimate the BTU friction coefficients as function of the vertical load. The isolator has been tested on the BPI test rig [3-6] to deduce the force-displacement cycles for different amplitude and frequency excitations; the cycles have been also obtained excluding the BTU main ball contact to put in evidence the rolling contribution.

Starting from these cycles, the device equivalent shear stiffness and the horizontal equivalent damping have been estimated for different operating conditions. Finally, to check the device insulation efficiency, a base isolated structure has been tested on a shaking table excited with harmonic law displacements.

The most commonly techniques used to study the dynamic behaviour of a base isolated structure under seismic loading are the hybrid simulation [7] and the shaking table tests. In the hybrid testing a physical part of the system interacts with a numerical one, in particular, in the case of the base isolation, the physical part is the isolator and the numerical one is the structure. The advantage of this method, with respect to the shaking table tests, is that it is not required the physical structure but, at the same time, the hybrid simulation requires a very complex control system in order to have good results.

The present paper reports the tests conducted to define the frequency response for a light isolated structure tested on a shaking table, imposing harmonic displacement laws, sweep functions and movement laws derived from accelerograms recorded during seismic events.

Experimental results have been also adopted to identify the parameters of a numerical model to highlight some peculiarities of the dynamic system. As the hysteretic cycles [2] have shown a frequency dependence motion, the restoring force model has been integrated with a viscous type contribution, proportional to the relative speed between structure and platform.

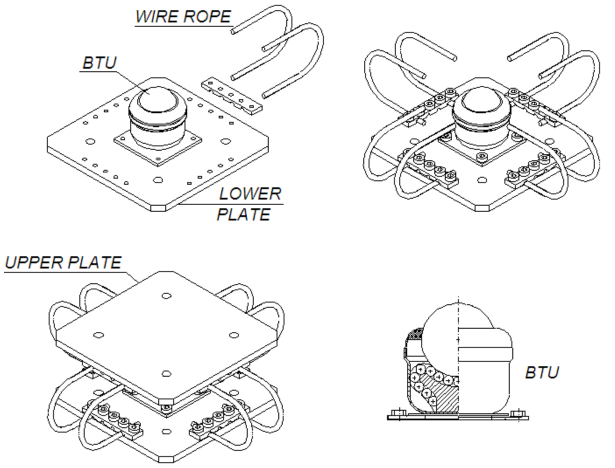


Fig. 1 WRS-BTU isolator

Fig. 2 Isolators with different wire rope lenght

2. Experimental investigation

The experimental investigation has been conducted to obtain the system frequency response function by adopting the classical techniques of the experimental modal analysis.

The experimental plant (Fig. 3) is constituted by a steel structure, simulating a cabinet, on four WRS-BTU isolators. The steel frame is equipped with additional masses in order to modify the overall mass *m*; the lower plates of the isolators have been connected on the platform of the shaking table; the platform can be moved with an assigned motion law by means of a controlled hydraulic cylinder [8]. In order to control the platform displacement taking into account the influence of the isolated steel frame on the system dynamics, different control strategies have been developed [9,10]. For the tests reported in this paper the adopted platform control has been obtained with a linear feedback displacement control and a feedforward action based on the nonlinear inverse model of the hydraulic actuation system [11].

Two accelerometers have been placed on the platform and on the cabinet respectively, with the sensitivity axis displaced along the moving direction.

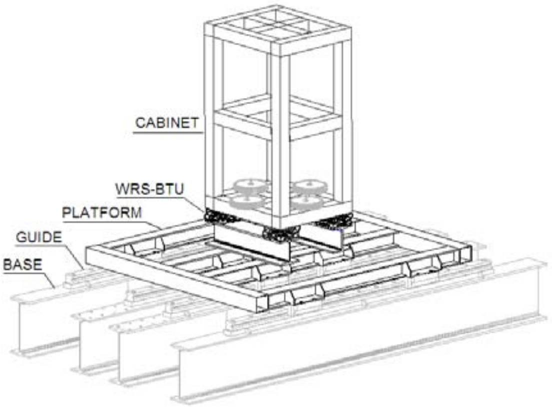


Fig. 3 The test rig

At first, mono-frequential excitation techniques have been employed; an harmonic motion, with assigned amplitude and frequency, has been imposed to the platform. This technique is accurate and simple to apply, however it requires much time and many elaborations to obtain adequate resolution for frequency response curve. The procedure has been limited to few frequency values, and the results, in terms of modulus, have been useful to check the results obtained by means of a multi-frequential excitation that allows to quickly deduce the frequency response.

In particular, a first set of tests have been performed exciting the platform with a constant amplitude of 5 mm and frequency values comprised in the 0.75 – 5.0 Hz interval, with a step of 0.25 Hz;

Fig. 4 and 5 show the accelerograms and their relative FFT obtained for platform displacement amplitude of 5mm and 10mm and for a frequency of 2Hz. From both diagrams it can be deduced that the cabinet acceleration is lower than the platform one and therefore the cabinet can be considered isolated.

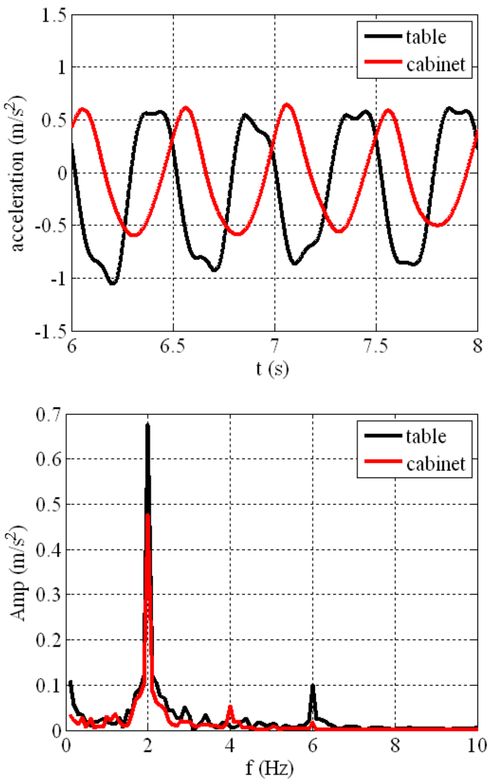


Fig. 4 Accelerograms and FFTs for Amp=5 mm and Freq= 2 Hz

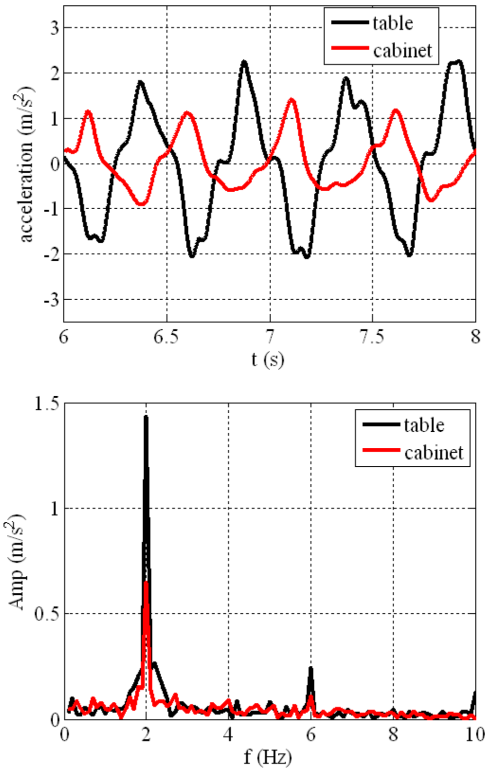


Fig. 5 Accelerograms and FFTs for Amp= 10mm and Freq= 2 Hz

Cabinet signals are characterized by several ultra-harmonic components that may be due to the presence of nonlinear wire rope springs but also to the fact that platform acceleration is not purely harmonic, as evidenced in the FFTs. In fact the hydraulic actuator control system is able to well track the desired displacement law but not the same happens for the acceleration as evidenced in Fig. 6, 7 where the comparisons between the desired platform displacements and accelerations for both tests are reported.

In particular, Fig. 4,5 show that the table acceleration spectra contain an appreciable ultra-harmonic 6 Hz component which gives an irregular shape to the acceleration signal; such component is filtered by the suspension and therefore it is absent in the cabinet acceleration.

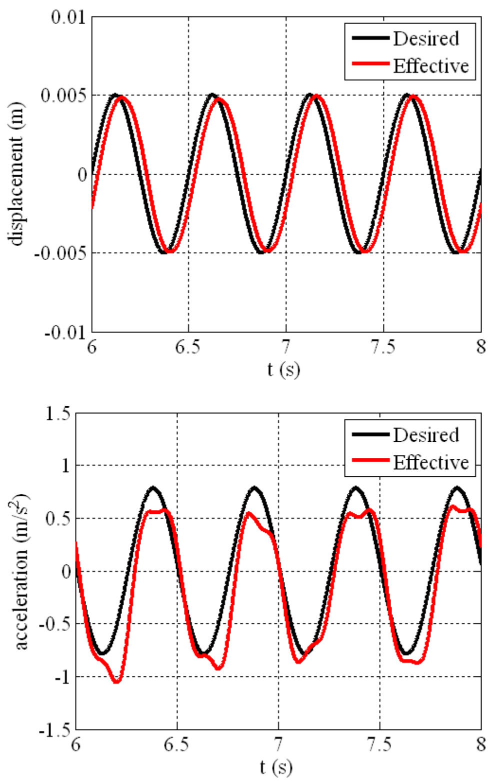


Fig. 6 Platform displacement and acceleration: comparison with the reference diagrams (Amp=5 mm and Freq= 2 Hz)

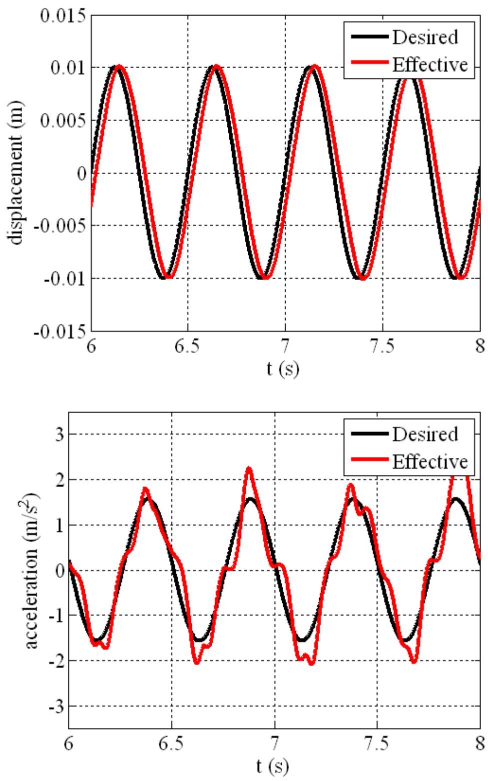


Fig. 7 Platform displacement and acceleration: comparison with the reference diagrams (Amp=10 mm and Freq= 2 Hz)

To highlight the acceleration transmissibility, in Fig. 8 the ratio between the cabinet and the platform amplitude acceleration is reported; in particular, the diagram shows the ratios between the harmonic components synchronous with the excitation frequency. It can be noted that the cabinet acceleration exceeds the platform one for frequencies close to 1 Hz while the acceleration is attenuated for frequencies higher than 1.5Hz.

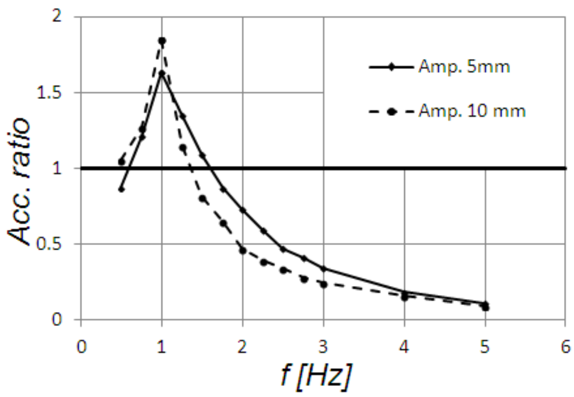


Fig. 8 Acceleration ratio vs. forcing frequency

Driving the platform with a frequency sweep motion characterized by a constant amplitude equal to 5 mm and a frequency increasing with a rate of

0.01 Hz/s until 3Hz both, platform and cabinet accelerations have been detected; reporting in the same diagram the corresponding frequency spectrum curves, it can be noted that the acceleration platform amplitude grows with parabolic law while the cabinet acceleration has a trend that resembles the acceleration response of a linear single degree of freedom system with constant parameters for which the acceleration frequency response has a peak in correspondence of the natural frequency f_n (whose amplitude depends on the damping ratio), passes for the characteristic point for a frequency value equal to $f_n\sqrt{2}$ and then it begins to grow again (Fig. 9).

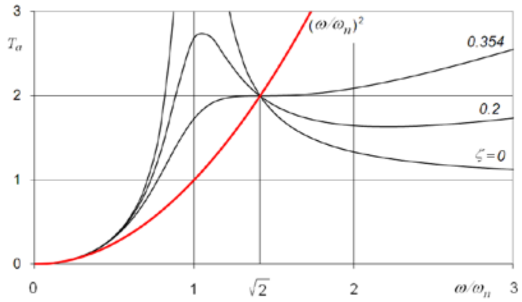


Fig. 9 Acceleration transmissibility for different value of the damping ratio

The frequency spectrum (Fig. 10) shows that the system is characterized by a damping that is able to contain the acceleration peak in correspondence of the resonance; for excitation frequencies higher than 1.5 Hz the isolation system reduces the amount of acceleration transmitted to the cabinet that is always lower than that of the platform.

The figure shows that this kind of isolation system is able to limit the acceleration transmission in resonance condition as well as for higher frequency excitation.

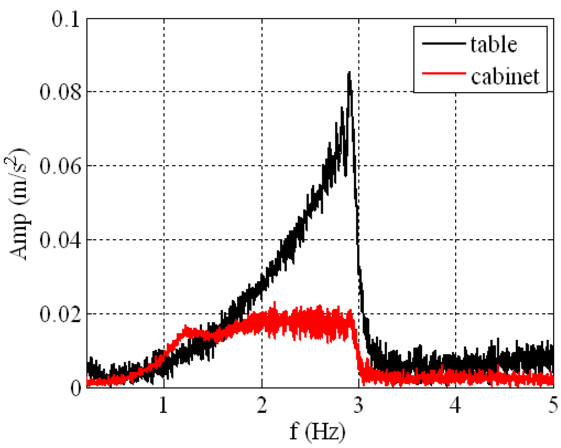


Fig. 10 Acceleration frequency spectrum for the sweep motion (m=185 kg)

Finally, the platform has moved with a law motion derived from accelerogram recorded during seismic events. The data refer to two earthquakes occurred in Italy (Friuli 1976; Irpinia 1980).

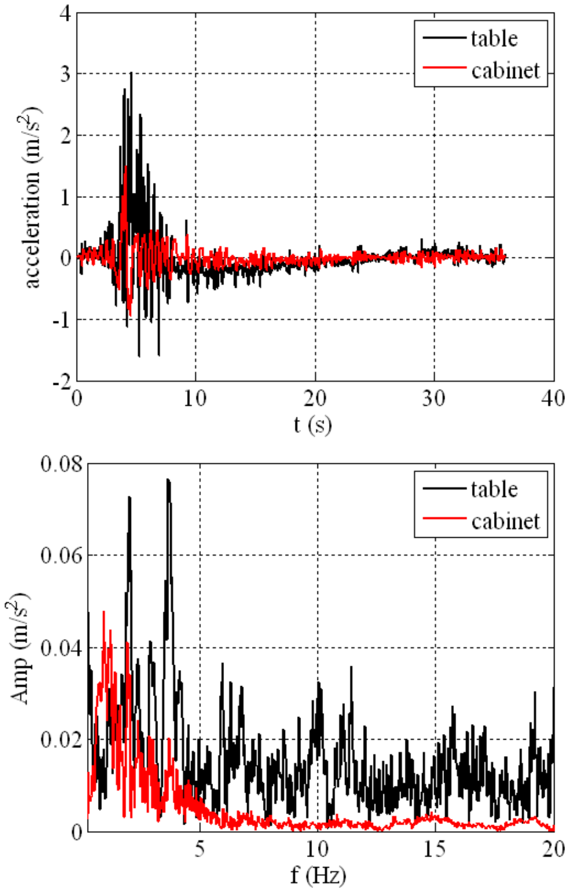
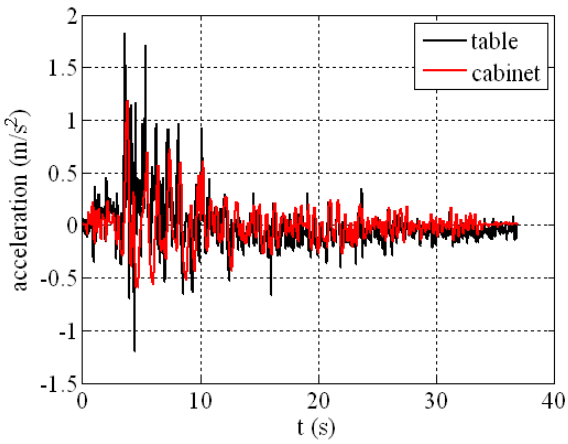


Fig. 11 Friuli earthquake (Italy, 1976); accelerograms and acceleration frequency spectrum (m=185 kg)



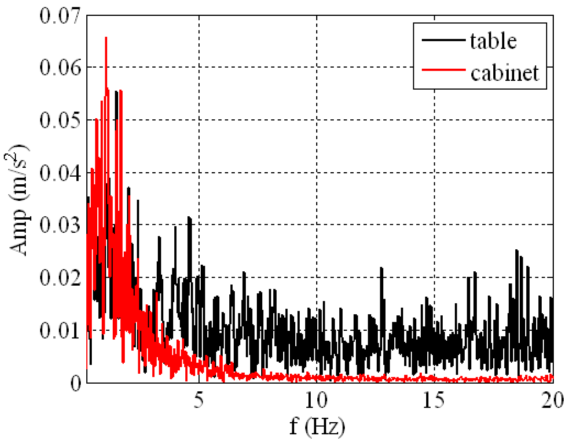


Fig. 12 Irpinia earthquake (Italy, 1980); accelerograms and acceleration frequency spectrum (m=185 kg)

The comparison between the acceleration time history of the cabinet and the platform one (Fig. 11,12) shows that the cabinet acceleration peaks are always lower than the platform ones and therefore the system can be considered isolated. The tests have been repeated with a lighter cabinet mass (165 kg) and the trend results similar even in terms of frequency spectrum.

3. Numerical investigation

The nominal sizes of the cabinet have been chosen to have a coincidence between the center of gravity and of the rigidity centre of the isolation system; in the model it can be assumed that the cabinet mainly moves along the direction of excitation and therefore it can be schematized as a single degree of freedom vibrating system. The dynamical behaviour of the isolated structure can so be simulated by integrating the following equation:

$$m\ddot{x} + 4F = 0 \tag{1}$$

being $x(t)$ the cabinet displacement and F the restoring force exerted by each isolator along the excitation direction.
Furthermore, by indicating with $x_g(t)$ the platform (ground) displacement and with: $d(t) = x\ (t) - x_g(t)$, the relative displacement between cabinet and ground, the cabinet motion equation in term of relative motion is:

$$m\ddot{d} + 4F = -m\ddot{x}_g \tag{2}$$

The isolator reaction F can be modeled as the sum of the following components:

$$F(t) = F_c(t) + F_r(t) \tag{3}$$

where, F_c represents the wire ropes action and F_r is the BTU rolling friction force.
The F_c component is given by the sum of four independent components:

$$F_c(t) = F_{el}(t) + F_h(t) + F_{nl}(t) + F_v(t) \tag{4}$$

being: F_{el}, is the elastic component, proportional to the displacement; F_h is the hysteretic component, F_{nl} is a nonlinear component proportional to the cube of the displacement, and F_v is the viscous damping component.
The hysteretic force can be derived from the Bouc Wen model. The analytical expression of the hysteretic force has been presented in [14, 15] and it is here reported:

$$\begin{aligned} F_h(t) &= k_w w(t) \\ \dot{w}(t) &= \rho \dot{x}(t)\left[1 - \mathrm{sgn}(\dot{x}(t)) w(t)\right] \end{aligned} \tag{5}$$

By properly choosing the set of parameters (ρ, k_w), it is possible to accommodate the response of the model to the real hysteresis loops; the use of parameter identification techniques is a practical way to perform this task [16, 17, 18].
The BTU rolling component $F_r(t)$ is expressed through a Coulomb model:

$$F_r = C_a N \cdot \mathrm{sgn}[\dot{x}(t)] \tag{6}$$

The values of the friction coefficient C_a, versus the vertical load N, have been experimentally identified [2].

4. Comparison between simulated and experimental data

To verify the model reliability, several simulations have been performed with the same data of experimental investigations. In particular, Fig. 13 and 14 show the comparisons with experimental results obtained with a harmonic excitation at 2 Hz, and amplitudes of 5 and 10 mm respectively.
The results reported in Fig. 13 show a good agreement between the experimental and simulation data; while, in the case of an amplitude of 10 mm the experimental results are slightly different with respect the simulation ones; this can be due to the hypothesis of a mono-dimensional

motion of the isolated cabinet that for a higher value of the ground acceleration amplitude it is not strictly appropriate.

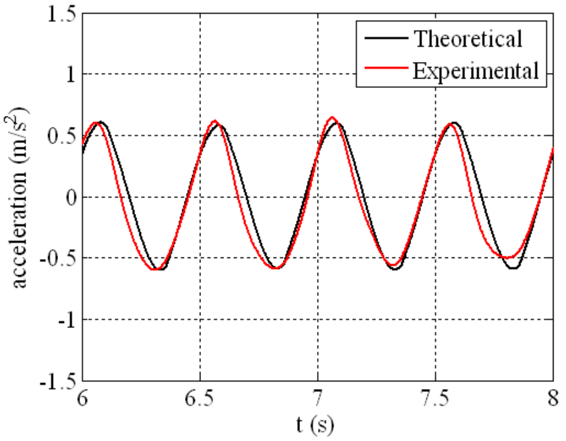


Fig. 13 Comparison between theoretical and experimental cabinet acceleration (Amp= 5 mm and Freq = 2 Hz)

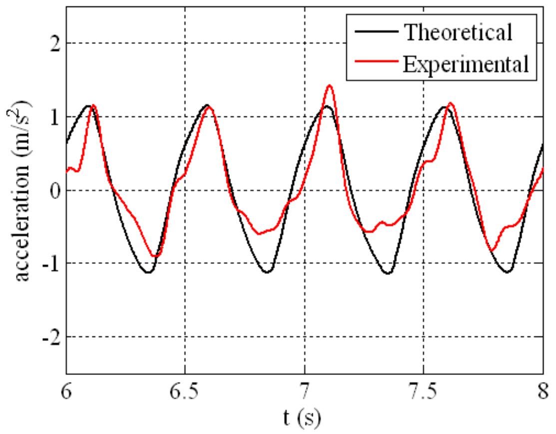


Fig. 14 Comparison between theoretical and experimental cabinet acceleration (Amp= 10 mm and Freq = 2 Hz)

Fig. 15 and 16 show the comparison with the sweep tests conducted with cabinet masses of 165 kg and 185 kg.
The frequency response of the numerical model for both the sweep tests clearly reproduce the experimental behavior of the cabinet.

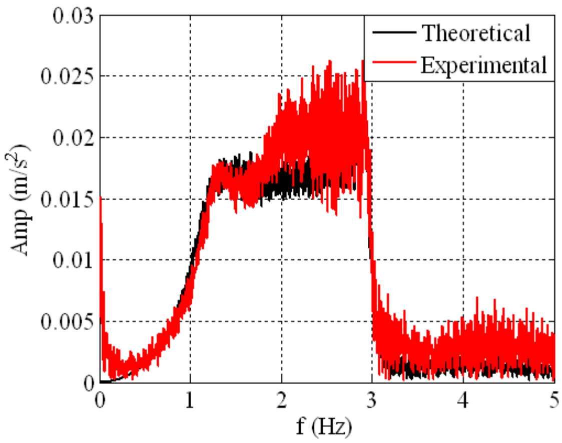


Fig. 15 Comparison between theoretical and experimental frequency spectrum of the cabinet acceleration (sweep test m = 165 kg)

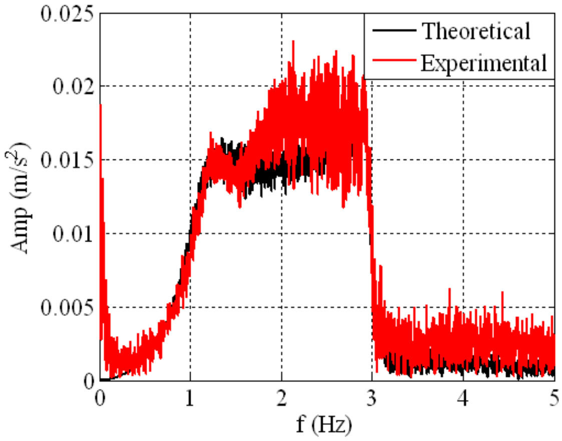


Fig. 16 Comparison between theoretical and experimental frequency spectrum of the cabinet acceleration (sweep test m = 185 kg)

Finally, Fig. 17 and 18 report the comparisons conducted by imposing seismic excitations [19].
Also in the case of a seismic ground motion the model is able to describe the nonlinear dynamics of the system; this is confirmed by both the tracking error between the experimental cabinet acceleration and the theoretical one, and the frequency content of the experimental and simulated spectra.

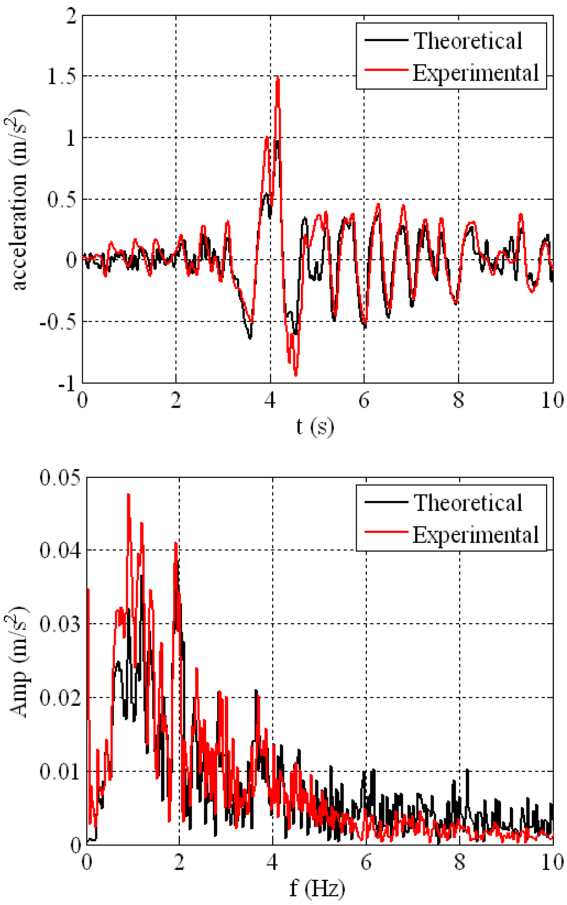


Fig. 17 Friuli earthquake (Italy, 1976); comparison between theoretical and experimental cabinet acceleration (m=185 kg)

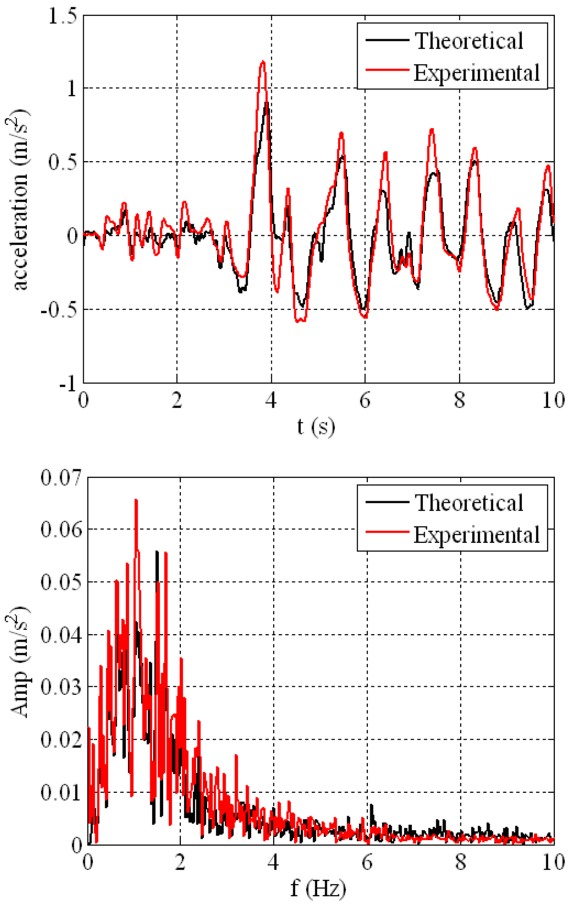


Fig. 18 Irpinia earthquake (Italy, 1980); comparison between theoretical and experimental cabinet acceleration (m=185 kg)

The comparisons between theoretical and experimental data for all the tests show that the model is validated and it can so be adopted to simulate operating conditions different from those of the experimental tests. The analytical description of the restoring force is able to reproduce the nonlinear characteristic of the isolator; this is an important results because a linear approach don’t allow to describe the nonlinear nature of the isolated steel frame [20].

The isolation system efficiency can be evaluated adopting different values of the cabinet masses. Fig. 19 and 20 report the cases for m = 198 kg and 132 kg respectively; in both cases platform motion is characterized by an amplitude of 10 mm and a frequency excitation of 2 Hz. The acceleration ratio is equal to 0.47 in the first case and 0.67 in the second case.

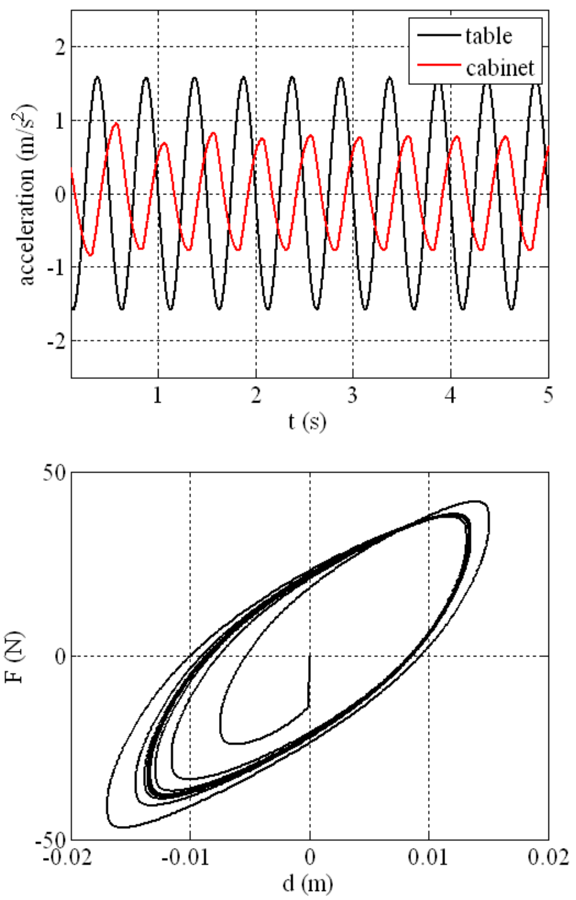


Fig. 19 Accelerations and restoring force for cabinet mass of 198 kg. Platform motion: Amp= 10 mm and Freq = 2 Hz

The simulation results show that an increase of the suspended mass value produces a reduction of the acceleration ratio.

For the same tests, the diagram of the restoring force as a function of the relative displacement between the platform and the cabinet is reported.

The maximum value of the isolator shear deformation for both the tests is less than the limit equal to 0.05 m.

5. Conclusion

An experimental investigation has been conducted to evaluate the efficiency of WRS-BTU seismic isolators.

The dynamic characterization of a laboratory cabinet, equipped with four isolators, has been performed with a shaking table.

The seismic performances of the isolators have been evaluated comparing the imposed ground acceleration and the cabinet one.

First tests have been conducted by means of harmonic and sweep excitation. Other tests have been executed even with recorded seismic excitation to confirm its efficiency.

The results of the experimental investigation have highlighted the frequency range where the WRS-BTU isolators can be utilized for vibration reduction. This aspect could be an important indication for the design of passive vibration control by means of these devices.

In order to have a mathematical description of the isolator restoring force, an analytical model for

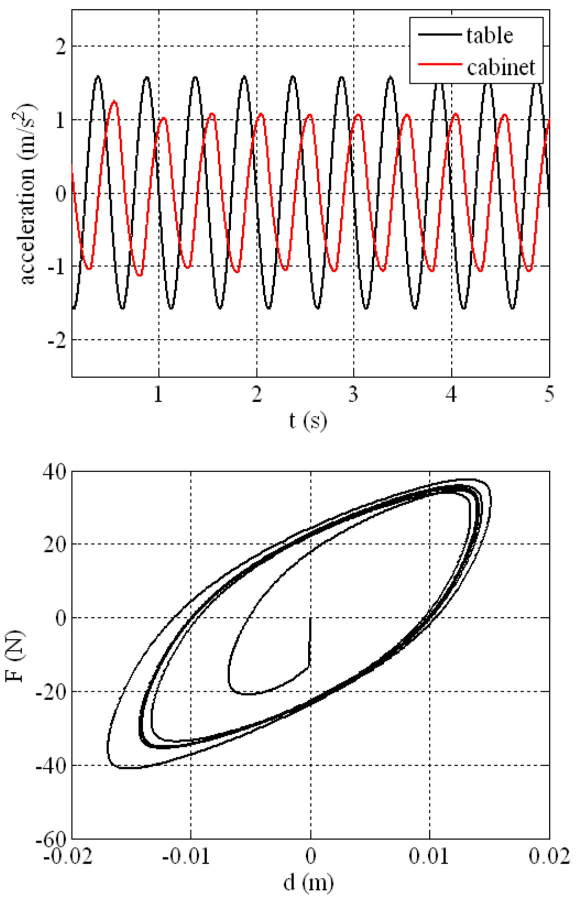


Fig. 20 Accelerations and restoring force for cabinet mass of 132 kg. Platform motion: Amp= 10 mm and Freq = 2Hz

describing the hysteretic behavior of the WRS-BTU isolator has been developed and experimentally validated. The analytical prediction of the response of the cabinet supported by the proposed isolators has been in good agreement with experimental results.

The validated nonlinear model has been also used to simulate operating conditions different from the those experimentally evaluated on the shaking table.

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