# Study on a recently-developed impact damper for reducing wind-induced cable-stay vibration

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**ABSTRACT:**

A method to increase the structural damping of stay cables for mitigating wind- and wind-rain-induced vibration (Matsumoto et al, 2010; Phelan et al, 2006; Zuo et al, 2010) has been developed at Vienna University of Technology. Advantages with respect to other damping devices, proposed by engineers in the recent past, are the simple technology and the fact that all components, required for the new mechanism, are located inside the stay cables. The new device works like an impact damper (Reed, 1967), embedded in the stay-cable in order to decrease the dynamic oscillation in various cable eigen-modes. In a full scale field test on stay-cables of 31-meter length, the damping ratio of a conventional stay and the one of a cable equipped with the proposed damping system were extracted. This study discusses the development of an analytical model for replicating the behavior, observed in the experiment. In its simplest form, the dynamics of behavior can be modeled as a mass-spring-dashpot secondary system, attached to the cable (Nayeri et al, 2007). A reduced-order analytical model for (simplified) forced and free-vibration dynamics of the cable with damping system has been derived at Northeastern University.

*Keywords: Cable-stayed bridges, wind-induced stay vibration, impact damper*

## 1. INTRODUCTION

Cables are often used to support static loads of different types of structures (Irvine, 1981). The progressive increment in the main span length of stay-cable-bridges contributes to more frequent large amplitude oscillation of the stays. The induced stresses in the cables may cause fatigue in the stays, at the anchorages, and in the bridge-deck. The bending stresses at the anchorage may not be negligible and may, on occasion, need to be considered (the higher the tension force the higher the sharp bend at the anchorage). Even small vibrations are sources of friction- corrosion and fatigue as the cable tends to kink. The stiffness of a cable is mainly due to the axial loads and is the so-called “geometrical stiffness”. The bending stiffness of a cable is usually marginal due to the thin cross section (Svensson, 2011).

The slenderness of the stay cables makes them prone to vibration due to wind-induced dynamic loading; various loading mechanisms have been observed and are not yet fully understood (Zuo et al, 2010). Wind, and a combination of wind and rain, can excite the cables directly and produce different types of oscillation. Cable can be excited if an upstream wind flow reaches the cable. The mechanism which usually causes the largest vibration amplitudes is the so-called “dry galloping”. Cable vibration may become self-excited if an axial or a perpendicular flow hits the cable. Influence of Reynolds number, leading to a change in the drag coefficient at certain wind velocities has been reported as one of the potential reasons (MacDonalds and Larose). Researchers (Matsumoto, 2010, Gimsing et al., 2011 and Svensson, 2011) report that other phenomena apart from “dry galloping”, such as wind-rain-induced excitation and ice galloping can lead to dynamic and aeroelastic vibration.

For example, if ice accumulates at the cable or a rain fillet runs along the stay- cable, is there a change in the circular cross section of the cable, and provokes at moderate wind speeds an initiation of galloping. In the case of wind- or wind-rain-induced oscillation full-scale observations and wind-tunnel results suggest that the actual

mechanism, explaining large cable vibrations, is still not completely understood and requires more investigation (Zuo and Jones, 2009; Zuo and Jones, 2010).
A less important excitation mechanism is wake galloping, where gusty wind form large obstacles on upstream flow can cause vibrations. Vortex shedding due to the Karman vortices lead to small amplitude vibrations in the range of the cable diameter or less. Motion-induced vibration is also possible due to oscillation of the pylon or the bridge deck; the latter are called parametric excitations.

The commonly used cables in stay cable bridges in Europe and North America consist of high strength steel strands with a low material damping. The damping ratio is approximately 0.1% with respect to critical value and is mainly produced by stretching the strands (Yamaguchi et al, 1998). The low hysteretic energy loss is believed to be the key factor of the cable vibrations. One way to increase the structural damping is to install dampers orthogonally to the stay's longitudinal axis, near the cable edges. The distance between the anchorage of the cable and the damping device amounts to 2-4% of the cable length (Gimsing et al., 2011). Among others, Kovacs approximated the optimal viscous damping for a transverse mounted damper close to the cable anchorage by comparing the two extreme states of the damper (Kovacs, 1982). On the one hand the performance of a cable without damping device is evaluated and on the other hand the effect of a rigid damper is taken into account, which means that the cable length would be slightly reduced and the frequency of the cables slightly increased. This semi-empirical interpolation leads to a solution where the maximum modal damping ratio would be half of the ratio between the damper distance from the anchorage to the entire cable length. A numerical analysis conducted by Pacheco et al. (1993) confirmed the results of the optimal damping constant for the viscous damper (Krenk, 2000; Main and Jones, 2002). Similar results have also been used to approximate the external viscous, visco-elastic, friction, semi-active and active dampers directly mounted on the cable anchorage.

Another countermeasure is to modify the surface of the cable surface in order to reduce the spatial correlation of the excitation forces on the surface of the cable due to wind-rain-induced excitation and vortex shedding. In Europe and North America these mainly include "rings" (Phelan et al, 2006), single or double helical fillets embedded on the cable surface, whereas mostly in Asia longitudinal channels are carved in the external cable pipe or dimples and bumps are added (Gimsing et al., 2011).

The present paper describes a new approach of cable vibration mitigation by mounting an "impact damper" along the whole cable length. A preliminary numerical model was developed which provides a good description of the dynamics of the system by accounting for the inelasticity during the impact. This category of dampers operates in areas where ruggedness, reliability and insensitivity to temperature extremes are required (Nayeri et al, 2007). Since cable-stayed bridges are exposed to extreme conditions, these features guarantee reliable operations in all environmental conditions.

## 2. DESCRIPTION OF THE DAMPING SYSTEM

The proposed device works like a vibration absorber, attached along the whole stay cable, in order to decrease the susceptibility to dynamic vibration at various eigen-frequencies. The apparatus is similar to a tuned mass damper, which was first studied by Den Hartog (1985). In its simplest form, it can be modeled as a secondary mass-spring dashpot attached to the cable (Den Hartog, 1985). The stay-cable vibration, mode by mode, can be approximated as a generalized single degree of freedom model. Kovacs also mentions in his paper that a punctual tuned-mass damper, attached to a stay-cable, could increase the damping ratio, as defined by the specific application, even though the technical realization may be challenging (Kovacs, 1982). An alternative approach could be to equip a taut-cable with a "multi-pendulum" along the cable (Fig. 1). The impulses due to the impact, originating from the multi-pendulum device, can lead to a momentum transfer from the primary system and energy dissipation due to internal friction, respectively.

The prototype damping system consists of a hollow tube attached to the stay-cable with an additional strand placed inside the tube (Fig.2). Under dynamic excitation, the energy dissipation can lead to an increment in structural damping. The strand is connected to the structure at one point only. In our method, the required components (strand, tube) are placed inside the primary structural system (the stay); energy is dissipated along the total length of the structure, whereas in conventional damping systems the damping device is exclusively connected to a few points along the axis of the structure.

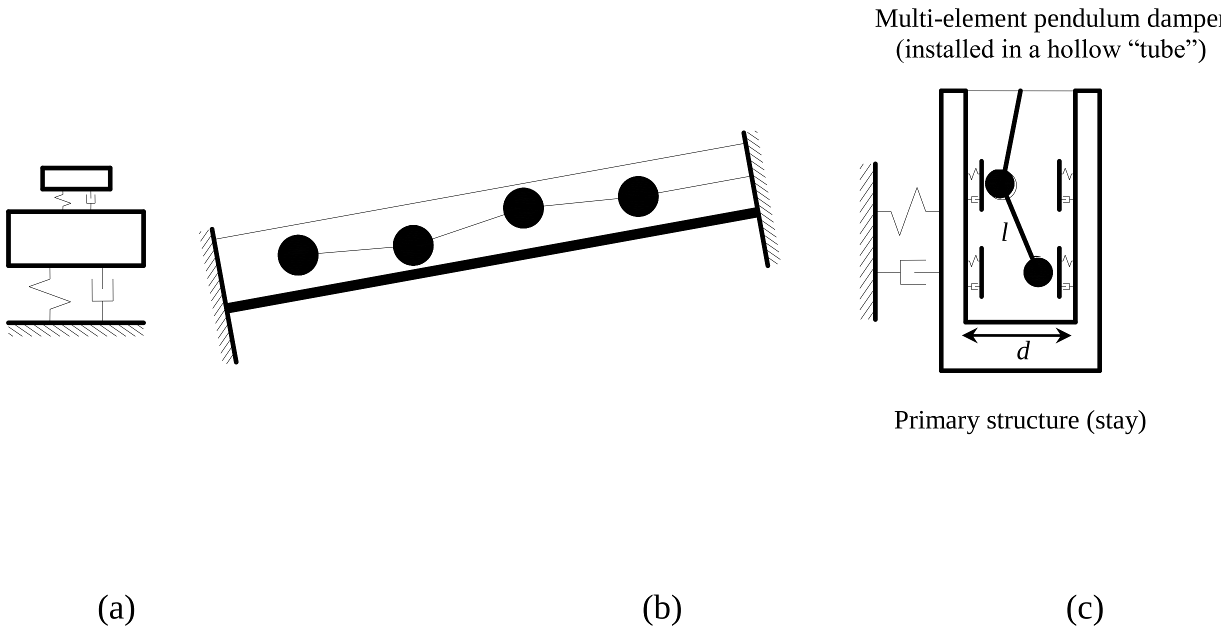


**Figure 1.** Model of the damping system: (a) Tuned mass damper (TMD - Den Hartog), (b) Continuous strand with discrete mass model of the impact damper, (c) Resilient-damper model (schematic; model with 3 DOFs)

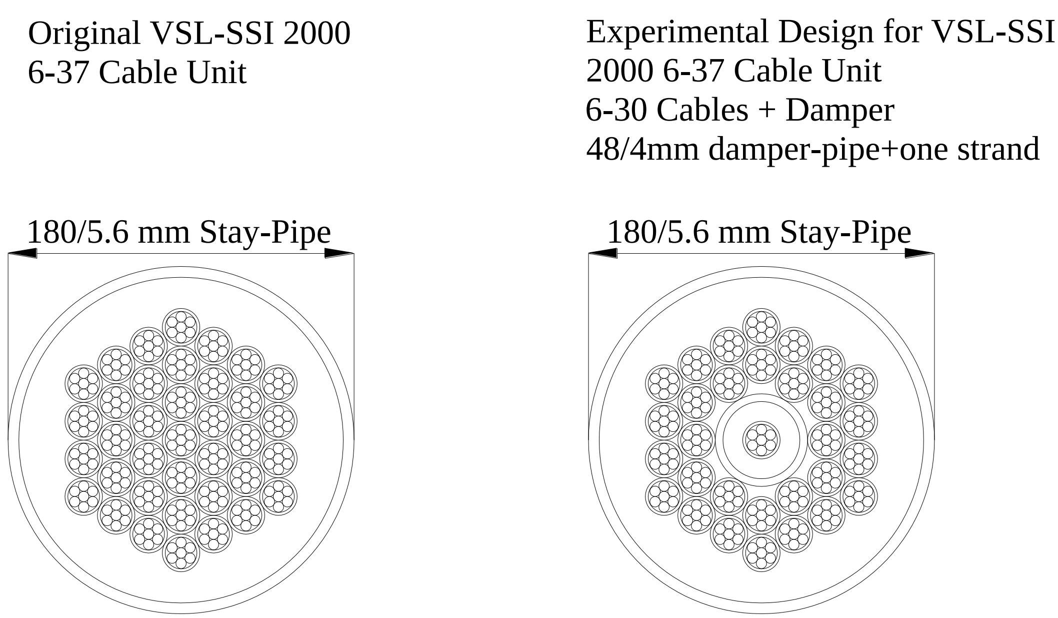


**Figure 2.** Stay cable without and with integrated Damping System

## 3. MECHANICAL MODEL

The passive control system is similar to a tuned mass damper (TMD), which was originally proposed and studied by Den Hartog. In its simplest form the behavior can be modeled as a secondary mass-spring dashpot system, attached to the cable (Den Hartog, 1985). In order to model the behavior of multi-element impact dampers, a series of “resilient” (non-linear) spring-dashpot units interacting with moving point masses, have been used (e.g., Masri 1965; Nayeri et al, 2007). The proposed multi-element pendulum damping system (Fig. 2b) is modeled in a similar way and, in its simplest form, can be thought as a 3DOF dynamic system. The main difference, in comparison with Den Hartog’s original model, is the use of a series of non-linear spring-dashpot mechanical “sub-systems” to simulate the interaction between main structure and damping device.

The damper efficiency for stay oscillation mitigation depends on many parameters. The main issue is the mass ratio between the stay-cable and the inner strand, which is installed inside a hollow tube; the impacts of the inner strand on the main cable are responsible for energy transfer. The second most important parameter is the clearance gap ($d$) between the inner strand and the hollow tube. Depending on these two parameters the efficiency has to be optimized for the anticipated vibration amplitude and the frequency of the stay cable. The frequency of the stay-cable is a function of the cable length, the mass per unit length and the pre-tensioning force of the cable. The coefficient of restitution is a dimensionless parameter between 0 and 1, which has often been used (e.g., Nayeri et al, 2007) to describe inelastic impact and local dissipation; this coefficient depends on the velocity and the material of the colliding masses (primary and secondary system); its value is limited due to the design of the damping system. The bending stiffness of the inner strand may also influence the damping ratio because the impacting mass along the multi-element pendulum can decrease. Finally, the inclination angle of the primary system (cable-stay) is also essential; a horizontal cable has more impacting capacity than a vertical cable. If the stay axis is horizontal the initial position (from rest) of the impacting strand is at the bottom of the hollow tube; the strand can move only if the acceleration of the primary system is above gravity acceleration. The gravity enables a more efficient impact and momentum transfer if the strand moves toward the lower side of the hollow tube in comparison with the impact occurring on upper side.

Figure 1c shows a three-DOF model of the resilient damper, the studied mechanical model; the following equations describe the equation of motion. These equations were adapted from Nayeri et al. (2007). The primary system with mass $M$ is supported by a spring dashpot and is connected to a two-element pendulum. The transverse vibration of the primary system is denoted by $x$ (absolute coordinates), whereas the vibration of each element of the pendulum with respect to the primary system (relative coordinates) is denoted by $z_k$ with $k$=1,2,...,$N$ ($N$=2 in Fig. 1c). Equation (1) describes the global equilibrium of the primary system (stay-cable), idealized as a lumped mass. Equation (2) describes the local equilibrium of each point-mass element of the multi-element pendulum.

$$\ddot{x} = -2 \cdot \zeta_n \cdot \omega_n \cdot \dot{x} - \omega_n^2 \cdot x + \frac{f(t)}{M} +$$

$$+\sum_{k=0}^{N} \mu_k \left[\omega_2^2 \cdot G(z_k) + 2 \cdot \zeta_2 \cdot \omega_2 \; \mathrm{H}(z_k, \dot{z}_k)\right] \qquad (1)$$

$$\ddot{z}_k = -\ddot{x} - \omega_2^2 \cdot G(z_k) - 2 \cdot \zeta_2 \cdot \omega_2 \cdot \mathrm{H}(z_k, \dot{z}_k) -$$

$$-(\mathrm{N} - \mathrm{k}) \cdot \frac{\mathrm{g}}{\mathrm{l}} \cdot z_{k+1} + (\mathrm{N} - \mathrm{k} + 1) \cdot \frac{\mathrm{g}}{\mathrm{l}} \cdot z_k \qquad (2)$$

The “contact mechanics” between each point-mass of the pendulum and the primary system is modeled by a nonlinear spring-dashpot force, as described in Nayeri et al (2007), and by the nonlinear functions $G(z_k)$ and $H(z_k, \dot{z}_k)$ (Fig. 3). The function $G(z_k)$ simulates the rigid barrier of the container and the function $H(z_k, \dot{z}_k)$ simulates the inelastic impact. The displacement of the main system $x$ is affected by the relative displacement $z_k$ of each particle of the pendulum. The quantity $f(t)$ is the external excitation force acting on the main system; $\zeta_n$ and $\omega_n$ are the damping ratio and the natural pulsation of the primary system; $\zeta_2$ is an equivalent damping ratio, simulating dissipation in the spring-dashpot model of impact damper; $\omega_2$ is the "natural pulsation" of each particle-mass impact damper; $\mu_k$ is the mass ratio between the primary system and each mass of the multi-element impact damper. The quantity $g$ is the gravity acceleration and $l$ is the "distance" between any two point-masses on the pendulum.

The motion of the pendulum is a function of the dimensionless parameter $g/l$, which accounts for the inclination of the pendulum and the primary stay, and also approximately simulates "bending stiffness" of the pendulum (i.e., simulating interaction or mobility between two adjacent elements of a pendulum). For example, a horizontal "chain" of masses with no "bending stiffness" corresponds to a $g/l$ factor equal to zero.

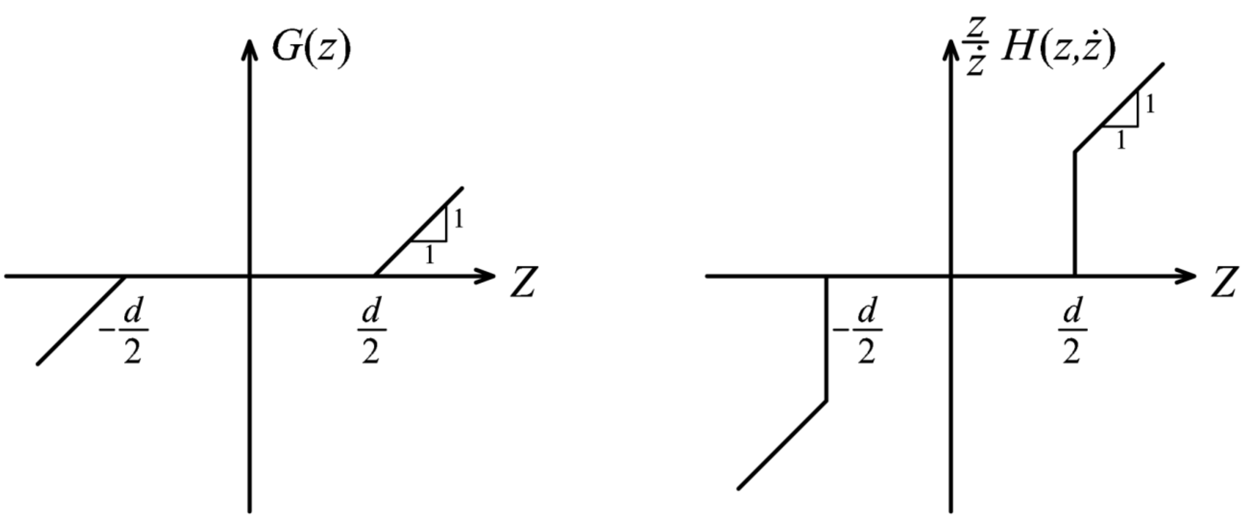


**Figure 5.** Nonlinear function a) $G(z_k)$ and b) $H(z_k, \dot{z}_k)$ (Nayeri et al, 2007)

Preliminary simulation results for a multi-element pendulum impact damper with $N$=1÷100 are shown in Fig. 6÷8. The primary system is excited by a broad-band forcing function simulated by a Gaussian white noise (Nayeri et al., 2007). Random excitation is used to possibly simulate, in a generalized form, various wind-induced excitation mechanisms on the stay. A Runge Kutta algorithm is employed to numerically integrate the equations of motion (Eqs. 1 and 2) and to derive the dynamic response of the system at steady-state, starting from rest at initial time $t$=0. In the estimation of the steady-state random response, ergodicity is assumed; moreover, the influence of transitory regime is eliminated by extending the integration over a very long duration (depending on $\omega_n$ and $\zeta_n$).

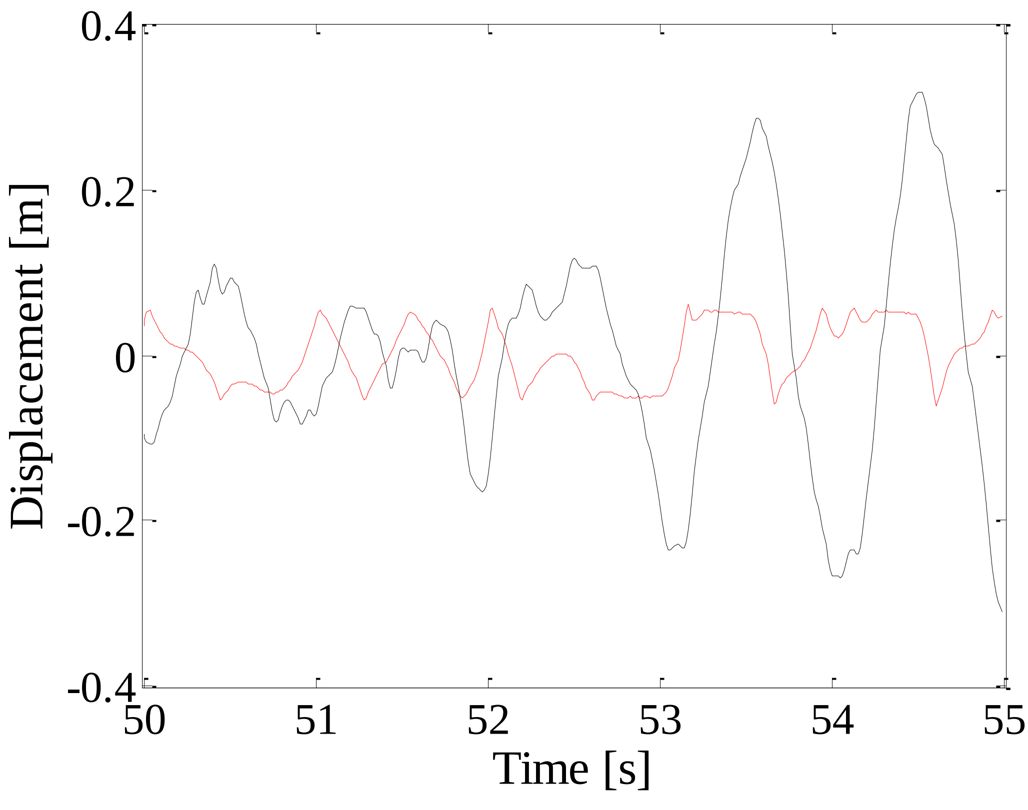


**Figure 6.** Time history of the absolute displacement of the primary system (black) and the relative displacement of the particle (red) inside the tube (single pendulum, $N$=2 with $\zeta$=0.01, $\zeta_2$ =0.10, $\omega_2/\omega_n$ =20, $\mu_k$ =0.1)

The study was conducted by comparing the root-mean-square (rms) dynamic oscillation of the primary system before ($\sigma_{x0}$) and after installation of the integrated damping system ($\sigma_x$). In Fig. 6 the time history of the absolute displacement of the primary system (black line) and the relative displacement of the particle (red line) inside the tube are shown, after the transient oscillation. It can be noticed how the impact of the particle affects the motion of the primary system. The parametric investigation considered various values of the generalized gap ratio $d/\sigma_{x0}$ and the factor $g/l$. Reduction of vibration is observed for all $g/l$ and a wide range of $d/\sigma_{x0}$ (Fig. 7)

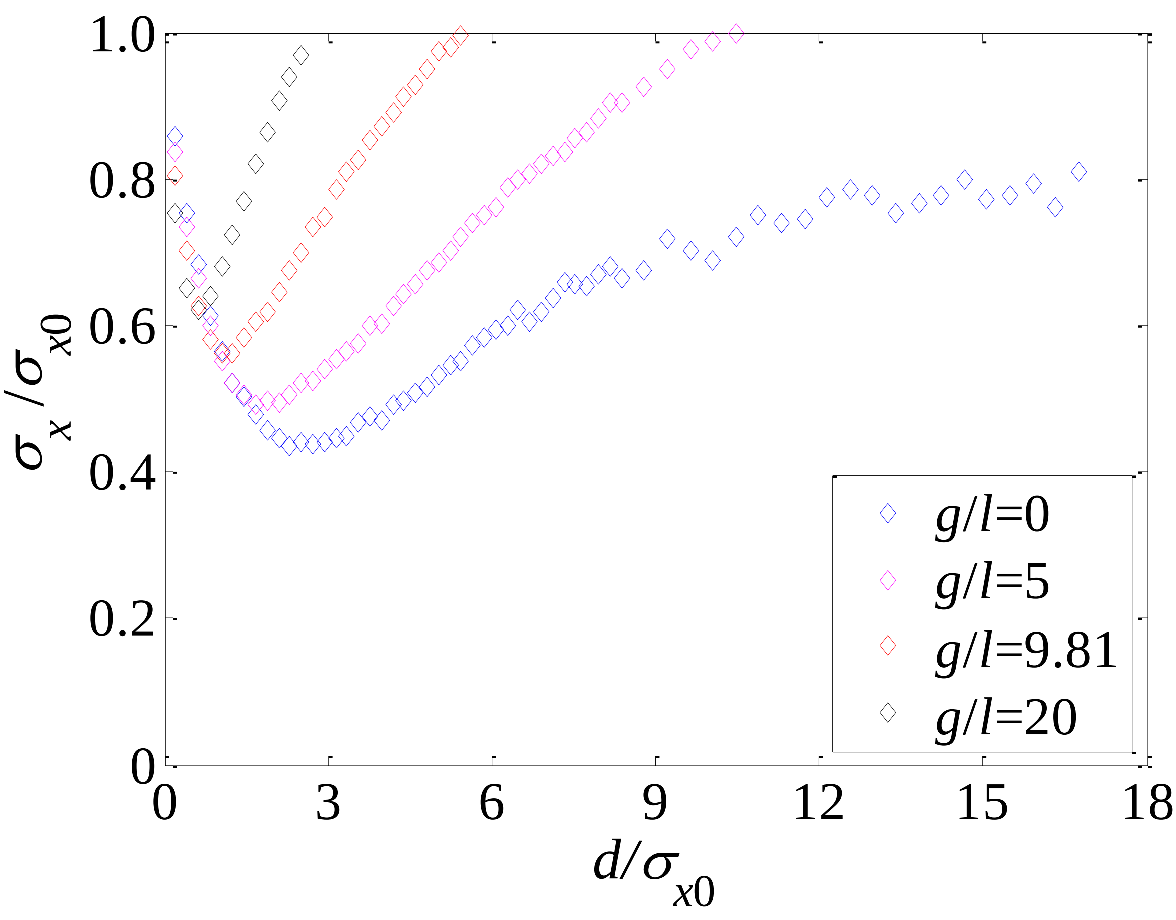


**Figure 7.** RMS response ratio of the primary system before and after damper installation - parametric study (double pendulum, $N$=2 with $\zeta$=0.01, $\zeta_2$ =0.10, $\omega_2/\omega_n$ =20, $\mu_k$ =0.1)

In Fig. 8 the performances of a single pendulum - vs. a multi-pendulum damper are compared. The higher numbers of particles in a multi-pendulum mitigate the vibration more efficiently, even though the mass ratio is the same. A multi pendulum with ten or more particles shows no significant decrement in the RMS levels; beneficial effects can exclusively be observed if a larger generalized gap ratio $d/\sigma_{x0}$ is used.

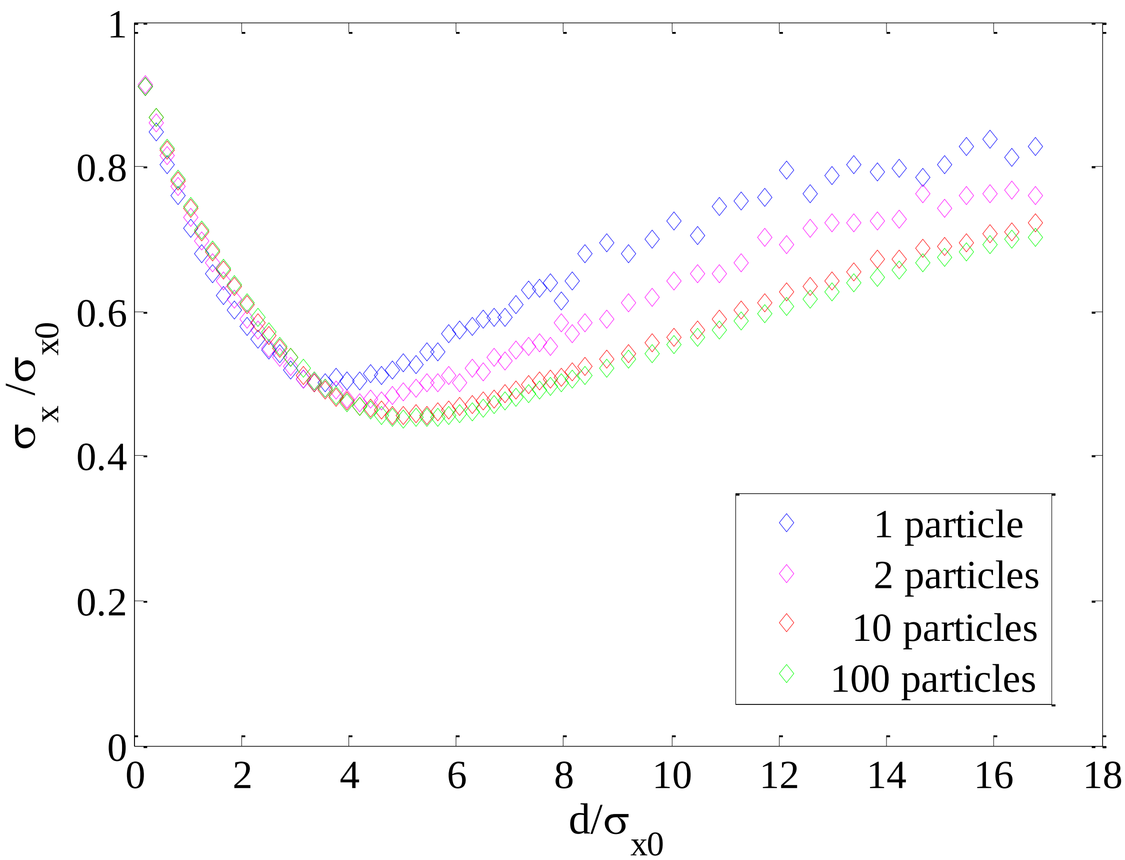


**Figure 8.** RMS response ratio of the primary system before and after damper installation - parametric study (comparison between single-pendulum and multi-pendulum impact damper, $N$=1÷100, with $\zeta$=0.01, $\zeta_2$ =0.10, $\omega_2/\omega_n$ =20, $\mu_k$ =0.1)

## 4. FULL-SCALE TEST

A full-scale test was carried out at Vienna University of Technology on a prototype as a proof of concept for the damping system. A post-tensioned girder with a length of 31.2 m served as reaction element for two stay cables. The beam was composed of post-tensioned precast elements (Fig. 5), used for creep and shrinkage tests, which were part of another research project investigating the balanced lift method for bridge erection (Kollegger et al, 2010). In order to avoid bending stresses in the beam, two stay-cables were used to induce axial stresses. The first cable was provided by a specialized company and corresponded to a simplified state-of-the-art stay-cable technology without the external duct, due to experimental reasons. The second cable was identical to the first cable, but it also included the new damping system, which was installed on the upper part of the stay.

Various tubes were used to obtain different damping values: a steel tube with 22.3mm interior diameter, two polyethylene high density (PE-HD) ducts with an inner diameter of 26.0 mm and 40.8mm. A “simple strand”, placed inside the steel tube, was used as prototype damper in the experiments. The strand is composed of seven steel wires of 5mm diameter with a tensile strength of 1860 MPa, which correspond to a conventional cable steel. Due to corrosion protection the wires are galvanized and covered with grease. The strand is sheeted with a HDPE and has a nominal diameter of 18 mm. For the tests with the (PE-HD)-duct, a PE-coated strand was used in order to avoid any corrosion. The taut cables were pre-stressed from 10% to 60% of the Guaranteed Ultimate Tensile Strength (GUTS) in 5 steps and with different measurements of acceleration at selected points along the stay were carried out during each step. The cable was excited by forced-vibration frequency sweeps and by manual excitations. Based on the dynamic response of the cables, the damping properties could be determined.

By comparing the results in frequency- and time-domain, the eigen-frequencies and the damping ratio of the first cable eigen-modes could be evaluated. The damping ratio of the cable without damping device in its first mode is approximately 0.1% (Fig. 7) at pre-stressing force of 1834 kN, which corresponds to a Eigen-frequency of 4.25 Hz. After installing the damper, damping ratios up to 0.8% (Fig. 8) were observed for the same mode.

**Figure 9**. Setup of the field experiment

**Figure 10**. Setup of the field experiment

**Figure 11**. Damper and shaker connected to the stay cable

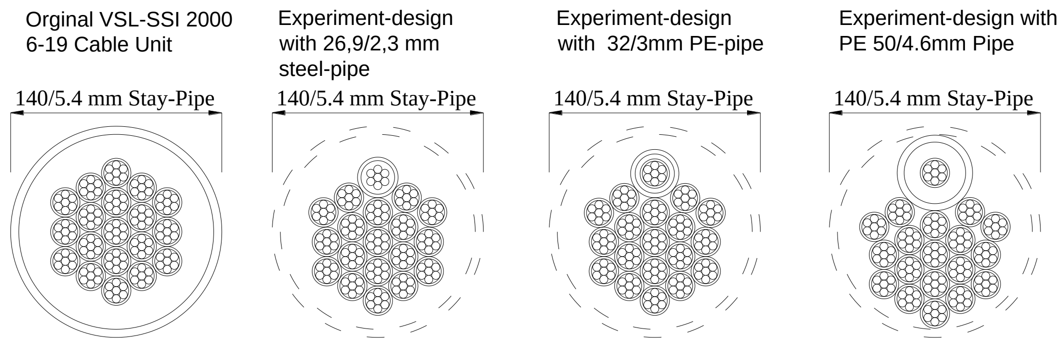


**Figure 12**. Setup of the different damping systems considered in the field experiment

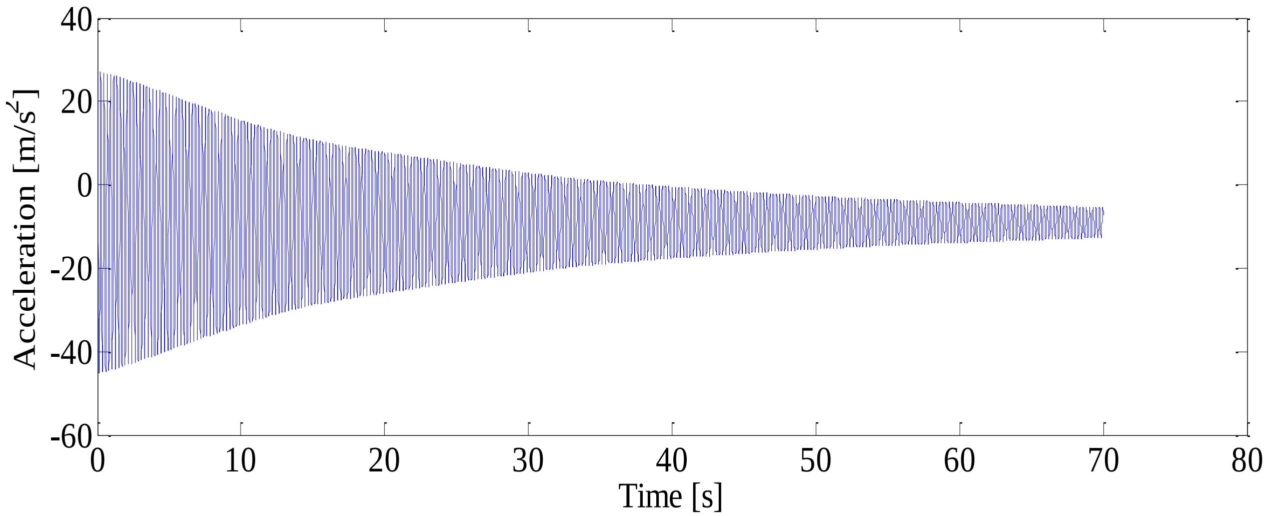


**Figure 13.** Free decay response without the integrated Damping System (First Eigen-frequency $f_1 \approx$ 4.25 Hz, Damping Ratio $\zeta_1 \approx$ 0.1%)

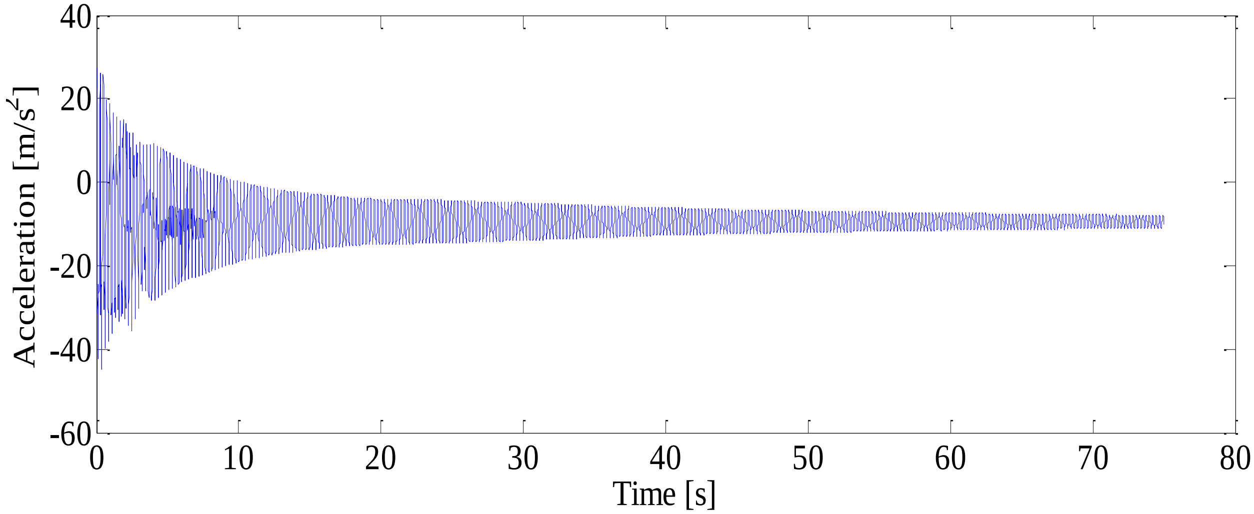


**Figure 14**. Free decay response with the integrated Damping System (First Eigen-frequency $f_1 \approx$ 4.25 Hz, Damping Ratio $\zeta_1 \approx 0.8\%$)

## 5. CONCLUSION AND FUTURE WORK

A numerical model for analyzing the performance of a recently-developed impact damper for reducing wind-induced cable-stay vibration has been described in this paper. The model is capable of reproducing the dynamic response of the fundamental modes of a stay-cable, influenced by incorporating the additional device. In parallel with the numerical studies, experiments were conducted to predict the attainable damping levels on a full-scale system. The experiments show that the damping device is effective in reducing dynamic vibrations.

Further Investigation is needed to study the mechanical behavior of the developed numerical model. Parametric studies, based on the proposed mechanical model, will be conducted to validate and compare the full scale field results. The placement of the damping system inside the stay is also advantageous because it facilitates the inspection along the cable. Also, no additional corrosion protection, contrary to the case of external dampers or cross-ties must be provided. Finally, the device has minimal visual impact on the structure, preserving the original esthetics of a cable-stayed bridge.

## ACKNOWLEDGMENTS

The study period of Mr. Philipp Egger as a Visiting Scholar in the Department of Civil and Environmental Engineering at Northeastern University (January - May 2012) was sponsored by the Austrian Marshall Plan Foundation. The full scale field tests were kindly supported by the Austria Wirtschaftsservice (aws) via the “Prize 2010 Prototype University Funding”.

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