



Implementation of active mass damping for wind-induced vibration mitigation: The Genoa Control Tower

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ABSTRACT

The newly constructed “Torre Piloti” in the Port of Genoa, a 65-meter steel control tower, is both a critical piece of infrastructure and a future architectural landmark. Owing to its slender geometry and exposed waterfront location, the tower is particularly susceptible to wind-induced vibrations. During the tower design phase, this raised concerns about occupant comfort, and prompted the need to implement effective vibration mitigation measures. This paper presents the development and implementation of an Active Mass Damper (AMD) system tailored to address the specific challenges identified. A simulation-driven design approach was adopted. Finite Element (FE) models, calibrated with wind tunnel test results and CNR-DT 207/2008 wind loading guidelines, were used to determine target damping levels. Time-history analyses guided the specification of AMD characteristics, such as mass, stroke, and actuator force, leading to the design of an inertial actuator powered by linear electric motors. A Multi-Modal Negative Acceleration Feedback (MMNAF) algorithm was developed to emulate ideal damping behavior across the structure’s fundamental modes. Four custom AMD units were installed within the tower’s upper structure, and a series of in-situ tests demonstrated that the control system effectively increased structural damping to the required levels, with measured responses closely matching numerical predictions. Ongoing monitoring via a distributed sensor network will support future analysis of the system’s performance under real wind events. This work demonstrates the successful application of advanced active control technologies in a challenging, high-performance structural context and provides a valuable reference for future implementations in similarly constrained environments.

1. Introduction

The Port of Genoa (Italy) has recently completed the construction of the new “Torre Piloti” (also referred to as “control tower” in this paper), a striking 65-meter (213-foot) tall steel structure conceptualized by renowned architect Renzo Piano [28] and designed, in different stages, by Buro Milan (<https://buromilan.com/>) (preliminary design) and Seteco (<https://www.setecogeo.it/>) (final design). The structure, shown in Fig. 1(a), is in the Levante Waterfront area of the Port (see Fig. 1(b)). This modern control tower was built to replace the original one, which tragically collapsed in 2013 following an accidental collision with the cargo ship “Jolly Nero” [1]. Hence, the new Torre Piloti not only serves as a critical component of port operations but also stands as a testament to innovative architectural design and engineering resilience.

The supporting structure of the Tower consists of four steel columns filled with concrete, spaced 3.6 m apart and interconnected by steel crosspieces. High-strength steel bars provide bracing on all four sides, enhancing the tower’s rigidity and limiting bending and displacement.

The control room’s floor features a fully glazed design supported by tapered cantilevered steel beams, which integrate X-Lam panels within the beam thickness. The roof (referred to as the “lens”) of the tower is a 30 × 30 m square, constructed with reticular beams that taper towards the edges, ultimately forming a continuous profile through welded plates.

The structure, shown in Fig. 1, is discussed in greater detail in the “Progetto Esecutivo” report [35], along with safety and functionality checks at the executive project level.

Because of its location and particular structural features, the elegant,

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slender structure is exposed to strong wind conditions. As a result, the structure underwent extensive investigations during the design phase, including computational fluid dynamics (CFD) simulations and wind tunnel testing [35]. These studies raised reasonable concerns regarding the levels of vibration that tower personnel might experience. Notably, given the most critical translational natural frequency of the structure (0.617 Hz, discussed in later sections), CNR regulations (CNR-DT 207/2008) specify a “comfort” threshold corresponding to curve (a) (office use) in Figure N.2 (reproduced below as Fig. 2). According to this figure, the allowable comfort acceleration is approximately 0.08 m/s^2 . However, wind tunnel results indicated that wind-induced accelerations at the control cabin level could reach as high as 0.58 m/s^2 [35].

Therefore, special consideration was given to the pronounced wind-induced vibrations, expected to exceed comfort acceleration limits. The challenge was to develop a solution capable of mitigating these vibrations effectively while accommodating for the structural and spatial constraints of the project.

The initial stages of the project involved studying the feasibility of using a passive control system, specifically a Tuned Mass Damper (TMD), to be installed inside the structure’s roof. It is well known that TMDs work by mounting a mass on damped springs, tuned to match the resonant frequency of the structure to mitigate vibrations. However, it was found that the characteristics required for this structure were incompatible with a TMD. The main challenges included the need for a large mass—around 30 tons—to be installed in a confined space, coupled with additional difficulties in managing accelerations due to the structure’s torsional vibration modes. These constraints made the TMD an impractical solution, highlighting the need to seek alternative approaches to effectively address the vibration issues.

A system based on Active Mass Dampers (AMDs) appeared to be more appropriate to address the problem at hand, hence a solution based on an AMD system was pursued. Unlike a TMD, an AMD involves a controlled, actively adjusted mass, where sensors and controllers continuously monitor and respond to structural vibrations by adjusting the position of the mass, which exerts the necessary control force in real-time. This dynamic adjustment allows AMDs to be more versatile and effective across a range of frequencies and conditions [26]. An overview of the working principles governing AMDs and the context in which they operate is provided in Section 2.

This paper presents the AMD system utilized to protect the Control Tower in Genoa from the dynamic effects of strong winds, detailing the design process from its preliminary stages to installation and field testing. It begins by discussing how the desired characteristics were achieved, followed by the final design and prototyping phase. The

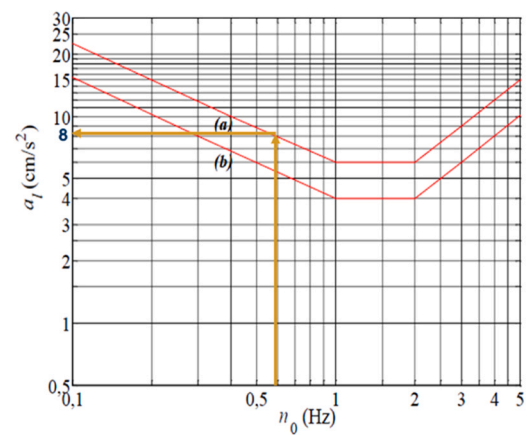


Fig. 2. Comfort acceleration level, TR = 1 year.

subsequent steps, including implementation, and field testing, are also examined.

Because AMDs are a relatively recent innovation in structural engineering, advanced non-linear dynamic analyses accounting for a broad spectrum of wind-generated loading conditions were initially undertaken. These analyses indicated that high performance could be achieved using just four compact AMD units, each with a mass ranging from 200 to 400 kg and capable of generating inertial forces up to 4 kN.

The final design of the vibration control system was achieved via close collaboration with ISAAC [18], the Milan-based company that patented, produced, and supplied the AMDs ultimately installed within the tower’s distinctive “lens” structure. Additional analysis and laboratory experiments were conducted before converging to the optimal solution. The final configuration comprised four “Electro-Pro” machines [17], each with a moving mass of 340 kg, a stroke of 1.8 m and a total length of 3 m. Central to this system is a sophisticated controller that, supported by a network of sensors distributed throughout the tower, monitors structural movement up to 1000 times per second. This real-time data allows the controller to adjust the positions of the active masses—driven by electric motors—to stabilize the tower and reduce sway, thereby minimizing structural stress. The system’s “active” nature is attributed to its reliance on dynamic adjustments made by the central controller and electric motors, ensuring optimal performance and stability.



(a) Photograph of the Control Tower



(b) Control Tower localization in the port of Genoa

Fig. 1. View of the structure.

2. Active mass dampers for vibration control: overview

The need to address the problem of dynamic loads on structures has fuelled great efforts in studying and developing Dynamic Vibration Absorbers (DVAs). Among passive DVA solutions, TMDs have proven to be an effective solution for controlling the dynamic response of structures subject to wind, earthquakes, and other vibration sources [12,20,22,25,37]. TMDs represent reliable cost-effective solutions and their employment has been broad over the past few decades, particularly for high-rise building applications. However, TMDs present some limitations, most notably concerning the narrow band around the tuning frequency in which they are effective.

To address this issue, and achieve greater flexibility and higher performance, semi-active and active DVAs have been devised in recent years [32,34,5]. Among recent passive/semi-active advances, triple friction pendulum-tuned liquid dampers integrate friction-pendulum mechanics with tuned liquid sloshing to absorb energy across a wider frequency band while maintaining mechanical simplicity and low maintenance [14]. Similarly, tuned inerter mass dampers, which add an inerter device to boost effective inertial response, enable compact, precisely tuned solutions with modest energy requirements suitable for constrained environments [24,39]. However, TMDs tend to require large mass and volume; tuned liquid and inerter-based systems, though more compact, are typically fixed-tuned and may underperform under evolving wind patterns or multi-modal, torsionally dominated responses. It should be noted that the Genoa Control Tower examined in this study presents torsional vibration modes and very limited roof-space above the control room, making conventional passive or semi-active devices problematic.

In this context, compact AMDs constituted an attractive solution thanks to their capability of actively generating recentring inertial forces using a relatively small mass, rather than relying on the mechanical coupling with the structure. Despite being the subject of many research studies and experimental programs AMDs remain a relatively recent innovation in the structural engineering field and have seen limited deployment compared to passive vibration control methods.

Koutsoloukas et al. [21], highlight that, among the 208 documented cases of DVAs installed in real-world buildings, only 8 belong to the AMD category, with no recorded applications in Western Europe. Furthermore, their study underscores the scarcity of scientific literature on the practical implementation of advanced vibration control systems. In this context, bridging the gap between academic research and industry becomes crucial. This paper is particularly significant as it documents the experience of the first major structure in Italy (and in Europe) to incorporate an active system for controlling wind-induced vibrations.

AMDs can be considered as an integral part of the structure,

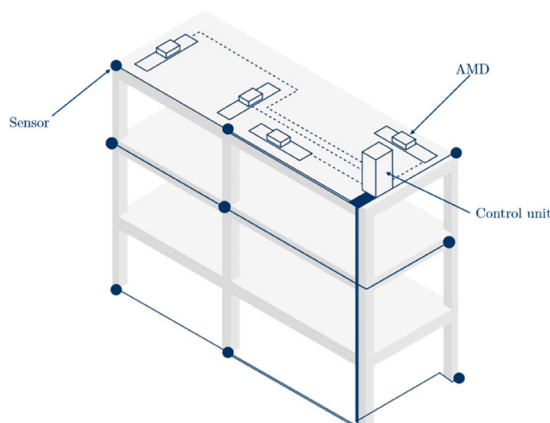


Fig. 3. Concept of a possible AMD configuration.

combining sensors and actuators connected to a central computer implementing the control algorithm. This is schematically outlined in Fig. 3.

Forces on the structure are exerted by inertial actuators, which produce the desired loads accelerating and decelerating a movable mass in response to the sensory inputs provided by accelerometers. To generate sufficient forces, a variety of actuating devices can be used, with hydraulic and electromechanical actuators being the most common solutions in structural engineering applications. The control force applied on the structure by each actuator is equal to the inertial force experienced by the moving mass, so its time/frequency characteristics depend on the algorithm implemented in the controller. An active vibration absorber is then characterized by the specific control law implemented, and many different solutions have been developed [30,31].

While feedforward strategies have also been developed, most studies have focused on feedback controllers ([3,4]). These are generally aimed at providing additional damping to the structure [29], either closing direct feedback loops on the structure velocity or enhancing the dissipation characteristics of one or more natural modes exploiting model-based techniques. The multi-modal Negative Acceleration Feedback (MMNAF) used in this project belongs to the second category, and it was designed to enhance the damping of the mode shapes dominating the dynamic response of the structure.

3. Numerical modelling of the structure

The dynamic characteristics of the structure were obtained through modal analysis of a 3D numerical model of the structure implemented in SAP2000 [7]. The model was constructed using "FRAME" elements to simulate the behaviour of all structural components (including ties and struts), assigning geometric and material properties as specified by the designers of the structure. Connections between BEAM elements were modelled as either continuous or pinned, according to the configuration defined during the design phase. The tower columns were fixed at the base, while all ties were connected to the ground with hinged constraints.

Since the primary objective was to realistically represent the structure's dynamic characteristics (rather than precisely model all elements to estimate the load demands on individual components), the roof (i.e., the lens), the control room roof, and the control room floor were modelled as equivalent SHELL elements to correctly simulate the masses at various heights in a simplified manner. For the same reason, the cantilevered staircase and the elevator were not explicitly modelled but were instead included in the model as equivalent masses. The total estimated mass of the structure was approximately 370 tons, distributed among the lens (191 tons), the control room (47 tons), and the tower (135 tons). Finally, a constant viscous damping ratio of 0.3 % was estimated and assigned to all vibration modes. This damping ratio represents a lower-bound estimate and is consistent with the value considered by the designers of the tower [35].

The first three vibration modes of the structure obtained from the analysis of the described model are presented in Fig. 4, and the values of the vibration periods and fundamental frequencies are reported in Table 1. It should be noted that the modal shapes and fundamental periods of the structure identified in this study are essentially identical to those outlined in the original designers' calculation report [35], which listed modal period values $T_1 = 3.79$ s, $T_2 = 1.62$ s, and $T_3 = 1.58$ s, identified through independent finite element model analysis.

4. Equivalent dynamic wind load calibration

In addition to a realistic numerical model of the structure, carrying out the design of a suitable AMD system via time history analysis required the definition of a dynamic input, representative of the expected wind loads, to be used in the numerical simulations. Details

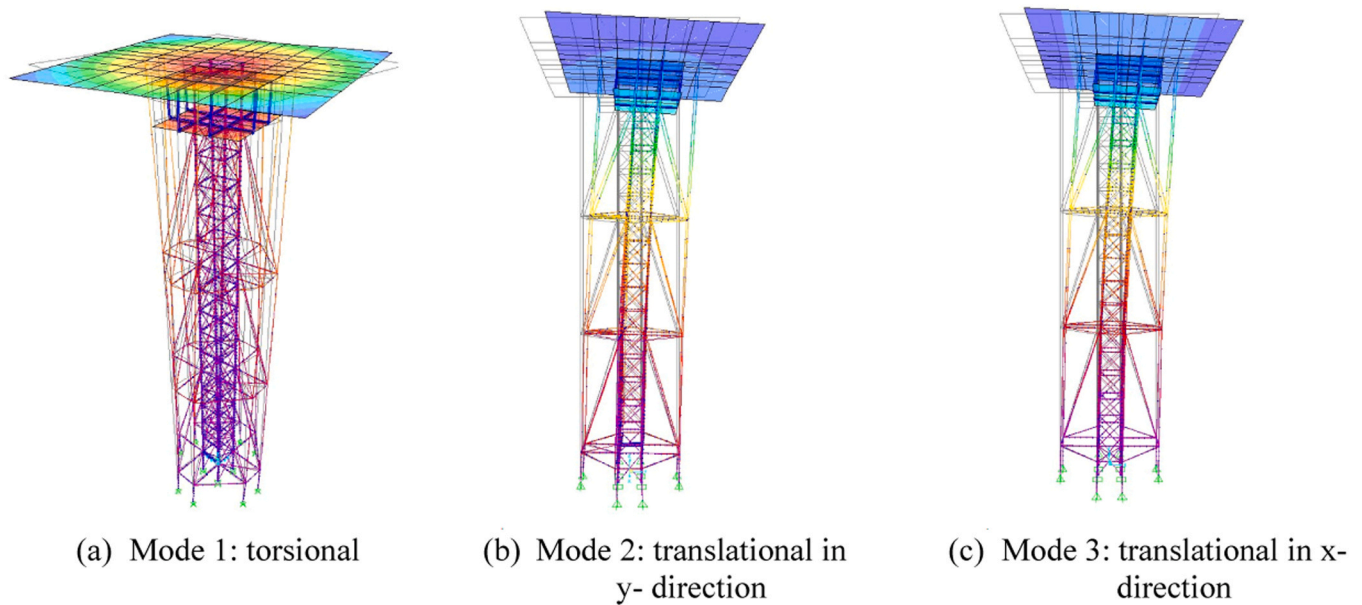


Fig. 4. Vibration modes of the structure.

Table 1

Fundamental periods and frequencies of the structure from the numerical model.

Mode number	Period [s]	Frequency [Hz]	Damping [%]
1	3.773	0.265	0.3
2	1.617	0.617	0.3
3	1.577	0.634	0.3

pertaining to this aspect are provided in this section.

4.1. Dynamic input definition

Modelling wind loads and their interaction with structures presents significant challenges due to the complex, turbulent nature of wind and the dynamic response of structures. Accurately capturing this interaction often requires sophisticated computational fluid dynamics (CFD) simulations and wind tunnel experiments. Complex phenomena include turbulence modelling, inflow boundary conditions, and near-wall treatments in numerical evaluations of wind loads on buildings [9]. Additionally, studies have employed CFD simulations to investigate the effects of nonlinear structural responses from extreme wind loads on aerodynamic feedback [11]. Notable contributions in this field have been made by researchers like Kareem (among many others), who has extensively studied wind-structure interactions and developed advanced modelling techniques (e.g., [6,10,13,19]).

In the present study, advanced aerodynamic modelling was beyond scope because the primary objective was to evaluate the relative performance of the proposed AMD system under realistic wind-induced accelerations, rather than to perform a comprehensive site-specific wind climate analysis. The target acceleration levels used to generate the equivalent loads were obtained from existing wind tunnel test results [35] that considered site-specific wind conditions [33], ensuring that the excitation magnitudes reflected realistic aerodynamic behaviour and already incorporated relevant aerodynamic phenomena.

Therefore, the wind load input for the time history analysis was generated by producing a random signal (i.e., “white noise”) and filtering it through a Von Kármán spectrum to simulate turbulence, following the approach of Beal [2]. A load duration of 1800 s (30-minute) was selected, in line with durations used in previous studies such as Minciarelli et al. [23], who analyzed 30-minute segments extracted from multi-hour wind records. The normalized force generated following this

process is shown in Fig. 5. It should be noted that this signal has a mean magnitude of zero, as the wind’s turbulent component primarily affects the tower’s vibration dynamics and consequently its accelerations.

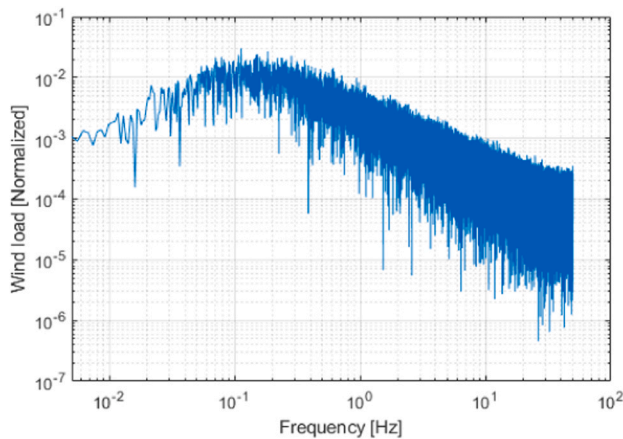
The normalized wind force was subsequently scaled in the numerical simulations to produce peak control room accelerations matching those measured in the wind tunnel tests (i.e., 0.58 m/s^2). It should be noted that these acceleration magnitudes are consistent with the maximum values specified by CNR-DT 207/2008 [8] for a nominal 200-year wind return period (i.e., 0.6 m/s^2). Accordingly, a load input corresponding to wind-induced accelerations representative of a 200-year return period was adopted as the target design input. To assess the performance of the AMD system under more frequent wind events (hence lower wind-induced accelerations), additional analyses were conducted using load inputs scaled to correspond to CNR-DT 207/2008 wind events with return periods of 1 year and 10 years.

Throughout this paper, for clarity and to facilitate discussion of the results, references to input magnitudes will be made in terms of the nominal CNR-DT 207/2008 return periods (1, 10, and 200 years). As mentioned, the 200-year return period event defined in CNR-DT 207/2008 is representative (with a slight margin on the conservative side) of the wind tunnel results.

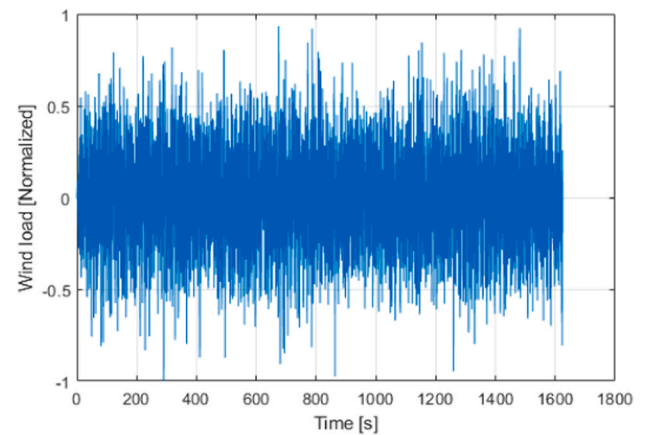
4.2. Wind force scaling

The wind-induced peak accelerations estimated at the control room level via CNR-DT 207/2008 guidelines are summarized in Table 2. Note that for all return periods, T_R , considered (i.e., 1, 10 and 200 years), CNR regulations require the longitudinal, transverse and torsional accelerations to be considered simultaneously in wind force generation and subsequent design phases.

Given the complexity of estimating a realistic distribution of representative dynamic forces (in terms of amplitude, duration, frequency content, and distribution) applied along the height of the structure [15], equivalent concentrated wind forces were applied at the top of the Control Tower to produce the desired dynamic effects. A set of two pairs of dynamic forces were applied simultaneously in two orthogonal directions, at the nodes labelled F1 to F4 in Fig. 6. The selected nodes are located on the perimeter of the tower control room and correspond to the locations tentatively selected to install the four mobile masses comprising the AMD system to be designed. This will be discussed in more detail in later sections of the paper.



(a) Normalized force response spectrum



(b) Normalized force time history

Fig. 5. Normalized wind force.

Table 2

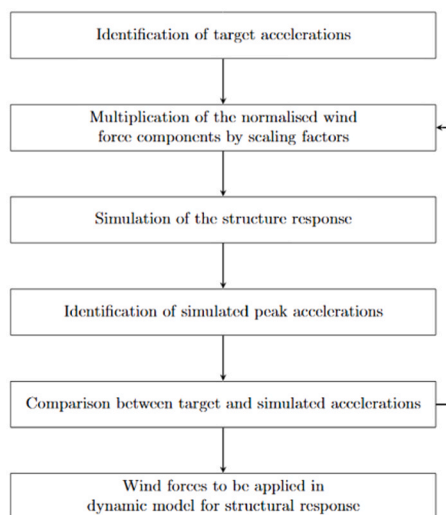
Expected Peak Accelerations According to CNR Regulations.

T_R [years]	Longitudinal acceleration [m/s ²]	Transverse acceleration [m/s ²]	Torsional acceleration [rad/s ²]
1	0.13	0.16	0.005
10	0.22	0.29	0.007
200	0.39	0.60	0.011

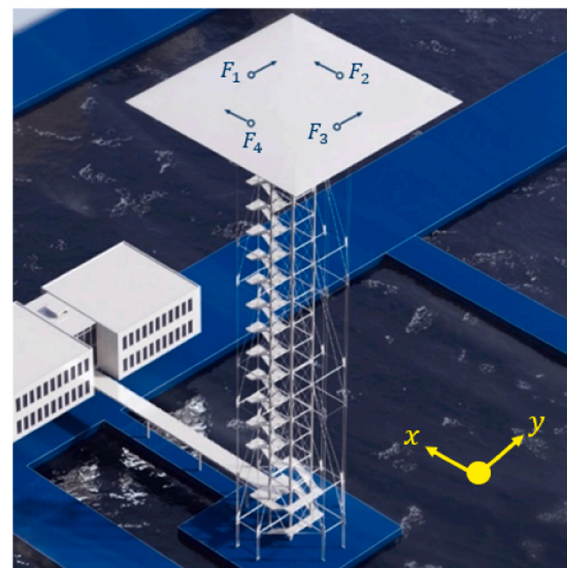
Hence, equivalent wind forces were obtained by scaling the normalized input generated in Section 4.1 to produce the larger of the acceleration components summarized in Table 2, i.e. the one along the transverse direction, along each axis. The process followed for calibrating the wind forces is outlined in Fig. 6, and can be summarized as follows:

1. The target acceleration is identified, and the normalized wind force defined in Section 4.1 is applied at the top of the structure as a concentrated load at the four selected nodes, in two orthogonal directions.
2. Scale factors are used to amplify the applied normalized wind forces and the structure, with inherent damping ratio of 0.3 %, is analysed in dynamic time-history mode. Peak accelerations are recorded at the control room level, at the four measurement points (Fig. 6, b).
3. The recorded peak accelerations are compared with target accelerations derived from Table 2. If the values match, the wind forces are correctly identified and are used as input to carry out the design of the AMD system, as described in the next section; otherwise, the scaling of the forces is modified, and the process is repeated until convergence is achieved. Optimal scaling of the wind forces was achieved via non-linear least squares analysis.

It should be noted that more rigorous methods for wind load



(a) Process to generate equivalent wind forces



(b) Nodes of application of equivalent wind loads and acceleration measurement points in the numerical model

Fig. 6. Process for generating equivalent wind forces for time history analysis.

derivation (such as those outlined in Annex G9 of CNR-DT 207/2008) exist that account for aerodynamic admittance, along-wind and cross-wind spectral separation, and spatial coherence effects. However, these were not adopted in the present study. The simplified loading approach described above, based on filtered white noise scaled to realistic acceleration levels from wind tunnel data and code values, provided a broadband turbulence-like excitation consistent with expected structural response characteristics while avoiding the significant modelling effort of full spectral decomposition. It is important to emphasize that, in the absence of wind tunnel results, more refined procedures should have been considered.

5. AMD design and prototyping development

The AMD system was designed considering a reference uncontrolled acceleration of 0.60 m/s^2 at the control room level. As discussed, this value corresponds to the maximum transverse acceleration permitted under the CNR-DT 207/2008 provisions for a nominal 200-year wind, and is consistent with peak accelerations measured in the wind tunnel tests.

Based on this conservative design approach, the preliminary features of the AMD system were first determined based on dynamic time-history analysis as outlined in this section, with the objective of reducing the peak acceleration in the control room of the tower to the identified comfort limit of 8 cm/s^2 .

The detailed design of the AMD components and prototype development and implementations was then carried out, as discussed in Sections 5.2 and 5.3.

5.1. Characterization of AMD target performance via time history analysis

The identification of suitable properties for the AMD system was based on the following iterative procedure:

1. The AMD system was first added to the numerical model of the structure presented in Section 3. The proposed AMD system consisted of four machines located in the same nodes used to introduce the introduce wind forces in the structure (Fig. 6, b). The initial AMD system's characteristics were estimated based on the structure's dynamic properties and a desired equivalent damping generated by the AMD system needed to generate (e.g., $\xi = 0.1$).
2. The numerical model, equipped with the AMD system, was analysed in dynamic time-history mode, using the wind forces described in the previous section. Peak accelerations at measurement points were recorded and compared with the values identified as comfortability limits (i.e., $a_{\text{peak}} = 8 \text{ cm/s}^2$).
3. If the comfortability thresholds were met, the control forces of the AMD system were extracted from the analyses, providing the reference duty cycle necessary to determine, in the subsequent design phase, the AMD actuators technical characteristics, the inertial mass and the chassis stroke. If comfortability thresholds were not met, the characteristics of the AMD system were modified, to generate a higher level of equivalent damping, and the process was repeated until convergence.

As discussed by Rebecchi et al. [27], each AMD was modelled as a two-node uniaxial equivalent viscous damper, with one node connected to the roof of the structure and the other node fixed in space (as shown in Fig. 7).

The AMD equipped structure was analysed in domain via NonLinear Modal History Analysis (Fast Nonlinear Analysis, FNA) [16] based on Ritz Vectors [36].

In the first iteration, the equivalent viscous dampers were assigned a damping constant “G” (also referred to as “Gain”), calibrated to provide the desired equivalent damping with respect to the first translational

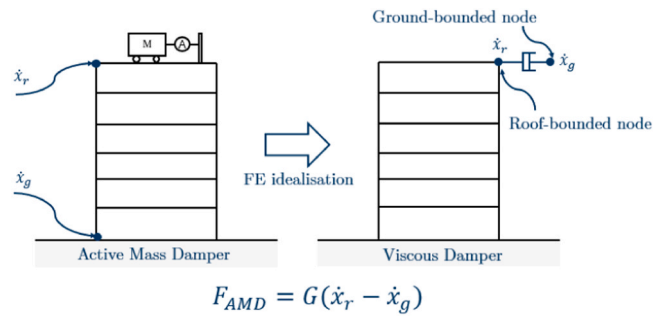


Fig. 7. AMD numerical modelling strategy.

mode of vibration of the structure.

For instance, given to target equivalent damping ratio of ξ^* starting from the uncontrolled damping ratio of ξ_0 , the gain value G was computed as

$$nG = 2M_1\omega_1(\xi^* - \xi_0) \quad (1)$$

where M_1 and ω_1 are respectively the mass and angular frequency of the first translational mode, which is dominant in the structural response; n represents the number of actuators acting on the mode shape, which is equal to 2 for the translation modes and to 4 for the torsional modes.

In subsequent iterations, it was found that assigning a constant $G = \frac{50kNs}{m}$ to each damper could reduce the wind-induced accelerations experienced at the control cabin level below acceptable thresholds. The viscous dampers effect on the structure is to bring the damping ratio of each mode from a value

ξ_0 , which stands for the structural damping of the uncontrolled structure, to a value

ξ^* . For the first three modes, the values of ξ^* , shown in Table 3, represent the design target that the active control systems must meet to ensure that the tower dynamics will behave consistently with the numerical simulations.

The results of the numerical analyses conducted on the structure equipped with the identified viscous dampers (intended to mimic the effects of an equivalent AMD system), are presented in terms of time histories, in Fig. 8, while Fig. 9 shows the corresponding RMS values evaluated on a moving window of 5 s, with the red dashed plot in the graph representing the region beyond comfort threshold. Only the results associated with a return period of 200 years are shown. The response for return periods of 1 year and 10 years appear qualitatively identical but scaled down proportionally to the force amplitude. The extremely significant impact of the AMD system on wind-induced acceleration is evident from the figures below, which show the response of the tower without additional damping (blue curve) and equipped with the AMD system (red curve).

The results of the analyses are also shown in Table 4, in terms of peak accelerations for both controlled and the uncontrolled scenarios. The peak acceleration remains well below the 8 cm/s^2 threshold associated with the 1-year return period, which is the reference provided by the CNR for comfort standards. For longer return periods, the peak values do exceed the comfort threshold; however, as illustrated in Fig. 9, these exceedances occur only at isolated time instants, even under such extreme conditions.

From the simulations relative to the 200 years return period, the

Table 3
Target modal damping values.

Mode	ξ_0	ξ^*
1	0.3 %	6.3 %
2	0.3 %	4.4 %
3	0.3 %	4.0 %

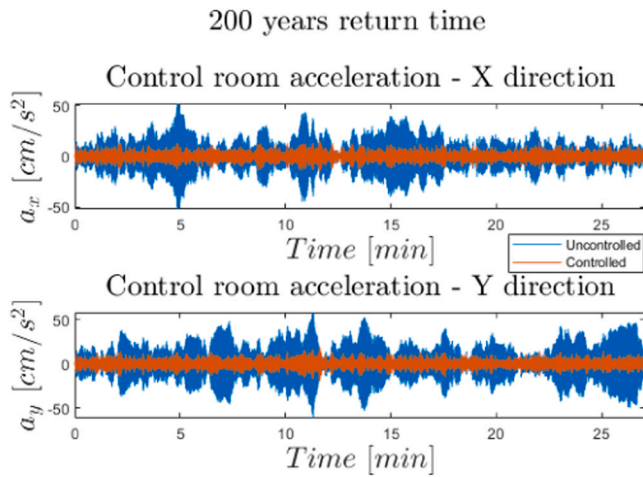


Fig. 8. Time history results for the structure without additional damping (blue) and with the AMD system (red) considering wind forces from CNR regulations, for $T_R = 200$ years.

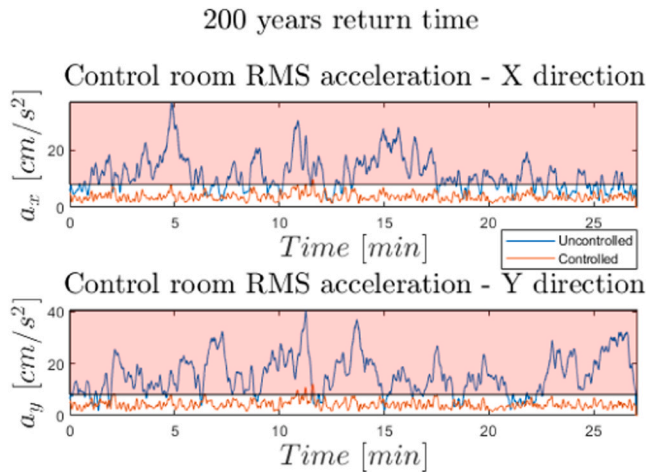


Fig. 9. RMS results evaluated on a moving window of 5 s for the structure without additional damping (blue) and with the AMD system (red) considering wind forces from CNR regulations, for $T_R = 200$ years.

force time histories associated to each damper were extracted (Fig. 10): the most demanding profile was chosen as the reference duty cycle for the AMD requirements.

From the simulation data, it was possible to identify the requirements for the AMD actuator: the signal is characterised by a peak force of 2600 N and a 5 s RMS force equal to 1560 N. These values univocally define the AMD in terms of force, while velocity, stroke and power consumption are dependent on the value of the inertial mass, as discussed later.

5.2. AMD actuator design

The actuator is the component devoted to the generation of the inertial forces used for controlling the structure. The target force profiles shown in Section 5.1 present relatively modest demands, compatible with a broad range of actuators. To ensure maximum reliability while minimizing maintenance needs, the search for a suitable actuator was restricted to linear motors, which are characterised by the absence of contact between stationary and moving parts, avoiding wear problems associated with transmission mechanisms required to convert rotary motion into linear motion. Table 5 summarizes the force-velocity characteristics of the selected motor in relation to the identified requirements: motor peak force (which can be sustained for a duration of 1 s) is sized to exceed the instantaneous maximum of the duty cycle, while the continuous force (which can be sustained indefinitely without causing overheating) is compared with the target RMS force.

The operational limits of an AMD, however, are not defined by the force characteristics of the actuator alone, but also by the velocity and displacement saturation limits, both of which are influenced by the inertial mass parameter. A larger mass poses additional installation challenges due to its size and weight, while a smaller mass necessitates more powerful actuators with larger strokes to achieve the required performance. In this case, the dimensioning was performed with the aim of reducing the weight of the AMD as much as possible. Following this observation, it was necessary to find the minimum value of mass M that satisfies constraints on both velocity and stroke:

$$M = \max(M_V, M_S) \quad (2)$$

where M_V is the minimum value of mass preventing speed-related voltage saturation of the electric motor, and M_S is the minimum mass which results in displacements compatible with the available stroke.

The velocity profile of the AMD moving mass can be evaluated by integration of the force profile of Fig. 10 as:

$$v(t) = \frac{1}{M} \int_0^t F(\tau) d\tau \quad (3)$$

by imposing a tentative value for the inertial mass M . Iteratively, it is possible to vary the mass value until the trajectory of the actuator in the force-velocity plane falls within the boundaries of the motor “admissible region”. The convergence process can be accelerated by making a well-informed guess based on the frequency characteristics of the reference force signal: from Fig. 10, which shows the FFT of the required damper force, it can be noticed that the force request appears concentrated around the frequency of the first bending mode (0.61 Hz in X direction). This is in line with expectations, given the filtering effect of such a slender and lightly damped structure. Hence, a simplified analysis can be carried out considering the duty cycle of the actuator as single harmonic force profile.

If the AMD action is reduced to a single harmonic component, velocity and force are related as follows:

$$\begin{cases} F(t) = F_0 \sin(\omega_0 t) \\ v(t) = -\frac{F_0}{M\omega_0} \cos(\omega_0 t) \end{cases} \quad (4)$$

Table 4

Accelerations recorded from numerical analyses at the control room level of the AMD equipped Tower.

Direction	$T_R = 1$ year		$T_R = 10$ years		$T_R = 200$ years	
	UNCONTROLLED a_{peak} , [cm/s ²]	CONTROLLED a_{peak} , [cm/s ²] (-74 %)	UNCONTROLLED a_{peak} , [cm/s ²]	CONTROLLED a_{peak} , [cm/s ²] (-74 %)	UNCONTROLLED a_{peak} , [cm/s ²]	CONTROLLED a_{peak} , [cm/s ²] (-75 %)
X	16.0	4.1	29.0	7.5	60.0	15.1
Y	16.0	5.4 (-66 %)	29.0	9.8 (-66 %)	60.0	19.6 (-67 %)

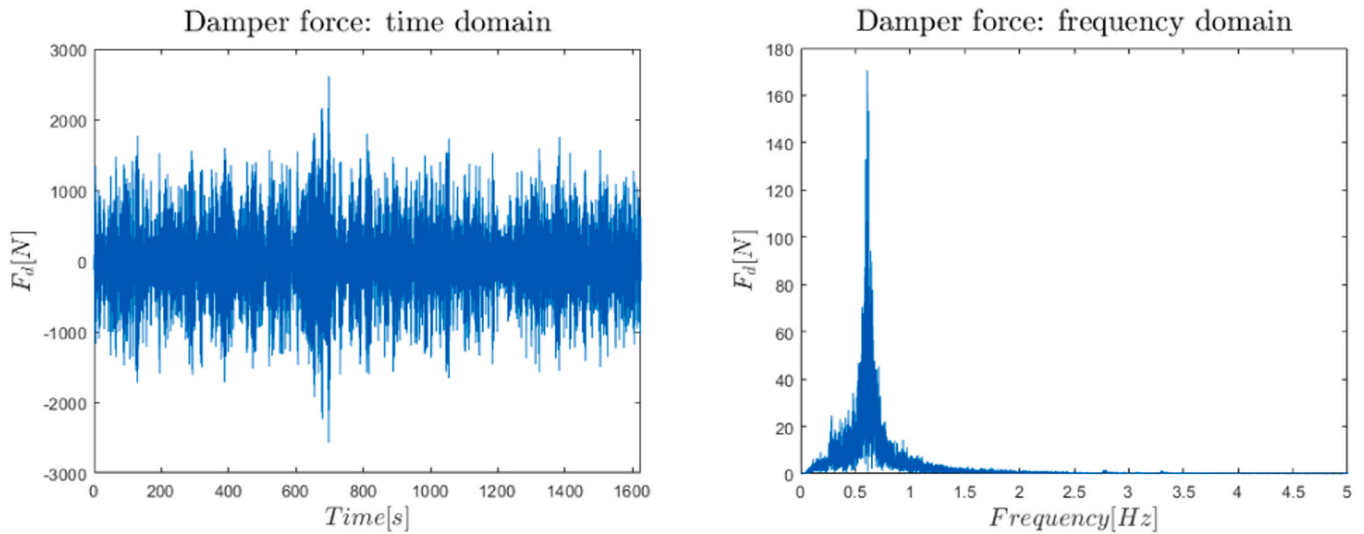


Fig. 10. Damper force vs time (left) and damper force vs frequency (right).

Table 5

Force-velocity characteristics of the selected motor.

	Motor characteristics	Requirements
Peak force [1 s]	5000 N	2600 N
Continuous force	1979 N	1560 N
Maximum velocity at maximum force	2.6 m/s	Mass dependent
Free velocity	3.1 m/s	Mass dependent

Where F_0 represents the amplitude of the reference sinusoidal force, set equal to the maximum expected force, and ω_0 is the angular frequency, coinciding with that of the structure first bending mode. Rearranging Eq. (4), the graph of an ellipsis is found in the Velocity-Force plane:

$$F^2 + M^2 \omega^2 v^2 = F_0^2 \quad (5)$$

The branch of motor curve affected by velocity saturation can be described by the following linear relationship in the Velocity-Force plane:

$$F = mv + q \quad (6)$$

The minimum value of mass M_v , necessary to avoid velocity saturation can be found by imposing the tangency between the two curves, which yields the equation:

$$M_v = c_v \sqrt{\frac{-m^2 F_{0i}^2}{\omega^2 (F_{0i}^2 - q^2)}} \quad (7)$$

Where c_v is a safety factor designed to accommodate higher velocities with respect to those reached by the single harmonic approximation, creating a buffer region; the use of this coefficient, whose value has been set to 1.3, is equivalent to imposing tangency of the ellipse on a shifted version of the velocity saturation line, called “reduced velocity saturation”. The solid lines in Fig. 11 represent the motor limits in terms of maximum force and velocity, while the dashed lines represent the maximum allowable continuous force and the reduced velocity saturation limits; the blue ellipsis is the simplified single-harmonic duty cycle with mass M_v .

The evaluation of the AMD velocity using Eq. (3), with a mass $M_v = 285$ kg found with Eq. (7), leads to the red trajectory of Fig. 12: almost all points lie within the single-harmonic approximation, with sporadic outliers that remain within the boundaries of the motor capabilities.

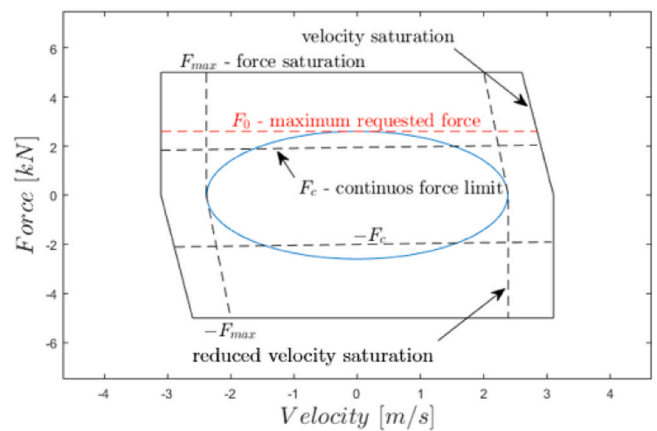


Fig. 11. Graphical representation of motor limits in the force-velocity space.

A second constraint on minimum mass arises from the necessity of avoiding stroke saturation: given the dimensions of the niche reserved for each AMD and considering practical issues pertaining to safe installation and access to the device, a maximum physical stroke equal to 2.2 m was defined (1.1 m per direction). To ensure extra-stroke capacity in case of extreme conditions, the admissible displacement for the in-

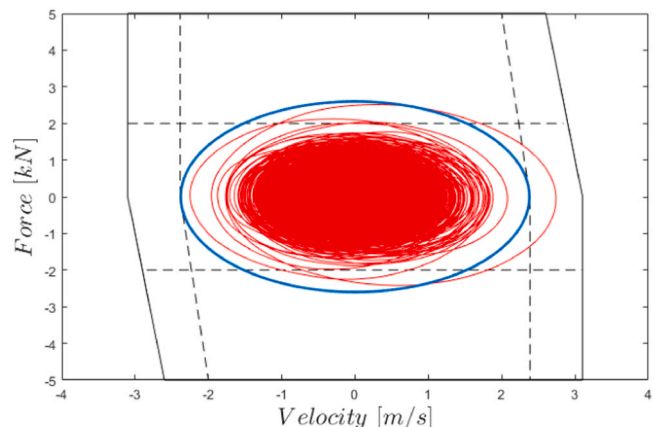


Fig. 12. AMD response in the force-velocity space.

ertial mass was restricted to ± 0.9 m. As a first step, value M_V was evaluated to ensure it was compatible with the stroke requirements, by analysing the time history of the inertial mass displacement via double integration. The computed profile was found to exceed the limits, suggesting that the stroke requirement imposes the greatest demand (i.e., $M_s > M_V$). Thus, M_s was obtained by adjusting its value to fit the AMD displacement time history within the allowable range, resulting in a final mass value of 340 kg. The displacement time history response of the AMD associated to the chosen design parameter is shown in Fig. 13.

Starting from the chosen linear motor and the defined mass and stroke, a mechanical design for the prototype was developed, resulting in the machine outlined in Fig. 14.

The overall mass of each device, considering both the inertial mass and the supporting frame, totalled 500 kg. Therefore, considering all four AMDs, the total mass added to the tower was about 2 tons, corresponding to 0.5 % of the weight of the structure.

5.3. AMD control system design

To obtain the desired additional structural damping, a Multi-Modal Negative Acceleration Feedback algorithm was implemented, in a form analogous to that proposed by Yang et al. [38]. This technique was preferred to others for the following reasons:

- It provides a straightforward method to close a feedback loop exploiting common measurement technologies, i.e., accelerometers to measure the structure's motion and linear encoders to control the motor displacement.
- The uncontrolled structure is lightly damped, leading to a narrow frequency band in the dynamic response.
- The wind-structure interaction, although capable of causing acceleration levels potentially incompatible with human comfort, is not expected to cause stiffness degradation, therefore the tower natural frequencies can be treated as time-invariant.

The Multi-Modal Negative Acceleration Feedback (MMNAF) strategy was selected as the most suitable compared to alternatives. Unlike Sky-Hook control, which broadens the frequency band at the cost of increased sensitivity to low-frequency noise and excessive AMD stroke demands, MMNAF effectively addresses the narrow-band response of this lightly damped steel structure. Similarly, the added complexity of LQR control was not evaluated to bring significant performance improvement in face of the increased computational cost.

The control architecture is schematically outlined in Fig. 15, where

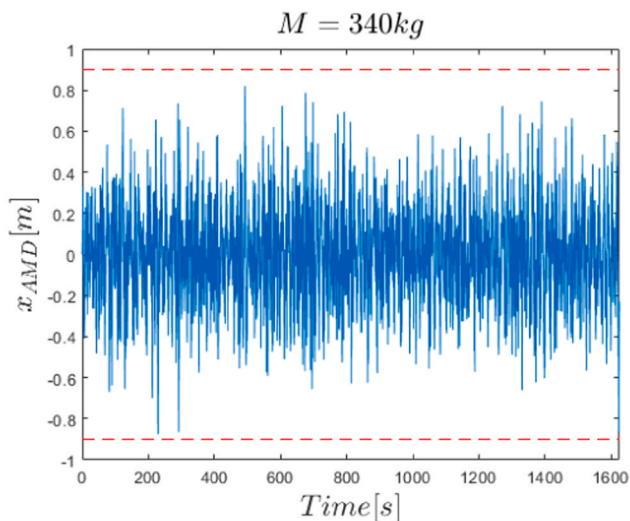


Fig. 13. AMD response in terms of displacement vs time.

the coloured blocks represent functions implemented in the control algorithm, while the white blocks are representative of physical processes.

The inner current loop of the motor, in combination with the position control law, effectively compensates for the minor frictional effects arising in the linear guides of the moving mass. Consequently, an accurate modelling or estimation of such nonlinear phenomena is not required, and the overall effect of the system on the structure can be derived from the nominal displacement $\mathbf{u}$ as shown in function blocks of Fig. 15.

The system behaviour is dependent on two inputs, the wind forces $\mathbf{F}_w$ and control forces $\mathbf{F}_C$.

Term G_{str} represents the dynamics of the tower, which can be expressed in physical coordinates as:

$$[M]\ddot{\mathbf{x}} + [C]\dot{\mathbf{x}} + [K]\mathbf{x} = \mathbf{F}_w + \mathbf{F}_C \quad (8)$$

Where $[M]$, $[C]$ and $[K]$ represent the mass, damping and stiffness matrices.

The same equation can be converted into modal coordinates as:

$$\ddot{\mathbf{q}} + 2[\Xi][\Omega]\dot{\mathbf{q}} + [\Omega]^2\mathbf{q} = \mathbf{Q}_w + m_{AMD}[\Phi]^T[B]\ddot{\mathbf{u}} \quad (9)$$

where matrices $[\Xi] = \text{diag}(\xi_i)$ and $[\Omega] = \text{diag}(\omega_i)$ collects the modal damping ratios and the natural frequencies respectively, obtained by the FE model (Section 3) and subsequently recalibrated on the exact output of Experimental Modal Analysis (Section 6.2.1, Table 7). The excitation term related to the AMDs is expressed as a function of the inertial masses acceleration vector $\ddot{\mathbf{u}}$, the transpose of the modal matrix $[\Phi]_{n \times n_c}^T$, and a selection matrix $[B]_{n \times n_c}$ used to insert the AMD forces in the corresponding structure nodes, with n number of structural nodes and n_c number of control nodes, i.e. the points where the actuators are located.

Vector $\ddot{\mathbf{x}}_C$ collects the acceleration of the control nodes, where the feedback accelerometers are also installed. These measurements are fed to the control algorithm, which evaluates the modal accelerations relative to the modes to be controlled by pre-multiplying them with the pseudoinverse of matrix $[\Phi_C]$, defined as the partition of the modal matrix $[\Phi]$ accounting only for the m modes considered and the n_a . Matrix $[\Phi]$ was derived from the FE and eventually validated by the experimental results. For each mode i , the modal control action $Q_{C,i}$ is then computed from the modal acceleration $\ddot{q}_i$ via the transfer function:

$$\frac{Q_{C,i}(s)}{\ddot{q}_i(s)} = \frac{-g_i}{s^2 + 2\xi_{c,i}\omega_i s + \omega_i^2} \quad (10)$$

where g_i is the gain chosen for the mode (computed from Eq. 1, following the iterative procedure discussed in Section 5.1), ω_i is the mode natural frequency and $\xi_{c,i}$ is the damping ratio of this second order controller. Since the control law is evaluated separately for each targeted mode, the value of ω_i used in Eq. (10) is that mode's own identified natural frequency.

$Q_{C,i}$ can be seen as the displacement of a virtual AMD mass acting along modal coordinate q_i . Multiplying vector $\mathbf{Q}_C$ by matrix $[\Phi_C]$, the physical displacement of each mass u_i is evaluated as the superposition of the effects of each mode. The resulting inertial forces represent the control input $\mathbf{F}_C$ to the structure.

The MMNAF controller requires knowledge of the modal parameters of the structure: during the design phase, the developed FEM model provided such information, which was eventually refined exploiting the data collected by testing the real structure.

Focusing on single mode, the absolute value of the harmonic transfer function from the disturbance force to the modal response for a NAF controller evaluated in correspondence of the natural frequency ω_i can be expressed as:

$$|H_{NAF}(\omega_i)| = \left| \frac{q_i(j\omega)}{Q_w(j\omega)} \right| = \frac{1}{2 \left(\frac{m_a g}{4m_i \xi_{c,i}} + \xi_i \right) \omega_i^2} \quad (11)$$

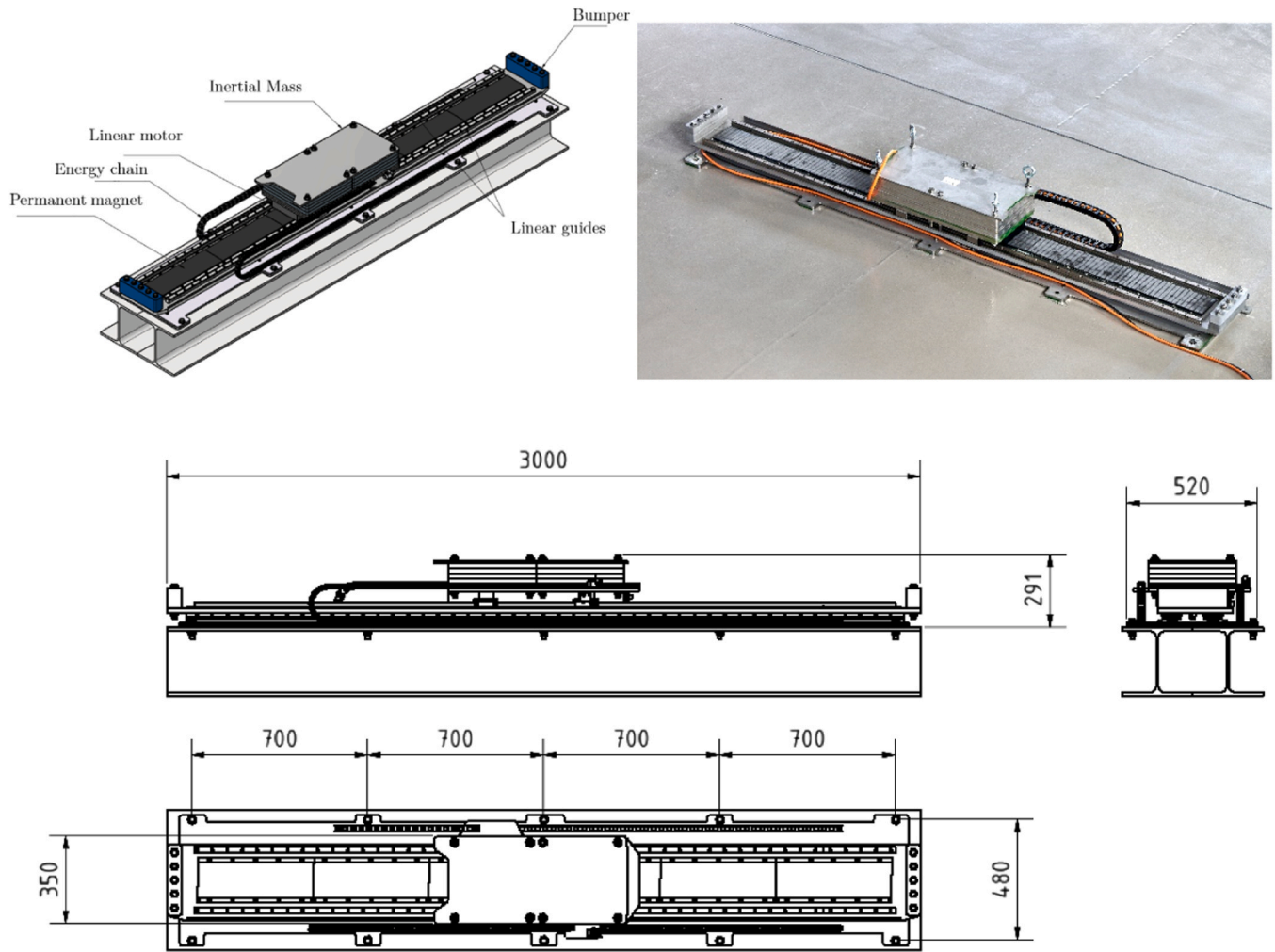


Fig. 14. View of AMD prototype.

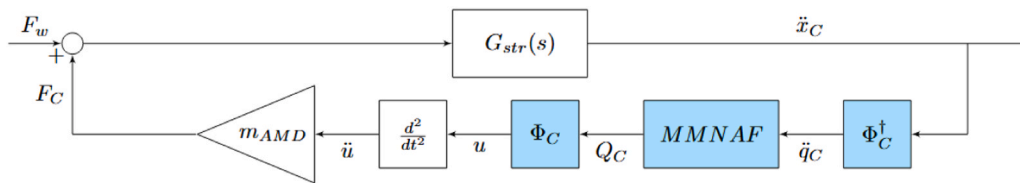


Fig. 15. Architecture of control algorithm.

It is possible to identify the additional damping introduced by the AMD as:

$$\xi_{add,NAF} = \frac{m_a g}{4m_i \xi_{c,i}} \quad (12)$$

where m_i represents the modal mass associated with the modal coordinate q_i .

Equating this expression with that of the additional damping provided by a damper with constant G :

$$\xi_{add} = \frac{G}{2m_i \omega_i} \quad (13)$$

the NAF parameters g_i and $\xi_{c,i}$ can be selected so that the damping effect on each mode matches the value associated with the design parameter G , used in the preliminary design phase (Section 5.1)

$$g_i = \frac{2G\xi_{c,i}}{M_{AMD}\omega_i} \quad (14)$$

The value of g_i is univocally defined only if $\xi_{c,i}$, a damping ratio associated to the controller transfer function, has been fixed. This parameter, proposed equal to 0.3 in the study of Yang, has been set equal to 0.5 to enhance the control system robustness.

Adopting a three-mode model of the tower ($g_i, i = 1, 2, 3$), the controller gains were computed starting from a single value of G , as defined in Eq. (14). This approach enables a multi-modal control law to be derived from the simple assumption of the AMD acting as an ideal viscous damper, analogous to the ideal Sky-Hook concept. The selected gain, $G = 50 \frac{kNs}{m}$, was shown through the numerical analyses in Section 5.1 to be consistent with the target performance requirements. The resulting gains were found to be stable, remaining lower than the stability threshold given by:

$$g_i < \frac{m_i}{m_{AMD}} \quad (15)$$

which is the ratio between the actuator mass and the modal mass of the selected mode.

6. AMD system installation and field testing

The AMD system designed and implemented as discussed in the previous sections was subsequently installed into the tower's roof structure and tested. This section provides an overview of the AMD installation and tower instrumentation phases, highlighting key results from field testing to evaluate the structure's dynamic response with and without the AMD system.

6.1. AMD system installation

The active vibration control system was installed during the month of August 2024. The system comprised the following elements:

- AMDs
- Central control unit
- Measurements nodes

The AMDs were lowered into the lens using a crane through four designated openings (closed and sealed after installation) and secured to dedicated steel beams (see Fig. 16). The actuator installation points match the locations considered in the numerical models during the simulation phase (see Fig. 17).

The Tower was heavily instrumented for both the operation of the AMD system and for monitoring purposes. The complete layout is reported in Fig. 17 and contextualized in the x-y reference system. The x-axis is parallel to AMD2 and AMD4, while the y-axis is parallel to AMD1 and AMD3. Regarding the array of sensors integrated into the tower, two categories can be distinguished based on their function:

- **Control nodes:** these are used to provide the real-time signals required by the system algorithm. Four control accelerometers (AC1–4) have been installed at the control nodes, each corresponding to the location of an actuator. These measurement points are depicted in orange in Fig. 17.
- **Monitoring nodes:** three monitoring accelerometers (AM1–3) were installed at selected monitoring nodes. These measure the oscillations of the structure to extract useful information about the structure's behaviour and assess comfort levels for the operators. The data from these sensors are transmitted to a dedicated server, which serves as input for the Structural Health Monitoring (SHM) analysis of the tower. Additionally, this acceleration data is accompanied by wind information, obtained from an ultrasonic anemometer placed

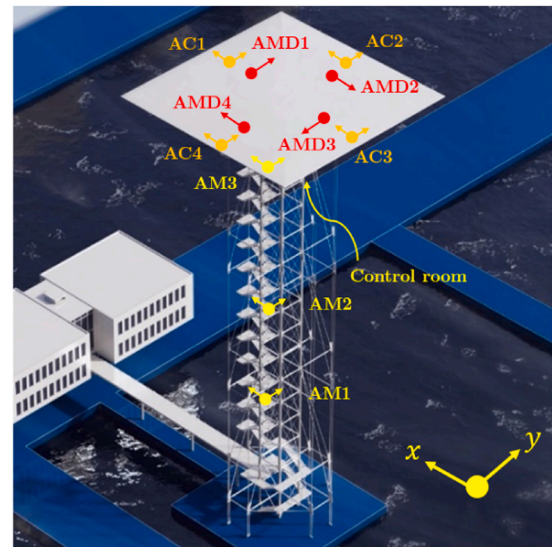


Fig. 17. Layout of AMDs and instrumentation.

on a pole springing from the top of the lens. The investigation of wind-structure interaction will be the focus of future studies. These nodes are shown in yellow in Fig. 17.

Both sensing and actuating devices were connected to a central control unit located in the tower's control room. The control unit contains all the electronics required for signal control and power transmission. It is equipped with a dedicated UPS and an electric generator to ensure the system remains operational in case the tower is disconnected from the grid.

6.2. Field testing

On September 9, 2024, dynamic field testing of the tower was conducted, beginning with a comprehensive series of frequency sweeps to determine the structure's natural frequencies. Once the experimental modal parameters were identified, multiple resonance excitation tests were performed, pushing the tower close to the limits established during the design/analysis phase. The excitation source for these tests was provided by the AMDs themselves, which were used in this context as harmonic exciter.

In the second phase, the AMDs operated in a hybrid mode, serving both as excitation sources and as active control mechanisms. The same testing protocol applied to the uncontrolled structure was repeated, enabling an assessment of the active damping system's effectiveness in reducing vibrations.

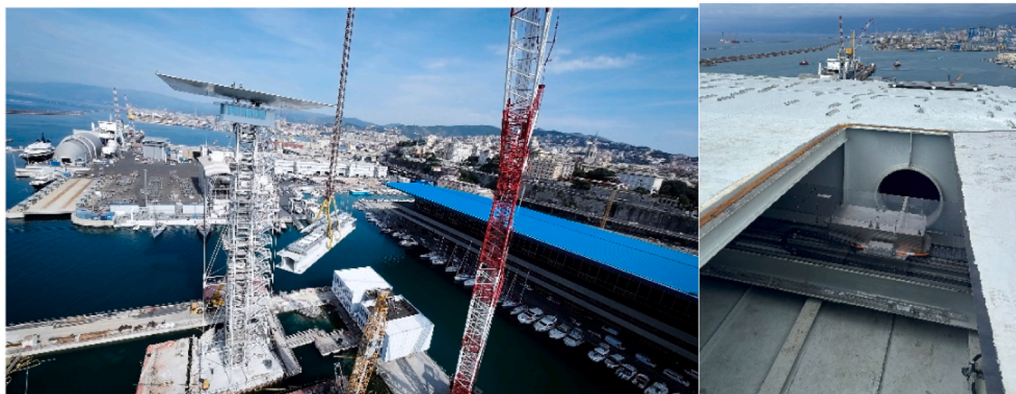


Fig. 16. Installation of AMDs inside the tower's lens.

A total of 20 tests were conducted, as summarized in Table 6. For brevity, this section discusses only a selected set of representative tests in detail; these tests are highlighted using bold characters in the table. Plots summarizing the results for the entire dataset are available upon request for interested readers.

6.2.1. Characterisation of the uncontrolled structure

The first tests (T1 and T2) were conducted to analyse the tower's dynamic behaviour using Experimental Modal Analysis (EMA). A frequency sweep from 0.2 Hz to 3 Hz was performed along both the X and Y axes of the structure, applying a constant inertial force of 750 N. AMD 1 and AMD 2 were used to excite the structure along the Y and X directions, respectively (see Fig. 17).

Table 6
Complete field-test matrix.

Test #	Test Description	Structure Condition	Output
T1	Dynamic identification with frequency sweep in Y direction (machine 1)	FRF Identification	FRF X & Y for Y excitation
T2	Dynamic identification with frequency sweep in X direction (machine 2)	FRF Identification	FRF X & Y for X excitation
T3	Dynamic identification with frequency sweep in Y direction (active system, machine 1)	FRF Identification	Damped vs. undamped comparison
T4	Dynamic identification with frequency sweep in X direction (active system, machine 2)	FRF Identification	Damped vs. undamped comparison
T5	Resonance excitation in Y-direction (1000 N)	Undamped structure	Excitation & displacement graph
T6	Resonance excitation in Y-direction (2000 N)	Undamped structure	Excitation & displacement graph
T7	Resonance excitation in Y-direction (4000 N)	Undamped structure	Excitation & displacement graph
T8	Resonance excitation in Y-direction (1000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T9	Resonance excitation in Y-direction (2000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T10	Resonance excitation in Y-direction (4000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T11	Resonance excitation in X-direction (1000 N)	Undamped structure	Excitation & displacement graph
T12	Resonance excitation in X-direction (2000 N)	Undamped structure	Excitation & displacement graph
T13	Resonance excitation in X-direction (4000 N)	Undamped structure	Excitation & displacement graph
T14	Resonance excitation in X-direction (1000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T15	Resonance excitation in X-direction (2000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T16	Resonance excitation in X-direction (4000 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T17	Resonance excitation in torsional mode (600 N)	Undamped structure	Excitation & displacement graph
T18	Resonance excitation in torsional mode (900 N)	Undamped structure	Excitation & displacement graph
T19	Resonance excitation in torsional mode (600 N, active system ON)	Actively damped structure	Damped vs. undamped comparison
T20	Resonance excitation in torsional mode (900 N, active system ON)	Actively damped structure	Damped vs. undamped comparison

During testing, acceleration measurements from all sensors were recorded alongside the excitation signals, enabling the estimation of a set of Frequency Response Functions (FRFs) in a MIMO (Multiple-Input Multiple-Output) configuration. The FRFs corresponding to the roof control nodes' responses in both directions are shown in Fig. 18.

By applying a nonlinear least-squares fitting algorithm to the experimental data, the modal parameters of the structure were extracted. As predicted by the FE model, the dynamic response is predominantly governed by the first three modes. A comparison of the experimental and numerical modal parameters is presented in Table 7.

The modal identification was performed in a MIMO (Multiple-Input Multiple-Output) framework, where all accelerometer signals were jointly fitted in the nonlinear least-squares algorithm. This approach produces a unique, consistent set of modal parameters, rather than independent estimates from each sensor. The damping ratios and frequencies reported in Tables 7 and 8 are therefore already aggregated across all sensors.

The experimental results show close agreement with the numerical predictions, with measured damping ratios slightly exceeding the conservative values assumed in the design. This difference is beneficial, as the higher damping further reduces structural accelerations and in no way undermines the validity of the findings or the demonstrated effectiveness of the proposed AMD solution.

Referring to the coordinate system illustrated in Fig. 17, the second and third modes correspond to bending motion along the X and Y axes, respectively, while the first mode is associated with torsional motion around the vertical axis.

Resonance tests were also performed on the uncontrolled tower by exciting the structure along the X and Y directions using AMD 1 and AMD 2, generating inertial forces of 1000 N, 2000 N, and 4000 N. Additionally, torsional excitation tests were conducted using AMD 1, with inertial forces of 600 N and 900 N.

Fig. 19 displays the structural response at different nodes for an excitation force of 4 kN in the X and Y directions, respectively (tests T7 and T13). A sinusoidal input was applied, tuned to match the resonance frequency of the natural mode associated with each axis. The figures show a 15 s segment from these tests, corresponding to steady-state conditions reached after the initial transient phase.

As observed, the structure exhibited significant displacements, with the response along the Y axis showing greater dynamic amplification. This behaviour is consistent with the lower damping ratio associated with the third mode. The responses of AC1–AC4, located at the same level in the floor area where the AMDs are installed, are strongly correlated as they capture essentially the same modal response. The maximum difference among them is within 5 %, which is not appreciable in the plotted results.

The torsional mode has also been excited using AMD1: Fig. 20 shows the accelerometers response in Y direction in response to a force of 900 N (Test 18).

6.2.2. Control system testing

To test the full potential of the active vibration control system, an algorithm capable of using an actuator to generate the exciting forces while also acting as part of the control system was implemented, exploiting the principle of superposition (illustrated in Fig. 21). In the implemented testing procedure, three actuators were programmed to act as dampers, with the fourth AMD receiving from the controller a reference u_i^{FB} necessary to exert a control force $F_{C,i}$ on the structure; such a control force is computed from the tower acceleration as explained in Section 5.2. Hence, AMD 1 was instructed to perform two tasks at once: apply a predetermined input force time history F_{in} , i.e. a sine sweep or a sinewave, while also responding in feedback with the force $F_{C,1}$ to the tower vibrations according to its intended use in normal operations. The superposition of the feedforward and the feedback actions comes from the sum of two reference positions, u_1^{FF} and u_1^{FB} respectively,

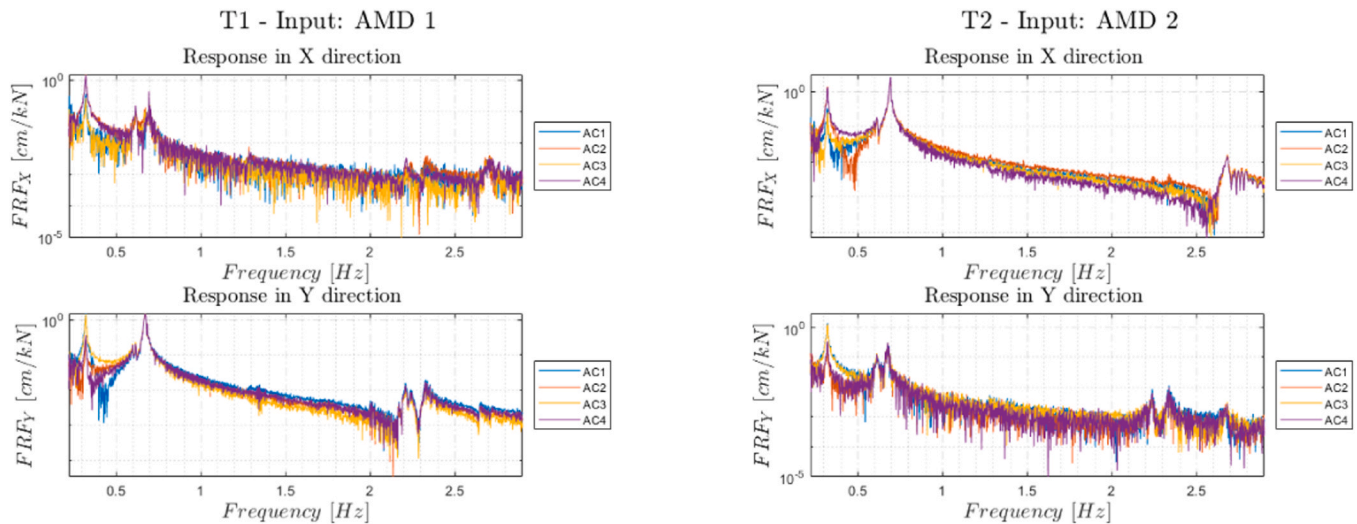


Fig. 18. Response of the tower obtained from tests T1 and T2.

Table 7

Measured frequencies and damping of the uncontrolled structure.

Mode	$f_{n,exp}$	$f_{n,num}$	$f_{n,exp}/f_{n,num}$	ξ_{exp}
1	0.32 Hz	0.265 Hz	1.21	0.61 %
2	0.67 Hz	0.617 Hz	1.09	0.65 %
3	0.69 Hz	0.634 Hz	1.09	0.37 %

commanded to the actuator.

Tests T3 and T4 were conducted using the same input forces used in tests T1 and T2, but this time the control system was activated. From the collected data, a new set of experimental FRFs was derived (illustrated in Fig. 22), and the estimation algorithm was applied again to extract the updated modal parameters, capturing the changes in the structure's dynamics due to the effect of the AMDs.

The control system was designed to increase the values of damping associated to the first three modes, targeting a value ξ^* established based on the numerical simulations discussed in Section 5, to ensure that the control room acceleration remains below the comfort threshold.

Table 8 shows the experimental damping ratios for both the

uncontrolled (ξ_{NC}) and controlled structure (ξ_C), also showcasing the target values ξ^* obtained via iterative numerical analysis in Section 5.1. It can be seen that the AMD system is capable of enhancing the damping coefficient on the desired modes by an order of magnitude, while also exceeding the target values. This is a deliberate outcome of the conservative design philosophy, in which the system is sized to deliver damping levels safely above the minimum target under conditions more demanding than those considered in the design case (here explicitly tested under resonance excitation) thereby ensuring robustness and inherently preventing under-design.

The increase in damping has a significant impact on the dynamic amplification of the controlled modes. This effect is evident in Fig. 23, which compares the experimental FRFs of the controlled and uncontrolled structure. The displayed results are extracted from the data of test T3, where the input force was applied along the Y axis, selecting as the FRF outputs the signals recorded from the measurement node in the control room.

To further investigate the structure behaviour in resonance conditions, a series of sinusoidal inputs were applied to the structure in both uncontrolled and controlled conditions (T5 – T20). These tests were designed to specifically target the natural frequencies identified from

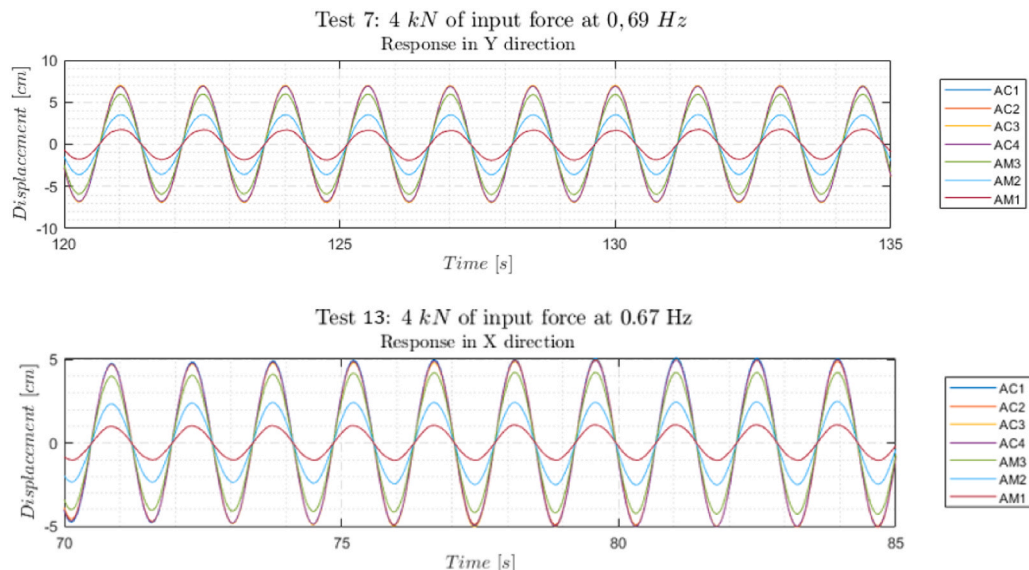


Fig. 19. Displacement-time response of the tower from Tests T7 and T13.

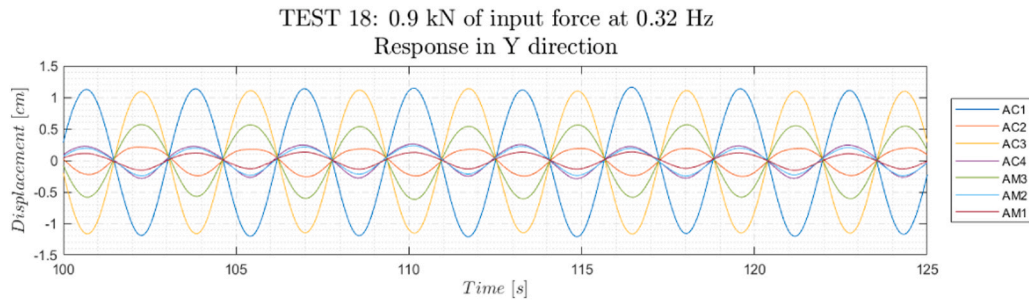


Fig. 20. Displacement-time response of the tower from test T18.

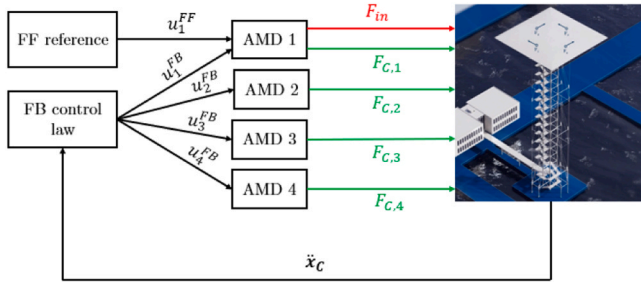


Fig. 21. Control system testing algorithm.

the EMA, introducing forces up to 4 kN. The input sinewaves were applied for 180 s, thus allowing the system to reach the steady-state condition necessary to observe the phenomenon of dynamic amplification in its fullness. Example results for these tests are reported in the time histories of Fig. 24, which shows the cabin vibration in the X direction in response to the excitation of the bending mode along the same axis, obtained exploiting actuator AMD 2. The vibration signals displayed for the uncontrolled and controlled case are extracted from test T13 and T16 respectively. An analogous comparison pertaining to the response along the Y axis is shown in Fig. 25, which presents an extract from the records of tests T7 and T10, performed using AMD 1 as excitation source.

Finally, Fig. 26 shows the performance of the control system in correspondence of the tower torsional mode (test T20).

7. Conclusions

This work presented a simulation-driven design process for implementing an Active Mass Damper system to ensure the operability and comfort of personnel within the slender steel control tower at the Port of Genoa, under dynamic wind loading. Through detailed finite element modelling and wind force simulations calibrated to normative standards, target damping requirements were identified to satisfy comfort criteria as defined by the CNR guidelines.

Based on these targets, time-domain analyses were performed to determine the necessary dynamic characteristics of the AMD units, including mass, stroke, and actuator capabilities. These specifications informed the mechanical design of an inertial actuator utilizing linear electric motors. During the modelling phase, AMDs were represented as viscous dampers, and to approximate their ideal behavior across the first three natural frequencies, a Multi-Modal Negative Acceleration Feedback (MMNAF) control algorithm was developed.

Following prototyping and production, the AMD system was installed inside the tower's roof structure. An extensive testing campaign was then carried out both with and without the active control system. Testing forces were applied using the AMDs themselves, leveraging their unique capability to combine excitation and control,

Table 8

Measured damping added by the AMD system.

Mode	ξ_{NC}	ξ_C	ξ^*
1	0.61 %	8.31 %	6.3 %
2	0.65 %	6.07 %	4.4 %
3	0.37 %	5.34 %	4.0 %

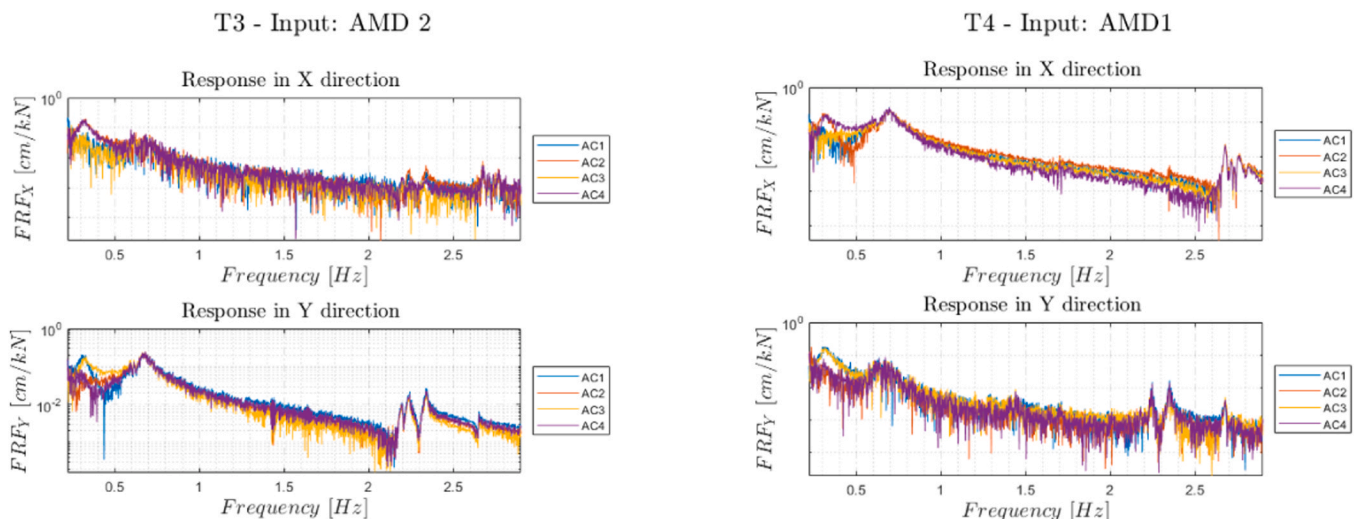


Fig. 22. Response of the controlled tower obtained from tests T3 and T4.

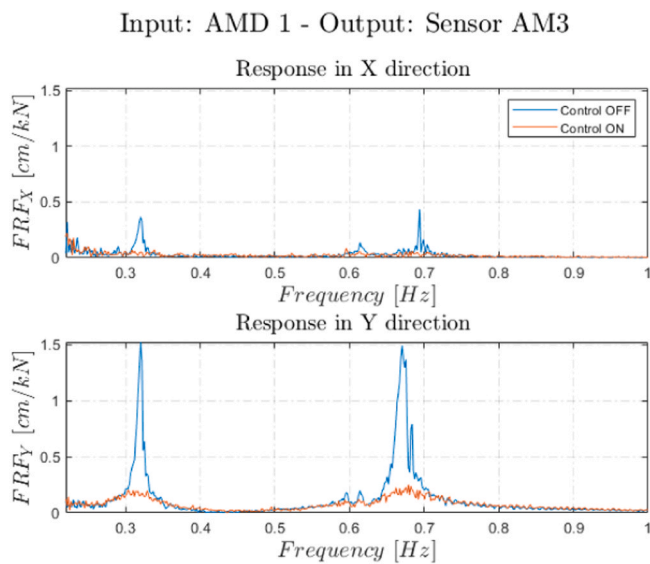


Fig. 23. Controlled vs uncontrolled frequency response from tests T3.

thereby enabling a robust validation of system performance. The experimental results confirmed that the structure's dynamic properties closely matched design-phase estimates and demonstrated that the active system successfully increased damping to the target levels, thereby improving occupant comfort.

Looking ahead, the tower's dense network of accelerometers continuously recording structural vibrations will offer a unique opportunity to collect high-resolution data on its response to real wind events. This will enable detailed, long-term evaluation of the control system's effectiveness under actual operating conditions, complementing the insights gained from controlled testing and enriching our understanding

of wind-structure interaction in service.

Beyond the technical achievements, this project exemplifies a successful integration of engineering ingenuity, innovation, and advanced control technology. The tower, a crucial component for port safety, is also poised to become an iconic architectural landmark within the cityscape. The process has yielded not only a functional and elegant solution to a complex engineering challenge but also valuable insights that pave the way for broader application of active vibration control technologies in future infrastructure. To this end, it should be emphasized that the design procedure presented in this paper—consisting of the following steps: (1) modal identification via FEM, (2) definition of loads in accordance with current regulations (e.g., CNR-DT 207/2008, wind tunnel tests), (3) specification of target performance criteria (e.g., maximum cabin acceleration for comfort), (4) numerical simulation with a simplified AMD model represented as a viscous damper, (5) sizing of the AMD under the most demanding project conditions, (6) selection of an appropriate control law based primarily on structural frequency and the number of relevant modes, and (7) experimental testing and validation—can be generalized and extended to the design of AMD systems intended to protect slender steel structures from the dynamic effects of wind loads.

CRedit authorship contribution statement

P. M. Calvi: Writing – review & editing, Writing – original draft, Supervision, Project administration, Formal analysis, Conceptualization. **L. Bogni:** Writing – original draft, Validation, Software, Formal analysis. **A. Bussini:** Software, Methodology, Investigation, Conceptualization. **S. Cii:** Validation, Methodology, Data curation, Conceptualization. **L. Bandini:** Writing – review & editing. **F. Ripamonti:** Writing – review & editing.

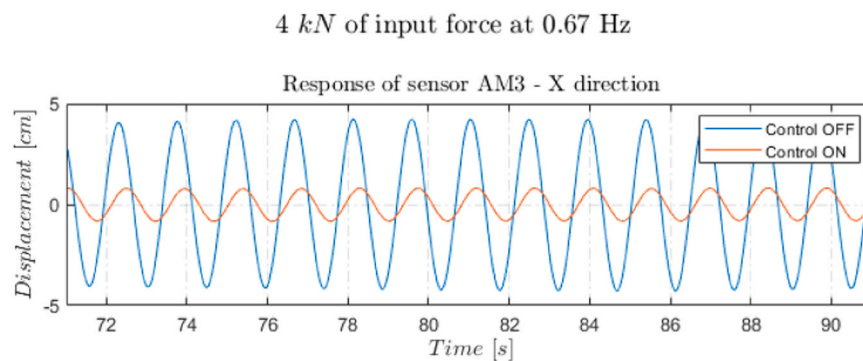


Fig. 24. Controlled vs uncontrolled displacement-time response of the tower along X direction.

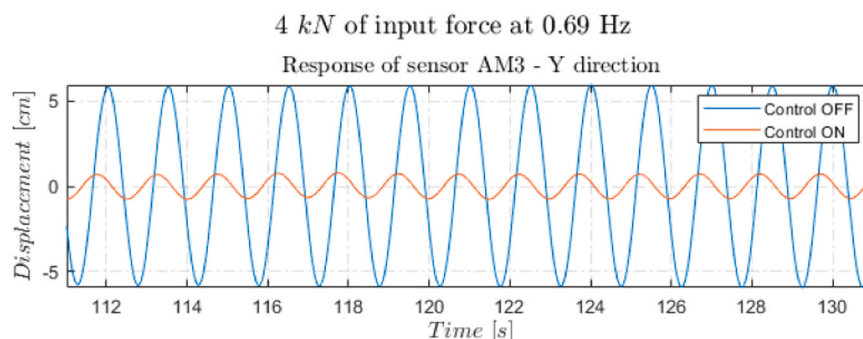


Fig. 25. Controlled vs uncontrolled displacement-time response of the tower along Y direction.

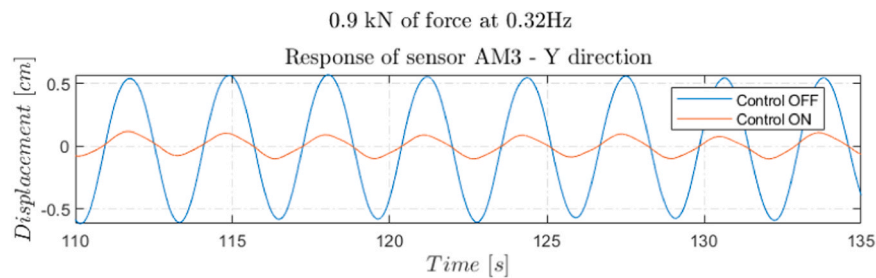


Fig. 26. Controlled vs uncontrolled displacement-time response of the tower, torsional mode.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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