

Energetically efficient, mediated mechanical system for precise control of hoisting operations

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We introduce a mechanical control system for energy efficient and robust hoisting crane operations. The control system efficiently translates the harmonic motion of a spring-loaded mediating system into the desired driving of the load, recycling most of the employed energy for subsequent operations. The control output is a shortcut-to-adiabaticity protocol borrowed from quantum mechanics. The control system reduces the single operation consumption in realistic working regimes, but it is in cyclical processes where the energetic advantage becomes substantial. The design of the control system and the control output is flexible enough to allow additional optimization of the robustness against perturbations.

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Introduction. Hoisting plants are essential to a variety of economic activities. They are intensively used in harbors, factories, logistic centers, and construction sites, among many others. They are typically powered by diesel or electric energy, which is mainly produced by burning fossil fuels [1]. Hoisting operations have increasingly been approached from an energy efficiency perspective [2], leading to control schemes devised to reduce energy consumption. A possible approach aims to recover the energy outflow during lowering operations using supercapacitors [3] or flywheels [4]. Alternatively, specific evolutions of crane parameters have been designed based on control theory [5]. For instance, optimal control has been used to reduce the energetic bill in various crane operation processes [6–8].

Recently, shortcuts to adiabaticity (STA) have been proposed as a robust alternative to optimal control of crane operations [9]. STA [10] are a set of mathematical tools to design the output of a given control operation by achieving the result of an infinitely slow evolution in shorter times. The origin of this set of techniques is in quantum mechanics, but a variety of models have already shown that they are easily adaptable to classical mechanics [11–16] and in particular to the control of crane operation [9]. In this context, an STA operation swiftly drives an initial equilibrium state to a final configuration without residual oscillations. Torrontegui *et al.* studied the STA transport of a load with an overhead crane and noticed that applying the driving force in a massive mediating system (MMS), whose trajectory determines the evolution of

the target system, but is not influenced by its back-action, leads to control operations that achieve the desired output regardless of the initial conditions of the system [17]. This is a desirable feature in any control operation, but as a counterpart, it was shown that the energy consumption of the operation was dominated by the power required to change the inertia of the MMS. Thus, the robustness provided by the MMS came with a price, increased energy consumption. This trend was later confirmed in different setups [18,19].

In this Letter, we present a control system that maintains the robustness provided by STA operations with MMS [17] while substantially reducing the energy consumption, becoming especially advantageous in cyclic operations. The proposed control system uses a spring-loaded MMS connected to the target system through an energetically passive guiding system (EPGS) that translates the oscillating motion of the MMS into the STA trajectory of the target system with no direct energy cost. Our approach removes the leading consumption term from the energetic bill of the operation, i.e., the inertial term of the MMS [17]. We test it for STA crane hoisting operations and analyze its robustness and energetic advantage compared to other control approaches.

The innovation of our proposal is twofold: (1) It introduces an EPGS that determines the output of the control protocol with no direct consumption and (2) it features a spring-loaded MMS that simultaneously plays a stabilizing role, detaching the operation of the actuating device from the specific conditions of the target, and the role of a mechanical battery, storing outgoing energy flows for subsequent operations. We argue that our approach establishes a control paradigm that provides robustness against disturbances and reduced energy consumption, which can be adapted, beyond specifics of each particular scenario, to any control operation, including the control of quantum systems.

Model of the target system. A hoisting operation consists of lifting a load that hangs freely from a pulley. The

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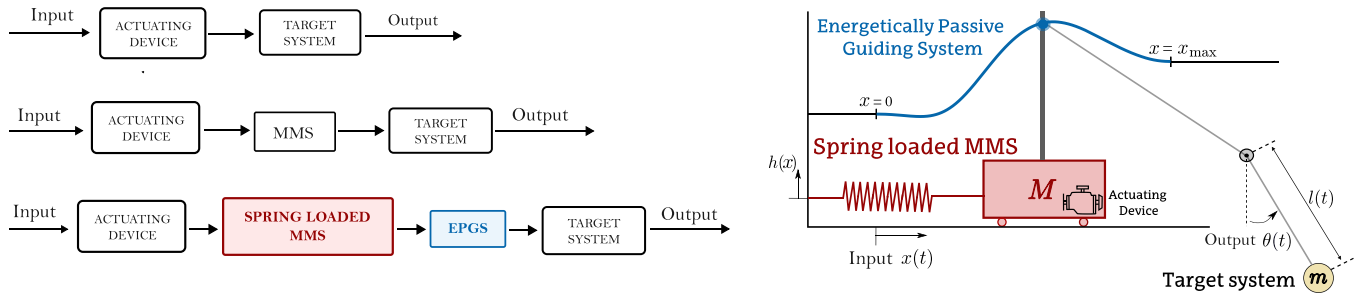


FIG. 1. Left: The three conceptual control approaches studied in this Letter, with the parts involved in each of them. Top: Direct control; middle: mediated control; and bottom: energetically efficient mediated control. Right: Schematic of the mechanical assembly that controls the hoist of our load m (target system). The control system (MMS) is simply a much heavier mass (M) than the one we intend to lift, which periodically oscillates following a harmonic oscillation. This MMS transfers the horizontal displacement to a cart moving in a guiding rail (EPGS) above the MMS. The cart connects with the pulley that lifts the mass m , such that its design, i.e., its height $h(x(t))$, effectively controls the lift as designed, $l(t)$.

time-dependent parameter that determines the trajectory of the load is $l(t)$, the length of the rope between the pulley and itself. The hoisting process is specified by the boundary values $l(0) = l_0$ and $l(t_f) = l_f$, where $l_0 > l_f$ and t_f is the duration of the operation. Additionally, we impose $\dot{l}(t_b) = \ddot{l}(t_b) = 0$ for smoothness (will give the protocol robustness against small deviations from the ideal design), where t_b stands for both boundary times, 0 and t_f . The Lagrangian that describes the target system is

$$\mathcal{L} = \frac{1}{2}m(\dot{l}^2 + l^2\dot{\theta}^2) + mgl \cos \theta, \quad (1)$$

where θ is the angle of the rope with the vertical and g is the gravity. Using now the Euler-Lagrange equation, $\frac{d}{dt}(\frac{\partial \mathcal{L}}{\partial \dot{\theta}}) - (\frac{\partial \mathcal{L}}{\partial \theta}) = 0$, we get $2l\ddot{\theta} + \dot{l}\dot{\theta} + g \sin \theta = 0$. Equivalently, we may use the horizontal displacement of the load as the variable to describe the motion of the target system, which, in the small oscillation regime (although in Appendix C we show that the method still works beyond this limit), leads to a simple harmonic oscillator $\ddot{q} + \Omega^2 q = 0$, where $q = l \sin(\theta)$ and we have defined $\Omega^2(t) = \frac{g}{l} - \frac{\dot{l}^2}{l}$. This motion is known to have an invariant [20] of the form $I = \frac{1}{2m}[\rho p - m\dot{\rho}q]^2 + \frac{1}{2}m\Omega_0^2(\frac{q}{\rho})^2$, where $p = m\frac{dq}{dt} = m\dot{q}$ is the momentum of the load and ρ is an auxiliary function that satisfies the Ermakov equation $\ddot{\rho} + \Omega^2\rho = \frac{\Omega_0^2}{\rho^3}$ with Ω_0 being an arbitrary value, which is typically defined as $\Omega_0^2 = \Omega^2(0) = g/l$ for convenience [note that this is satisfied by the boundary condition $\ddot{l}(0) = 0$]. Following the invariant-based inverse engineering theory [21] of the STA, this invariant guarantees reaching the unexcited target state so long as a set of boundary conditions is satisfied: $\rho(0) = 1$; $\rho(t_f) = \gamma = \sqrt{\Omega_0/\Omega_f} = \sqrt{l_f/l_0}$; $\dot{\rho}(t_b) = \ddot{\rho}(t_b) = 0$. See Appendix A for further details on invariant-based STA and Appendix B for the design of the STA trajectory for $l(t)$.

Control system. We introduce the energetically efficient mediated control (EEMC) system (see Fig. 1), in which the actuating device acts on a spring-loaded MMS, whose mass M largely exceeds the mass of the target m . The MMS initiates the operation with a large amount of potential energy (full extension of the spring) and periodically oscillates following a harmonic motion $x(t) = \frac{x_{\max}}{2}(1 + \cos(\omega t))$, where $x_{\max}$ is the amplitude of the oscillation and ω is the angular frequency.

A rigid bar directly transfers the horizontal displacement $x(t)$ to our EPGS, a cart moving along a preassembled rail whose height above the MMS determines the trajectory of the control parameter as

$$h(x) = \sqrt{[L - l(t[x])]^2 - [D - x(t)]^2}, \quad (2)$$

where L is the total length of the rope, D is the distance between the pulley and the cart at the maximum contraction point, and $l(t[x])$ is the design for the lift we numerically obtained in the previous section after changing to position parameters, which can be deduced from the harmonic oscillation equation as $t(x) = \frac{1}{\omega} \arccos(1 - \frac{2x}{x_{\max}})$. This equation can in principle lead to complex results, or at least to rough dynamics depending on the evolution, particularly if short hoisting times are chosen. We chose relaxed final times and smart $l(t)$ designs to avoid it (see Appendix B).

We consider two alternative control systems to drive the STA hoist (see Fig. 1) and compare them in terms of robustness and energy consumption. Direct control (DC) is the most straightforward, directly exerting the force over the pulley that controls the length of the rope. It has a simple implementation and is the most energy efficient of all, but is highly dependent on the initial configuration of the load, with a subsequent drop in the fidelity to reach the desired target state (see Ref. [17]).

In mediated control (MC), the actuating device acts on an MMS that is directly connected to the target with a rope that passes through a pulley. Thus, the horizontal motion of the MMS directly translates to the dynamics of the target system. Compared to DC, MC removes the dependence of the control operation on the initial configuration of the target and provides precise operations, but does so at a much higher energy consumption [17,18]. Computations for MC are performed with $h = 0$ for all times, so $l(t) = x(t)$.

As we show below, EEMC retains the robustness of MC (see Appendix C), with an energy consumption barely larger than for the DC in the ideal case. It requires a more cumbersome setup and a large initializing potential energy, but, as we will argue, it becomes very cost effective for cyclical processes. The spring-loaded MMS acts as an energy recycling reservoir that reduces dissipation, and, in the low-friction and many-cycle limit, EEMC yields controls that approach reversible operations with reduced entropy production per cycle.

Energy consumption. The energy required to perform the control operation is equal to the energy consumed by the actuating device to produce the force that drives it. We consider a simplified model, with friction only over the MMS, a reasonable approximation since $M \gg m$, and massless EPGs. However, our results remain valid in a more realistic scenario that includes a finite EPGs mass, as far as it is small compared to the mass of the MMS.

We first write the Lagrangian for the target and MMS in our EEMC method:

$$\begin{aligned}\mathcal{L}_{\text{TS}} &= \frac{1}{2}m(l'^2\dot{x}^2 + l^2\dot{\theta}^2 + mgl \cos \theta), \\ \mathcal{L}_{\text{MMS}} &= \frac{1}{2}M\dot{x}^2 - \frac{1}{2}M\omega^2\left(x - \frac{x_{\max}}{2}\right)^2 + F_k x,\end{aligned}\quad (3)$$

where we applied the chain rule such that $\dot{l} = l'\dot{x}$. In the second line, on top of the kinetic energy and the potential energy of the control mass, we added a term produced by some external force F_k we tagged as *kick force*, which will be responsible of keeping the control system following its ideal harmonic motion, necessary to compensate for friction and back action. To account for the friction, we will obtain the equations of motion by generalizing the Euler-Lagrange equation as $\frac{d}{dt}\left(\frac{\partial \mathcal{L}}{\partial \dot{q}}\right) - \left(\frac{\partial \mathcal{L}}{\partial q}\right) + \frac{\partial \mathcal{F}}{\partial \dot{q}} = 0$, where $\mathcal{F} = \mu\dot{x}^2/2$ is Rayleigh's dissipation function [22] and the Lagrangians are given in Eq. (3). Additional dissipation channels could be considered here with different Rayleigh terms. For example, friction in the EPGs would result in a (two-dimensional) dissipation term nearly proportional to $\dot{x}$, or dissipation in the pulley in an additional term in the back-action of the target proportional to $\dot{\theta}$. For simplicity, we stick to the single Rayleigh term model and consider different effective friction coefficients for comparison. The resulting set of equations of motion is

$$0 = 2l'\dot{x}\dot{\theta} + l\ddot{\theta} + g \sin \theta, \quad (4)$$

$$\begin{aligned}F_k &= ml'(l'\ddot{x} + l''\dot{x}^2 - l\dot{\theta}^2 - g \cos \theta) + M\ddot{x} \\ &\quad + M\omega^2\left(x - \frac{x_{\max}}{2}\right) + \mu\dot{x}.\end{aligned}\quad (5)$$

Once F_k is known, we can define the power exerted as $P = F_k\dot{x}$. Following discussions in Refs. [17,18], the energy consumption of the operations is

$$\mathcal{E} = \int_0^{t_f} P_+ dt + \eta \int_0^{t_f} P_- dt, \quad (6)$$

where P_+ (P_-) is the positive (negative) part of the power and $-1 \leq \eta \leq 1$ accounts for the possibility of storing and reusing the energy that flows out of the system during negative power intervals (see Appendix D). Here, we will stick to the case $\eta = 0$, although a small negative η would be more realistic, especially for strong braking forces.

Results. Figure 2 compares the kick forces of EEMC and MC in a benchmark operation. This example shows that the EEMC requires much smaller forces; therefore, it has a huge energetic advantage per cycle. Moreover, braking forces (positive forces in our case) appear only briefly at the end of the driving in the guided case, whereas the unguided benchmark case has notable braking forces. These do not affect our analysis since we set $\eta = 0$ [see Eq. (6)] but a braking force may

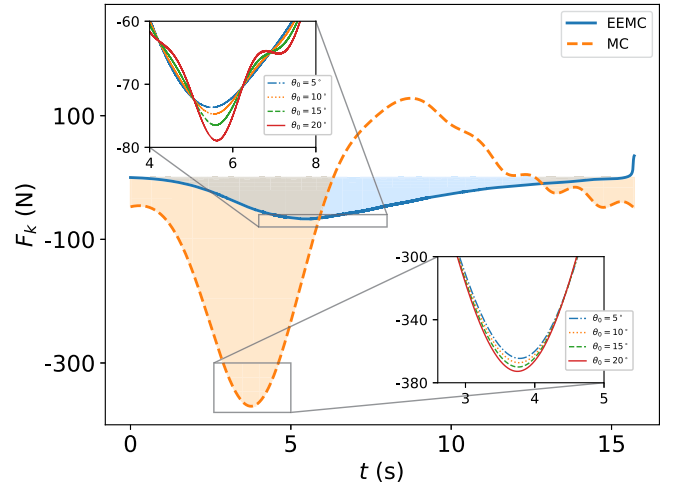


FIG. 2. Kick force F_k vs time for the EEMC (solid blue) and the MC (dashed orange). Shaded regions indicate the forces relevant for the consumption ($\eta = 0$). Insets show the different kick forces corresponding to varying initial conditions of the load. Parameters are $m = 5$ kg, $M = 1000$ kg, $\omega = 0.2$ s $^{-1}$, $L = 20$ m, $D = 12$ m, $l_0 = 5.5$ m, $l_f = 0.5$ m, $\mu = 0.02$, and $x_{\max} = 8$ m.

also have some sort of consumption, a further argument in favor of our approach. Notably, the inset in Fig. 2 shows that the kick force required to drive the operation with the EEMC barely changes for substantially different initial orientations of the load (comparable to the MC case), showing its robustness against changing initial conditions, retaining the benefits of the MC over the DC. Moreover, the design of the EPGS $h(x)$ also barely changes, and simulations show that these small deviations from case to case barely affect the excitation energies.

Figure 3 shows the energy consumed with the MC and the EEMC vs the friction coefficient. It reveals a reduced consumption of the EEMC for low-friction setups. As a benchmark, the minimum energy required for the lift would be the difference in potential energy, $mg\Delta l = 245$ J. A monotonic DC would consume exactly this amount of energy, while the ideal case EEMC only exceeds it by 1.2%. The ideal MC exceeds it by 96.8%. As friction increases, the advantage drops until the situation is even reverted. The rationale is that, with the more complex setup, the kick force in Eq. (5) has two leading terms ($\sim M$). In the ideal case, these two terms practically offset each other, as the term $\sim M\ddot{x}_{ho}$ gives exactly $-M\omega^2 x_{ho}$, but as we get away from the ideal driving, the kick force compensates to follow the perfect x_{ho} dynamics, at the cost of diverging from the ideal $\ddot{x}_{ho}$, so they do not cancel out so clearly. Anyhow, mechanical engineering companies report friction coefficients ranging from 0.001 to 0.005 with standard methods (even lower values are achievable through more advanced mechanisms such as cylindrical roller bearings) [23], values well within the regime where the EEMC yields reduced energy consumption. In the lower panel, Fig. 3 shows how the consumption scales as the mass of the MMS increases (recall that heavier MMS imply increased robustness of the operation against external perturbations). Here, the trend is clearly favorable for the EEMC.

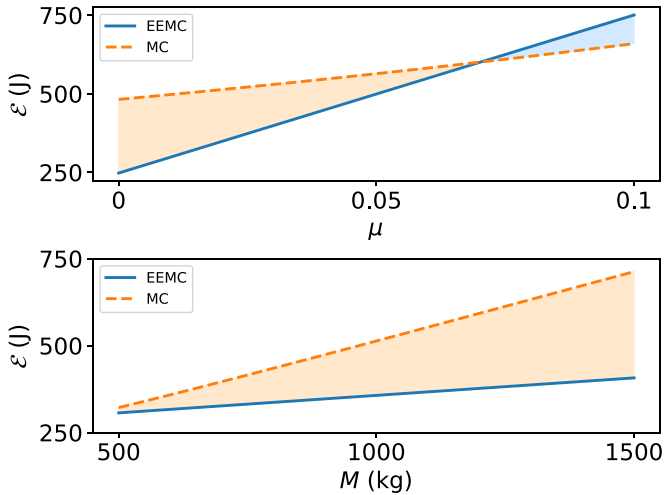


FIG. 3. Upper panel: Consumption energy for the EEMC (blue solid) and the MC (orange dashed) for different values of friction. Lower panel: Consumption energy for the EEMC (blue solid) and the MC (orange dashed) for different values of the MMS. Orange shaded regions indicate favorable regimes for the EEMC. Except for μ in the upper panel and M in the lower panel, the rest of the parameters are the same as in Fig. 2.

Figure 4 shows a comparison of a cycle-averaged operational cost $\bar{E}_{\text{op}} = \frac{E_{p,\text{min}} + NE_{\text{cyc}}}{N}$ for both EEMC and MC methods, for different friction coefficients. For the MC, as expected, we see a flat behavior, as there is no initialization cost. For the EEMC, we have a one-time cost in the form of the elastic potential, but the per-cycle operational cost is much lower, cycle-averaged cost quickly converges to lower values. Even considering the largest of frictions for EEMC, and no friction for the MC, we see the former to be eventually advantageous, albeit it takes more cycles to reach advantage. This is a solid proof that, even considering effectively higher

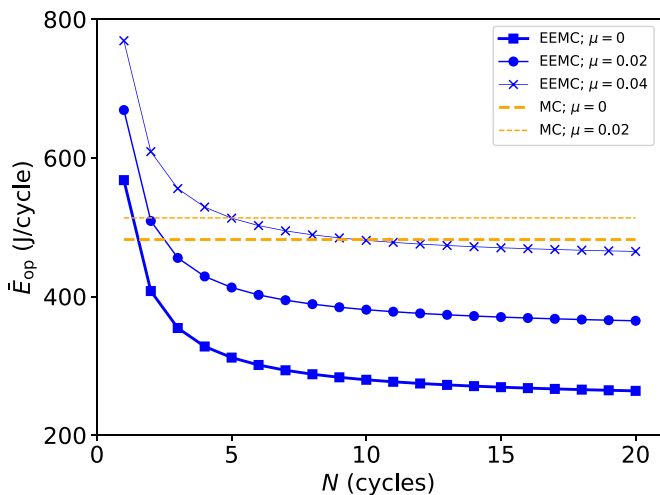


FIG. 4. Per-cycle operational cost $\bar{E}_{\text{op}} = \frac{E_{p,\text{min}} + NE_{\text{cyc}}}{N}$ for the EEMC (solid blue) and MC (dashed orange) vs the number of cycles. Decreasing line thicknesses indicate increasing values of the friction coefficients. For the EEMC, $E_{p,\text{min}} = \frac{1}{2}M\omega^2(x_{\text{max}}/2)^2 = 320$ J, and for the MC $E_{p,\text{min}} = 0$.

Rayleigh coefficients, EEMC still leads to reduced entropy production in the long term.

Conclusions. We have introduced the EEMC, a control system that provides robustness against varying initial conditions of the target state and reduced energy consumption. It features a spring-loaded MMS that stabilizes the control operation and recycles most of the energy of the operation for later use, and an EP GS that translates the oscillating motion of the MMS into the desired control output with no direct energy demand. We have presented a possible implementation of the EEMC for an STA crane hoisting operation, and shown that the combination of an STA trajectory and the EEMC yields an energetically efficient control operation that swiftly lifts a load without final excitations.

The EEMC system requires an initial input of potential energy; however, it leads to reduced energy consumption in a single operation, quickly reaching cost efficiency in cyclical operations. The consumption scales worse with friction compared with alternative control systems. However, it shows a clear energetic advantage for a realistic range of friction coefficients [24–26]. On the other side, the proposed method scales much better against M (mass of the MMS), so the EEMC system provides reduced consumption for increasing robustness.

The method of choice to design the control output, known as “invariant-based inverse engineering,” offers ample flexibility. On top of the conditions that guarantee a fast and excitationless driving of the target system, we could add additional conditions simultaneously, e.g., minimizing the consumption energy, minimizing the usage of material for the EP GS, or limiting its maximum slope, reducing the driving to the shortest possible time, or making the driving more robust with respect to external noise. Moreover, the EEMC system could be used to produce control outputs designed with other techniques such as optimal control theory.

Finally, we would like to highlight that the same road that led from quantum mechanics to classical mechanics can lead the way back. In fact, energy efficiency is also a major concern in quantum technologies [27]. The energy cost of implementing STA has been thoroughly analyzed in the literature [28–33] and it is widely accepted within the community of quantum thermodynamics [34] that STA are not a *free lunch*. Here, we put forward an example of low-cost STA implementation via passive mediation between a precharged work reservoir and the target system. In fact, our proposal virtually removes the need of active time-dependent driving, allowing the control operation to run almost autonomously, similar to what autonomous quantum machines seek [35]. The possibility of devising autonomous STA in other platforms is a promising avenue to reduce their energy cost, in particular in quantum thermal machines, where the energy required to generate external coherent control is orders of magnitude larger than the energy flows occurring within the quantum system [18].

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Data availability. All codes and data used in this Letter are available on GitHub and accessible through the Zenodo link [36].

Appendix A: Design of the STA trajectory. The dynamics described by the Harmonic Oscillator is known to have an invariant [20] of the form

$$I = \frac{1}{2m}[\rho p - m\dot{\rho}q]^2 + \frac{1}{2}m\Omega_0^2\left(\frac{q}{\rho}\right)^2, \quad (\text{A1})$$

where $p = m\frac{dq}{dt} = m\dot{q}$ is the momentum of the load and ρ is an auxiliary function that satisfies the Ermakov equation

$$\ddot{\rho} + \Omega^2\rho = \frac{\Omega_0^2}{\rho^3}, \quad (\text{A2})$$

with Ω_0 being an arbitrary value, which is typically defined as $\Omega_0^2 = \Omega^2(0) = g/l$ for convenience [note that this is satisfied by the boundary condition $\dot{l}(0) = 0$].

If the auxiliary function satisfies the initial conditions

$$\rho(0) = 1, \quad \dot{\rho}(0), \quad (\text{A3})$$

the invariant of motion in Eq. (A1) is equal to the mechanical energy of the load,

$$I(0) = \frac{p^2}{2m} + \frac{1}{2}m\Omega_0^2q^2 = E(0). \quad (\text{A4})$$

In addition, if $\rho(t)$ satisfies the final conditions,

$$\rho(t_f) = \gamma = \sqrt{\Omega_0/\Omega_f} = \sqrt[4]{l_f/l_0}, \quad \dot{\rho}(t_f) = 0, \quad (\text{A5})$$

the invariant of motion at final time takes the form

$$I(t_f) = \gamma^2 \frac{p^2}{2m} + \frac{1}{2}m\Omega_f^2q^2 = \gamma^2 E(t_f). \quad (\text{A6})$$

Since I is indeed an invariant of motion, we can state that $I(0) = I(t_f)$ and therefore

$$E(t_f) = \frac{E(0)}{\gamma^2} = \frac{\Omega_f}{\Omega_0} E(0). \quad (\text{A7})$$

That is, any trajectory of the control parameter $l(t)$ that satisfies the Ermakov equation (A2) with an auxiliary function $\rho(t)$ that fulfills the boundary conditions in Eqs. (A3) and (A5) leads to a final load state whose mechanical energy corresponds to the result of an infinitely slow transformation.

Appendix B: Choice of ansatz and satisfying boundary conditions. In the main text, we have attempted different approaches and ansätze to try and satisfy the numerous boundary conditions for $\rho(t)$, $l(t)$, and their derivatives. However, the results shown in the main text are always based on the same approach. We will detail here the different options and our reasons for eventually choosing this solution.

A nonstandard approach that has been attempted in similar problems with additional boundary conditions over variables that require solving a differential equation is a semidirect approach [37]. In this approach, we attempt an ansatz for $l(t)$ that satisfies the boundary conditions for $l(t)$ and its derivatives, and leaves some additional free parameters. Our choice was

$$\begin{aligned} l = & -l_0 + (-a_6 - 3a_7 + 10l_0 - 10l_f)t^3/t_f^3 \\ & + (3a_6 + 8a_7 - 15l_0 + 15l_f)t^4/t_f^4 \\ & - 3(a_6 + 2a_7 - 2l_0 + 2l_f)t^5/t_f^5 + a_6t^6/t_f^6 + a_7t^7/t_f^7, \end{aligned} \quad (\text{B1})$$

where a_6 and a_7 are the optimization free parameters and t_f is the final time, that is, half an oscillation period of the spring $t_f = \pi/\omega$.

The procedure now requires obtaining $\rho(t)$ by solving Ermakov's differential equation, and fixing the free parameters to satisfy the remaining boundary conditions for ρ and its derivatives. This can only be approximately achieved with a numerical method, in our case, a Nelder-Mead optimization algorithm. As the cost function, we borrowed the expression for the ground-state energy for the Ermakov equation from quantum mechanics:

$$F_{\text{cost}} = \frac{\hbar}{4\Omega_0} \left(\dot{\rho}^2 + \Omega^2\rho^2 + \frac{\Omega_0^2}{\rho^2} \right). \quad (\text{B2})$$

Although this approach has worked well in the past for similar problems that can only be seminumerically solved, we find it less accurate than the inverse problem in this case. Moreover, it leads to solutions that require very large and sharp changes in the design of the EPGS $h(x)$.

The inverse approach eventually chosen for the calculations in the main text starts with a similar ansatz with additional free parameters, in this case starting with $\rho(t)$. We attempted a polynomial and a sum of Fourier cosine ansätze, with up to three free parameters:

$$\begin{aligned} \rho_{\text{poly}} = & 1 + (-10 - a_6 - 3a_7 - 6a_8 + 10\gamma)t^3/t_f^3 \\ & + (15 + 3a_6 + 8a_7 + 15a_8 - 15\gamma)t^4/t_f^4 \\ & + (-6 - 3a_6 - 6a_7 - 10a_8 + 6\gamma)t^5/t_f^5 \\ & + a_6t^6/t_f^6 + a_7t^7/t_f^7 + a_8t^8/t_f^8, \end{aligned} \quad (\text{B3})$$

$$\rho_{\text{cos}} = (1 + \gamma)/2$$

$$\begin{aligned} & + \frac{1}{16}(9 + 32a_3 + 80a_4 + 144a_5 - 9\gamma)\cos(\pi t/t_f) \\ & + \frac{1}{16}(-1 - 48a_3 - 96a_4 - 160a_5 + \gamma)\cos(3\pi t/t_f) \\ & + a_3\cos(5\pi t/t_f) + a_4\cos(7\pi t/t_f) + a_5\cos(9\pi t/t_f). \end{aligned} \quad (\text{B4})$$

We now solve the same differential equation as before to obtain $l(t)$, and fix the free parameter so that they satisfy the boundary conditions for $l(t)$ and its derivatives. The cost function followed this time was

$$F_{\text{cost}} = mg|l(t_f) - l_f| + \frac{1}{2}m\dot{l}(t_f)^2. \quad (\text{B5})$$

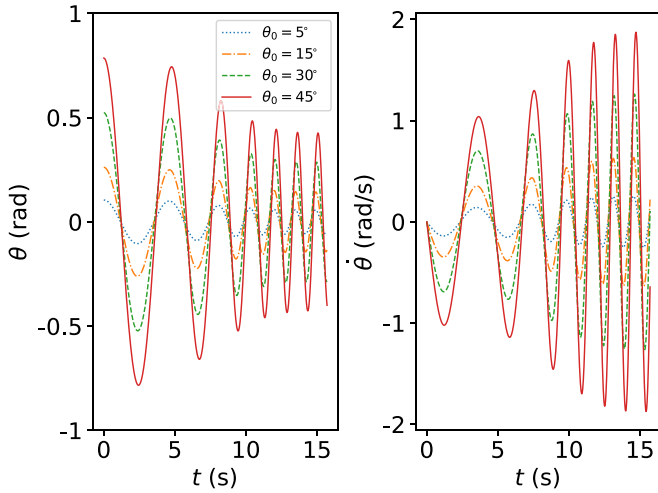


FIG. 5. Left: Time evolution of the oscillation of the hanging load $[\theta(t)]$ for four different initial angles: $\theta_0 = 5^\circ$ (blue dotted), $\theta_0 = 15^\circ$ (orange dash-dotted), $\theta_0 = 30^\circ$ (green dashed), and $\theta_0 = 45^\circ$ (red solid). Right: Time evolution of the derivative of the angle $\dot{\theta}(t)$ for the same cases.

By trial and error, we found this inverse approach to give better results, both due to smaller final excitations and because they lead to more reasonable and smoother designs of the rail. Particularly, the cosine ansatz (B4) leads to the smoothest designs, so it is the one we followed throughout all calculations in the main text.

Appendix C: Effectiveness of the STA beyond the small oscillation regime. The ideal lift of the hanging mass $l(t)$ is designed within the small oscillation approximation, which ensures that the Lagrangian (1) takes the form of a harmonic oscillator. However, this ideal design only guarantees the negligible excitation coming from the precision on the $l(t)$ and $\dot{l}(t)$ terms, which does not *a priori* set any control over the oscillation angle. Adding an MMS not only helps to flatten the effect of the initial angle θ_0 on the design of the EPGS but also keeps excitations from uncontrolled oscillations in check.

In Fig. 5, we solve Eq. (4) to show the time evolution of θ and $\dot{\theta}$ for four different initial conditions, ranging from 5° to 45° , well beyond the small oscillation regime. The simulations show how, although we get an expected oscillation velocity, it does not get out of control. Moreover, the oscillations do not get bigger in time; in fact, they are somewhat tamed by this mechanism.

Figure 6 shows the actual effectiveness of an MMS. For the same parameters used in the main text, we compare initial excitation energies caused by the initial angle θ_0 with the final excitation energies after the respective evolutions. For all studied initial angles, the final excitation is over an order of magnitude smaller than the initial one, showing that the method goes beyond just controlling the undesired oscillations, acting effectively as a shock absorber. Second, even when considering initial angles as large as 45° , far beyond the small oscillation regime, the method still works, showing similar qualitative and quantitative benefits.

Appendix D: Sign convention and value of η . In our model [see Fig. 1], coordinate x has its origin at the leftmost point of

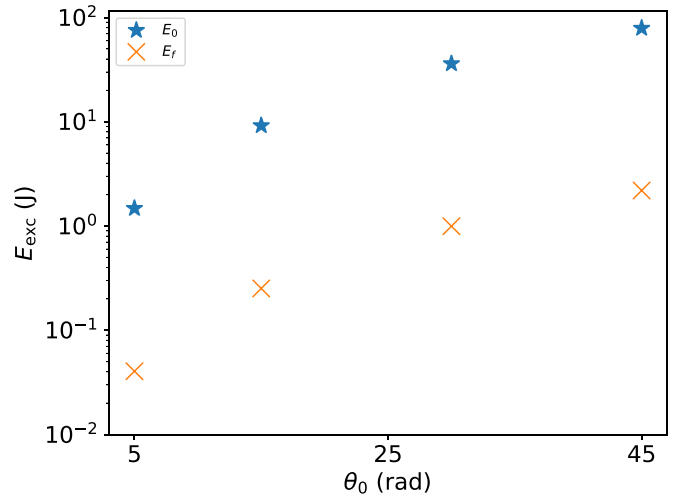


FIG. 6. Total excitation energy of the target system [as given by Lagrangian (1)] at initial and final times vs the initial angle θ_0 .

the oscillation of the spring-loaded MMS, positive values of x being those to the right of the origin as usual. $\dot{x}$ is therefore positive when the MMS is moving rightward and negative when the MMS is moving leftward. Likewise, the force F_k applied in the horizontal plane by the actuating device is positive when it is applied in the right sense and negative in the left sense. The instantaneous power exerted by the actuating device, computed as the scalar product F_k and $\dot{x}$, is (1) positive during accelerating phases, when the force acts in the same sense as the motion (both $\dot{x}$ and F_k are either positive or negative) and (2) negative during braking phases when the force acts against the motion ($\dot{x}$ and F_k have different signs).

Computing the energy required to drive a control operation as the integral of the instantaneous power exerted by the driving force implies a relevant and often overlooked assumption: The energy that flows out of the system during negative power intervals is somehow recovered by the actuating device. Such an assumption, however, does not correspond with the reality of the majority of industrial processes. Here, we account for the energy demand of a real actuating device by integrating separately time intervals where the driving force is providing energy to the system (positive power P_+) and time intervals where energy is flowing out of the system (negative power P_-) [see Eq. (6)] of the main text. Furthermore, the integrals of the negative power sections are multiplied by a factor $-1 \leq \eta \leq 1$ that accounts for the possibility of storing and reusing the energy that flows out of the system. In more detail:

(1) $\eta = 1$ corresponds to a mechanism able to fully accumulate the braking energy and give it back on demand, i.e., perfect regenerative braking.

(2) $\eta = 0$ corresponds to the limit in which braking fully dissipates the energy loss of the system with negligible energy consumption, i.e., an ideal braking system that allows for precise control of the motion with no active force.

(3) $\eta = -1$ corresponds to an active driving of the system during all phases of the motion, i.e., the actuating device requires energy to produce the driving force even during braking phases.

The abovementioned shows results with a reference value of $\eta = 0$. However, it has previously been argued that precise control operations such as the one proposed here require active driving at all times, hindering the possibility of energy recovery [17,18]. A scenario described by $\eta < 0$ is, in fact, more favorable for the EEMC. Note in Fig. 2 that MC requires stronger braking forces than EEMC, result-

ing, in the absence of an energy recovery mechanism, in increased consumptions for the MC. Therefore, we have presented results that demonstrate reduced energy consumption of the EEMC in a technically challenging scenario ($\eta = 0$), but we must emphasize that the advantage of our proposal is further increased in more realistic scenarios ($0 < \eta < -1$).

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