# Technical Notes on Input Shaping in Overhead Crane Systems

**Khalid Alghanim**

Mechanical Engineering Department, Kuwait University, P. O. Box: 5969, Safat, 13060, Kuwait; khalid.alghanim@ku.edu.kw

**Abstract:** Input shaping is an effective open-loop strategy for reducing residual payload oscillation in overhead crane systems, yet its practical performance depends on more than nominal oscillation cancellation. This study develops a coherent analytical treatment of input-function selection, robustness assessment, energy and power interpretation, terminal swing dynamics, input nonuniqueness, double-pendulum modeling, output shaping, and nonzero initial conditions. The main contribution is the integration of these design issues into a consistent set of physical and mathematical criteria for crane-command synthesis. The analysis clarifies the distinction between payload oscillation and structural vibration, explains the limitations of ideal impulse and discontinuous step commands, identifies numerical and analytical issues associated with high-degree polynomial profiles, and highlights exponential and low-order harmonic profiles as tractable smooth alternatives when actuator limits are enforced. Robustness is examined through sensitivity measures over finite parameter intervals rather than by local derivative cancellation alone. Energy arguments show that, for fixed boundary states in an ideal lossless system, total work is determined by the prescribed initial and final states, whereas peak power, root-mean-square power, and actuator effort remain meaningful design objectives. The study further shows that input scaling provides a simple means of adjusting terminal trolley speed after oscillation-suppression conditions are satisfied, and that nearly singular double-pendulum descriptions should be reduced when one pendulum mass is negligible. These findings support implementable input-shaping designs with clearer terminology, stronger physical consistency, and improved relevance to crane operation.

**Keywords:** input shaping, overhead cranes, oscillation suppression, robustness, energy and power, output shaping, nonzero initial conditions

**Citation:** Alghanim, K. 2026. Technical notes on input shaping in overhead crane systems. *TK Techforum Journal (ThyssenKrupp Techforum)* 2026(2): 96–104. https://doi.org/10.71448/tk202625

Received: 04-February-2026
Revised: 05-May-2026
Accepted: 16-May-2026
Published: 13-June-2026



## 1. Introduction

Overhead cranes transport suspended payloads whose dominant lateral motion is governed by pendulum dynamics. During acceleration and deceleration, trolley motion can excite payload swing, reduce positioning accuracy, increase cycle time, and raise safety concerns. Input shaping addresses this problem by modifying the commanded trolley motion so that oscillatory components induced during the maneuver cancel at the final time. Classical Zero-Vibration (ZV), Zero-Vibration-and-Derivative (ZVD), and Extra-Insensitive (EI) shapers have therefore become central tools for residual oscillation suppression in crane control [1–3]. Multi-mode, constrained, adaptive, and hybrid shapers further broaden the applicability of the approach to systems with varying cable length, actuator limitations, and multiple dynamic modes [4,5].

Despite this maturity, several design issues remain insufficiently connected. Many formulations emphasize residual oscillation cancellation at a nominal frequency, while the physical realizability of the command, the conditioning of the resulting algebraic equations, the distinction between energy and power, and the consequences of nonzero initial states are treated separately. These issues are especially important in overhead cranes because the command must be implemented by actuators with finite bandwidth, finite force, and operational limits, and because the suspended payload may not be at rest when the maneuver begins.

The scientific motivation of this study is to establish a consistent analytical basis for evaluating input-shaping commands used in crane systems. The originality lies in linking command smoothness, robustness, power demand, terminal swing phase, input scaling, and model reduction within the same physical interpretation. This connection provides practical design guidance: a shaped command should not only eliminate residual oscillation for a nominal model, but should also remain well conditioned, actuator-compatible, and meaningful under parameter variation and nonzero initial conditions.

The discussion clarifies terminology, compares candidate input functions, examines robustness measures, interprets energy and power, and identifies modeling conditions that affect single- and double-pendulum crane descriptions. These elements support rigorous synthesis and assessment of overhead-crane input shapers.

## 2. Vibration versus oscillation

The suspended payload motion in an overhead crane is more accurately described as oscillation than as vibration. In structural dynamics, vibration commonly refers to elastic deformation or high-frequency small-amplitude motion of a mechanical component. By contrast, the payload motion in a crane is predominantly a low-frequency pendulum swing generated by trolley acceleration and deceleration. The distinction is not merely semantic: it affects modeling assumptions, the interpretation of performance indices, and the selection of control objectives.

The established shaper names ZV, ZVD, and EI are retained because they are standard in the literature. However, when describing the payload response, the term "oscillation" better represents the relevant physical phenomenon. This terminology separates pendulum swing suppression from structural vibration control and helps avoid ambiguity when both flexible crane structures and suspended payload dynamics are present.

## 3. Input functions

The trolley input function is a central design variable because it determines both the excitation applied to the payload and the feasibility of actuator tracking. Throughout this study, $u(t)$ denotes trolley displacement and $f(t) = \ddot{u}(t)$ denotes the commanded trolley acceleration. The choice of $f(t)$ determines the residual oscillation, the terminal trolley speed, the peak force demand, the jerk content, and the numerical conditioning of the synthesis equations.

Traditional input shaping is often expressed as a sequence of ideal impulses [1,3]. These impulses are useful mathematical objects for deriving cancellation conditions, but they should not be interpreted as directly realizable actuator forces. A mechanical drive cannot generate acceleration changes of zero duration or infinite bandwidth. In implementation, ideal impulses are approximated by finite-duration commands, and the approximation error can be significant for sensitive shapers such as ZV. Actuator delay, saturation, rate limits, and motor-current dynamics can therefore convert a nominally valid impulse shaper into an inaccurate command.

Piecewise-constant acceleration commands provide a more practical alternative because they can approximate impulse sequences while preserving explicit control over the maneuver duration and final speed. Their main limitation is the discontinuity at switching instants. These discontinuities introduce jerk, increase tracking error in low-bandwidth drives, and may excite unmodeled flexible modes. Step-based commands are therefore attractive for simple equipment, but their switching times and amplitudes should be selected with actuator limits and payload-swing constraints in mind.

Harmonic input functions offer smoother profiles and can reduce high-frequency content in the command [6]. Low-order harmonic descriptions with a small number of low-frequency terms are particularly useful because they remain analytically tractable and can be tuned to avoid the natural frequency of the suspended load. By contrast, a large number of harmonic terms or terms with high frequencies can produce undesired beating, near-resonant excitation, and rapidly varying force demands. A harmonic shaper is therefore most effective when its frequency content is deliberately separated from the payload and structural modes.

Polynomial profiles are also valuable because they can enforce boundary conditions on displacement, velocity, acceleration, and sometimes jerk. Their main difficulty is numerical conditioning. High-degree polynomials can produce ill-conditioned coefficient matrices, especially when the polynomial is written directly in the physical time variable. This difficulty is reduced by using the normalized time $\tau = t/t_f$, where $t_f$ is the maneuver duration, but high-degree profiles can still amplify coefficient errors and generate large intermediate command values. Closed-form response derivations also become cumbersome because terms such as $\int t^n \cos(\omega t)\, dt$ increase rapidly in algebraic complexity as the degree $n$ increases.

Exponential and exponentially modulated trigonometric functions provide another smooth class of inputs. Terms such as $e^{-nt}\cos(\omega t)$ have compact Laplace-domain representations and are generally easier to integrate with pendulum response functions than high-degree polynomials. They can also represent commands that decay smoothly within the maneuver. Their use should nevertheless be constrained by boundary conditions, acceleration limits, jerk limits, and final-speed requirements. Under these constraints, exponential terms provide a tractable option for designing smooth crane inputs with manageable analytical complexity.

## 4. Robustness conception

Robust input shaping addresses uncertainty in system parameters, especially variations in cable length and the associated natural frequency. A ZVD shaper improves robustness by forcing the residual-oscillation sensitivity curve to have zero slope at the nominal design point [2,3]. This local flattening reduces sensitivity to small parameter errors, but it does not by itself guarantee small residual oscillation over a finite operating interval.

Let $\lambda$ denote an uncertain parameter such as cable length or natural frequency, and let $S(\lambda)$ denote the resulting residual-oscillation measure after a maneuver. Local derivative cancellation imposes conditions such as $S(\lambda_0) = 0$ and $S'(\lambda_0) = 0$ at the nominal value $\lambda_0$. A broader robustness objective is obtained by evaluating sensitivity over the intended operating interval:

$$J_S = \int_{\lambda_{\min}}^{\lambda_{\max}} S(\lambda) w(\lambda)\, d\lambda, \qquad S_\infty = \max_{\lambda \in [\lambda_{\min}, \lambda_{\max}]} S(\lambda), \tag{1}$$

where $w(\lambda)$ is a weighting function. The integral objective $J_S$ reflects the accumulated residual oscillation over the interval, whereas $S_\infty$ captures the worst admissible response.

This distinction is important because a curve may have zero slope at the nominal point while increasing rapidly away from it. For example, $ax^2$ has zero slope at $x = 0$, but a large coefficient $a$ produces a steep increase outside a small neighborhood. Similarly, a ZVD shaper may be locally robust while still producing unacceptable residual oscillation when the cable length moves outside the immediate vicinity of the nominal value.

Methods that directly minimize sensitivity measures over a finite interval, including Minimum Vibration and Integral (MVI) and Zero Vibration and Minimum Integral (ZVI) shapers, more directly address this limitation [3]. A ZVI shaper enforces zero residual oscillation at a selected nominal condition while minimizing residual oscillation over neighboring conditions. An MVI shaper relaxes the exact nominal cancellation condition and instead reduces residual oscillation over a prescribed continuous or discrete set of parameter values. These designs should be compared using the same maneuver duration, actuator limits, and residual-oscillation measure so that robustness is not confounded with command length or input magnitude.

**5. Initial conditions, energy, and power**

Initial conditions determine the mechanical state from which input shaping begins. A common idealization assumes that the trolley starts from rest and that the payload begins with zero angular displacement and zero angular velocity. The maneuver then accelerates the trolley to a desired cruising speed $v_c$ while returning the payload to zero angular displacement and zero angular velocity [2]. In practical operation, however, the payload may already be swinging, and the initial state must be included explicitly in the shaper design.

For an ideal frictionless crane with prescribed initial and final states, the net work input is fixed by the change in total mechanical energy. If the system starts from rest with no payload oscillation and ends with the trolley and payload translating together at speed $v_c$ with no residual oscillation, the required change in translational kinetic energy is

$$\Delta E = \frac{1}{2} m_{\text{tot}} v_c^2, \tag{2}$$

where $m_{\text{tot}}$ is the total translating mass. Under these boundary conditions, minimizing net energy input is not a meaningful independent objective for the ideal lossless model because the initial and final energies are already prescribed.

Power and actuator effort remain meaningful design quantities. The instantaneous mechanical power is

$$P(t) = F(t)\dot{u}(t), \tag{3}$$

where $F(t)$ is the horizontal actuator force. Distinct input functions that produce the same net work can have very different peak power, root-mean-square power, regenerative power, jerk, and force demands. Commands that move the trolley back and forth during a maneuver may leave the same final energy but increase positive and negative power exchange. Conversely, extending the maneuver can reduce peak force and power, although excessive duration may be unacceptable for productivity. A rigorous input-shaping design should therefore distinguish fixed net energy from optimizable power and actuator-effort measures.

**6. Force and acceleration**

In the absence of losses, disturbances, and payload oscillation, a crane moving at constant trolley speed requires no continuous horizontal force. This follows directly from Newton's law for the translating rigid-body component: if the acceleration is zero and no external horizontal resistance is present, the required rigid-body driving force is zero. This statement applies only to the ideal cruising condition in which the payload is not swinging relative to the trolley.

When the payload continues to oscillate during constant-speed trolley motion, the actuator may still need to supply a time-varying force to maintain the prescribed trolley velocity. The horizontal component of cable tension alternates with the payload phase. The net work over complete oscillation cycles may be small in an ideal model, but the instantaneous force and power can be significant and should be included in actuator sizing.

Under the sign convention used in $L\ddot{\theta} + g\theta = \ddot{u}$, and for a massless trolley idealization, the horizontal driving force associated with the swinging payload can be written as

$$F = m\Big(\ddot{u} + L\dot{\theta}^2 \sin\theta - L\ddot{\theta}\cos\theta\Big). \tag{4}$$

For small swing angles, this expression becomes

$$F \approx m\big(\ddot{u} - L\ddot{\theta}\big). \tag{5}$$

Using the linearized equation of motion, the force reduces to $F \approx mg\theta$. This result provides a direct interpretation: the force required to regulate trolley motion is proportional, to first order, to payload angular displacement. The proportionality also shows why force and power can remain nonzero even when the trolley acceleration itself is zero.

**7. Simplified input and single input-step**

Input simplicity is important for small facilities and low-cost crane systems because complex profiles may require high-bandwidth drives, accurate sensing, and advanced motion controllers. A simple acceleration command may be slower than a highly optimized profile, but it can be easier to implement and more reliable when actuator capabilities are limited [7].

For the linear single-pendulum model with zero initial conditions and constant trolley acceleration $f_0$, the equation

$$L\ddot{\theta} + g\theta = f_0, \qquad \omega = \sqrt{g/L}, \tag{6}$$

produces

$$\theta(t) = \frac{f_0}{g}(1 - \cos\omega t), \qquad \dot{\theta}(t) = \frac{f_0\omega}{g}\sin\omega t. \tag{7}$$

Residual oscillation is eliminated when $\cos\omega t_f = 1$ and $\sin\omega t_f = 0$, which gives

$$t_f = \frac{2\pi n}{\omega}, \qquad n = 1, 2, 3, \ldots. \tag{8}$$

The final trolley speed is $v_c = f_0 t_f$, so $f_0 = v_c/t_f$. Larger values of $n$ reduce acceleration and peak swing but increase maneuver duration. This single-step solution therefore reveals a useful trade-off among simplicity, cycle time, acceleration amplitude, and maximum payload angle.

A two-step acceleration command provides additional design freedom while remaining structurally simple. With two acceleration levels and a switching time, the command can satisfy final angular displacement, final angular velocity, and final trolley speed over a wider set of maneuver durations. It can also reduce peak swing, jerk, or power relative to the single-step solution. Because the command contains few parameters, it is well suited for use with multi-pendulum models and for inclusion of measured actuator delay or switching latency.

**8. Final-swing dynamics**

The terminal phase of a maneuver must be consistent with both the payload phase and the trolley-velocity constraint. For the linearized model

$$L\ddot{\theta} + g\theta = \ddot{u}, \tag{9}$$

when the payload is close to the equilibrium position, $\theta \approx 0$, the relation reduces locally to $\ddot{u} \approx L\ddot{\theta}$. Thus, the sign of the terminal trolley acceleration is tied to the sign of the payload angular acceleration under the chosen sign convention.

Consider a positive final trolley velocity and a command that approaches this velocity monotonically from below. Near the end of the maneuver, such a command requires nonnegative acceleration before settling to zero. The corresponding terminal payload phase must therefore satisfy $\ddot{\theta} \geq 0$ while $\theta$ and $\dot{\theta}$ approach zero. A terminal approach that would require negative acceleration is incompatible with a monotone approach to the final speed; it would imply that the trolley must first exceed the desired cruising speed and then decelerate.

This observation does not prohibit nonmonotone commands. It states that the terminal swing phase, the sign convention, and the velocity constraint must be mutually consistent. If the command is constrained to be monotone in velocity, the payload should approach equilibrium from the dynamically compatible side. If nonmonotone velocity is allowed, the design must account for the additional force and power consequences of the terminal deceleration.

**9. Maneuvering time**

Maneuvering time strongly affects input amplitude, residual oscillation, and numerical conditioning. Very short maneuvers generally require large acceleration and force. Increasing the maneuver duration can reduce these demands, but for a fixed input parameterization the resulting algebraic system may become ill conditioned at particular values of $t_f$.

Most analytical shapers determine a vector of unknown input parameters $p$ from a system of equations

$$A(t_f)p = b, \tag{10}$$

where the rows of $A(t_f)$ represent terminal conditions such as zero angular displacement, zero angular velocity, and specified final trolley speed. Large input amplitudes occur when $A(t_f)$ is nearly singular. In that case, small changes in the terminal conditions or in the model parameters can produce large changes in the command parameters. The determinant of $A(t_f)$ can indicate singularity for square systems, but the condition number is a more reliable measure of numerical sensitivity.

Ill conditioning also has a physical interpretation. When two terminal conditions become nearly dependent at a particular maneuver duration, satisfying one condition nearly enforces the other. The input then contains more independent parameters than are effectively needed at that duration, and the solution may use large cancelling amplitudes. Avoiding such maneuver times, normalizing time variables, and comparing commands under common amplitude and duration constraints improve the reliability of the synthesized shaper.

**10. Input uniqueness**

Shaped inputs are commonly generated by selecting a functional form and determining its unknown coefficients from the crane response. For the linear single-pendulum model with $f(t) = \ddot{u}(t)$ and zero initial conditions, Duhamel's integral gives

$$\theta(t) = \frac{1}{\omega L}\int_0^t f(\beta)\sin(\omega(t-\beta))\,d\beta. \tag{11}$$

The residual-oscillation conditions at $t_f$ are homogeneous in $f(t)$. Therefore, if a nonzero command $f(t)$ satisfies the residual-oscillation cancellation conditions, then $\alpha f(t)$ satisfies the same cancellation conditions for any constant scaling factor $\alpha$.

The final trolley speed, however, is not invariant under this scaling because

$$v_c = \int_0^{t_f} f(t)\,dt. \tag{12}$$

Scaling the command changes $v_c$ by the same factor $\alpha$. Thus, the oscillation-cancellation part of the solution is nonunique, while adding a specified final speed fixes the scale. This property is useful in design: one may first find a normalized command that cancels residual oscillation, and then scale it to obtain the required cruising speed.

For multi-step commands, the same idea can reduce the number of unknowns. If the first step amplitude is nonzero, it may be normalized to unity while the remaining relative amplitudes and switching times are solved from the oscillation constraints. A final scaling factor then enforces the desired trolley speed. This normalization reduces algebraic complexity and improves numerical conditioning without changing the physical cancellation mechanism.

## 11. Robust shapers

Robust shapers reduce residual oscillation sensitivity to uncertain parameters such as natural frequency or cable length [2,3]. Higher-order robustness is often obtained by adding derivative constraints, input segments, or optimization conditions. These additions can increase maneuver duration and input complexity, so their benefit should be assessed against actuator limits and operational requirements.

A common residual measure for the linear oscillator is the normalized state amplitude

$$R = \sqrt{\theta^2 + \left(\frac{\dot{\theta}}{\omega}\right)^2}. \tag{13}$$

For small angles, energy conservation gives

$$\frac{\dot{\theta}^2}{\omega^2} = 2(1 - \cos\theta_{\max}) \approx \theta_{\max}^2, \tag{14}$$

when the payload passes through $\theta = 0$ after reaching the maximum angle $\theta_{\max}$. Thus, the angular-position and angular-velocity terms contribute at the same order to the residual state amplitude. A robustness condition based only on angular displacement may be simpler, but it should be accepted only when the resulting angular-velocity residual is also bounded by analysis or measurement.

Robust shapers often appear less sensitive partly because they use longer maneuver durations than non-robust shapers. Longer durations may reduce excitation even without derivative robustness. A fair comparison should therefore isolate the effect of the shaper structure by using common maneuver durations, common acceleration limits, or common power limits. Under such controlled comparisons, any reduction in residual oscillation can be attributed more clearly to the robustness design rather than to a slower command.

## 12. Double-pendulum model

Double-pendulum models are useful when the crane hook, hoist cable, and payload introduce two significant swing modes [5]. They are also more sensitive to modeling assumptions than single-pendulum models because the inertia matrix can become poorly conditioned when one mass is much smaller than the other.

For the single-pendulum model, if the final angular displacement, final angular velocity, and final trolley acceleration are all zero, the equation $L\ddot{\theta} + g\theta = \ddot{u}$ reduces to $\ddot{\theta} = 0$ at the final time. This confirms consistency between the final boundary conditions and the absence of residual excitation.

For the double-pendulum system, the corresponding reduced final condition can be expressed as

$$M\begin{bmatrix}\ddot{\theta}_1 \\ \ddot{\theta}_2\end{bmatrix} = 0, \qquad M = \begin{bmatrix}(m_1 + m_2)L_1^2 & m_2 L_1 L_2 \\ m_2 L_1 L_2 & m_2 L_2^2\end{bmatrix}. \tag{15}$$

The determinant is

$$\det(M) = m_1 m_2 L_1^2 L_2^2. \tag{16}$$

When $m_1 > 0$, $m_2 > 0$, $L_1 > 0$, and $L_2 > 0$, the matrix is nonsingular and the only exact solution is $\ddot{\theta}_1 = \ddot{\theta}_2 = 0$. As $m_1$ approaches zero, however, the determinant approaches zero and the model becomes ill conditioned. In the limiting case $m_1 = 0$, the two-coordinate description contains a redundant degree of freedom and should be reduced rather than treated as a full double-pendulum model.

This reduction has practical importance. When the upper mass is negligible relative to the payload, the physical behavior approaches that of a single pendulum. Using the reduced model avoids unnecessary terminal conditions, unnecessary input segments, and numerical sensitivity caused by an almost singular inertia matrix.

## 13. Output shaping

Output shaping uses inverse dynamics rather than prescribing the input directly [7]. A desired payload response $\theta_d(t)$ is first selected, and the corresponding trolley acceleration is then obtained from the governing equation. For the linear single-pendulum model,

$$\ddot{u}(t) = L\ddot{\theta}_d(t) + g\theta_d(t). \tag{17}$$

This approach is attractive because it allows the designer to impose smoothness and terminal conditions directly on the payload motion.

The desired output must satisfy boundary conditions that prevent free response from appearing after the maneuver. For zero initial swing, these conditions include $\theta_d(0) = 0$, $\dot{\theta}_d(0) = 0$, $\theta_d(t_f) = 0$, and $\dot{\theta}_d(t_f) = 0$. In addition, $\ddot{u}(t)$ should be compatible with the required initial and final trolley accelerations. In particular, if the final input acceleration is required to vanish, then $L\ddot{\theta}_d(t_f) + g\theta_d(t_f) = 0$, and with $\theta_d(t_f) = 0$ this requires $\ddot{\theta}_d(t_f) = 0$.

For nonzero initial conditions, the output function must match the measured or specified initial payload state. Any mismatch in $\theta_d(0)$ or $\dot{\theta}_d(0)$ generates a homogeneous response that cannot be removed by inverse dynamics alone. Output shaping is therefore most reliable when initial conditions are known and when the chosen output function has sufficient smoothness to avoid discontinuities in acceleration, jerk, and actuator force.

## 14. Nonzero initial conditions

Nonzero initial conditions are important when a new maneuver begins before the payload has fully settled [2]. In this situation, the initial angular displacement and angular velocity define an initial oscillation energy. An open-loop shaper that is designed for one specified initial state delivers the work required to move that state to the prescribed final state. If the actual initial state differs from the assumed one, the energy balance changes and residual oscillation generally remains.

This energy interpretation explains why open-loop robustness to arbitrary initial-state errors is fundamentally limited. A shaper cannot remove unknown initial oscillation energy over a continuum of initial states without information about that state or a feedback mechanism that adapts the command. Derivative constraints with respect to initial-state variables may reduce sensitivity locally, but they cannot make a fixed open-loop command insensitive to all possible initial displacements and velocities while also enforcing a specified final trolley speed and zero residual oscillation.

This limitation differs from robustness to cable-length or natural-frequency variation. Changing cable length changes the dynamic timing of the oscillator but does not by itself prescribe a different initial or final mechanical energy. Consequently, sensitivity to frequency can be reduced by shaping the command over a parameter interval, whereas unknown initial-state energy requires measurement, estimation, or feedback if broad robustness is required.

## 15. Conclusion

Input shaping for overhead cranes requires simultaneous attention to residual oscillation, command realizability, model conditioning, robustness, energy interpretation, and actuator effort. The analysis clarifies that payload motion should be treated as pendulum oscillation, that ideal impulses are mathematical devices rather than physically realizable actuator commands, and that smooth finite-duration inputs must be evaluated against acceleration, jerk, force, and power limits. Polynomial, harmonic, exponential, step-based, and output-shaped commands each offer useful advantages, but their effectiveness depends on how well their boundary conditions, frequency content, and numerical conditioning match the crane dynamics.

The main findings are that local derivative robustness is insufficient as a sole performance measure, that sensitivity should be assessed over the operating parameter interval, and that comparisons among shapers should control for maneuver duration and actuator constraints. For ideal lossless motion between fixed boundary states, net energy is determined by those states, whereas peak power, root-mean-square power, force demand, and jerk remain meaningful objectives. The scaling property of the residual-oscillation equations shows how a normalized command can be adjusted to achieve a desired cruising speed. The double-pendulum analysis further shows that nearly singular models should be reduced when a mass is negligible, and the output-shaping discussion shows that desired payload responses must satisfy initial, final, and acceleration-compatibility conditions to avoid free response.

These conclusions support crane-command designs that are physically implementable, analytically well conditioned, and robust over clearly specified operating ranges. Future studies should report actuator-effort measures together with residual oscillation, compare shapers under equal duration or equal input limits, and include measured initial-state effects when payload motion is not fully settled before a new maneuver begins.

## Conflict of Interests

Author declares that there is no conflict of interests regarding publication of this study.

## References

[1] Khorshid, E., Al-Fadhli, A., Alghanim, K., & Baroon, J. (2021). Command shaping with reduced maneuvering time for crane control. *Journal of Vibration and Control, 27*(11–12), 1311–1323.
[2] Mohammed, A., Alghanim, K., & Andani, M. T. (2021). A robust input shaper for trajectory control of overhead cranes with non-zero initial states. *International Journal of Dynamics and Control, 9*(1), 230–239.
[3] Alghanim, K., Mohammed, A., & Andani, M. T. (2019). An input shaping control scheme with application on overhead cranes. *International Journal of Nonlinear Sciences and Numerical Simulation, 20*(5), 561–573.
[4] Yang, W., Li, S., & Luo, X. (2024). Data driven vibration control: A review. *IEEE/CAA Journal of Automatica Sinica, 11*(9), 1898–1917.
[5] Majeed, M. A., Alali, M., Alghanim, K., & Alfadhli, A. (2025). Integrated time-optimal rigid-body and zero-vibration shapers on a two degrees of freedom overhead crane system. *Journal of Vibration and Control, 31*(11–12), 2340–2350.
[6] Alghanim, K., Alfadhli, A., Alshehemah, A., & Mohammed, A. (2024). Crane systems performance optimization through harmonic input shaper. *International Journal of Dynamics and Control, 12*(6), 1788–1800.
[7] Alghanim, K. (2025). Practical vibration suppression in overhead cranes using a minimalistic input shaping method. *Progress in Engineering Science, 2*(2), 100070.