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Reduce the sway of the crane payload using on-off damping radial spring-damper

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Abstract
Operating cranes is challenging because payloads can experience large and dangerous oscillations. One of the solutions to reduce the sway oscillation of the crane payload is to use a passive damper. This method is simple but has limited effectiveness because without adaptability and flexibility to prevent adverse movements and amplify the favorable ones. This paper proposes an on-off damping controller for the semi-active damper to improve the passive damper. The on-off damping control aims to amplify the radial motion of the damper to increase the energy dissipated, then reduce the sway vibration. The effectiveness of the proposed controller is verified by numerical simulations of a 2D crane.

Keywords: Coriolis force, Anti-sway control, On-off damping

1. Introduction

The crane payload suspended by cables is highly flexible. External disturbances, such as wind or motions of the support unit, can lead to excess sway. These excess sways reduce the crane's operating speed, affect the durability of the cable, and cause danger. Since the crane is a popular device, reducing swaying vibrations for the payload is extremely valuable. The anti-swing control strategies proposed in the literature is often carried out by active method including open-loop and closed-loop control. The closed-loop (feedback) techniques use the crane measurements such as swing angle to provide control commands (Kim. D. and Park. Y, 2017)(Kim D. H. and Lee J. W, 2006)(Masoud. Z, 2007)(Lee. H. H et al., 2006). The feedback control can suppress disturbance but it causes unexpected motions that make it difficult for the human operator to drive the crane. The human operator is also a feedback controller, and competing feedback controllers can degrade performance (Vaughan. J et al., 2010). On the other hand, a typical open-loop technique, namely input shaping, modifies the desired velocity command before it is issued to the crane motors (Lawrence. J and Singhose.W 2010)(Vaughan. J et al., 2008)(Blackburn. D et al., 2010). The input shaping techniques are easy to apply and do not require sensors. However, they cannot handle external disturbances or initial conditions. To overcome the limitations of the above two techniques, some researchers have proposed a combination of open- and closed-loop control.

In addition, passive damping systems are introduced to control the payload swinging in some recent studies (La. V.D et al., 2015 and 2019)(Nguyen. K.T, 2021). The radial spring-damper is a typical type of passive system that can be used in anti-sway crane control (La. V.D, 2015). It works in the principle of nonlinear Coriolis damping. The radial spring-damper is simple in application, does not conflict with the crane operator, and can counteract disturbances or initial conditions. However, without adaptability and flexibility, passive damping has some shortcomings. For example, the too-small passive damping can not prevent the resonance motion but the too-large passive damping reduces the damper motion and dissipation energy. On-off damping is the simplest way to improve the adaptability of the damping. The device producing on-off damping is much cheaper and easier to control than the one producing continuous state damping (Yalla. S.K et al., 2001). The on-off damping, as a semi-active method of vibration control, has been studied widely from the point of view of both control strategies and implemented devices, see for example (Savaresi. S.M, 2010)(Tseng. H.E. and Hrovat. D, 2015)(Casciati. F et al., 2006) and references therein.

The on-off damping strategies have been developed extensively (Savaresi. S.M, 2010)(Tseng. H.E. and Hrovat. D,

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2015)(Casciati. F et al., 2006)(La. V.D, 2012)(Couillard. M et al., 2008) but all the studied controllers are inapplicable to the problem in this paper because the damper's radial force is not codirectional with the sway motion. The novelty of this paper is to propose an on-off damping radial spring-damper for the semi-active shock absorber, suspended between the payload and the crane cable to improve the efficiency of the passive dampers.

2. Problem statement

In (La. V.D et al., 2015, 2019) the passive radial spring-damper was proposed to reduce the sway motion of a pendulum. In this paper, the on-off damping is considered as shown in Fig.1. This model is often applied in industrial house cranes or the form of tower cranes in construction.

In Figure 1, the swaying movement of the payload creates a centrifugal force. Due to the damping spring is connected between the payload and the cable, the radial movement of the payload is activated. This movement, in part, creates a Coriolis damping that acts on the swaying motion to control that movement. In case the damping is passive, if the damping is too great it will prevent the radial motion, and thus the Coriolis damping will decrease. Conversely, damping that is too small will cause a resonance motion. Where damping is turned on-off will overcome the limitations of too high or too low damping.

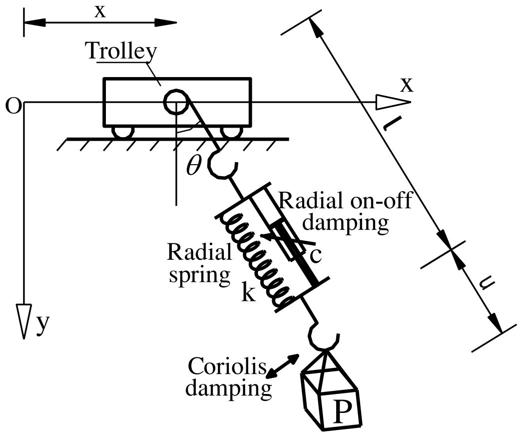

Fig. 1 Model of a tower cranes'payload attached with radial spring and on-off damper

Let us denote θ as the sway angle and u as the radial displacement (measured from static position). Denote m as the pendulum mass, k and c respectively are the spring and damper coefficients, l is the distances from the pendulum center of mass to the pivot in the static condition, and g is the acceleration of gravity, u_0 is a static deflection of spring, x *is* trolley displacement.

To illustrate the proposed technique we give the following assumptions:

- Only consider the crane in the plane with one command to be executed: Movement of the trolley, x changes (not considering cable lift command: l = const). Assume that the payload is lifted at a certain height and then the crane driver moves the trolley.
- The mass of the springs and dampers can be ignored compared to the payload mass.
- The stiffness of the cable is large enough when compared with the spring stiffness to ignore cable deformation.

On the coordinate system in Fig. 1, the position of the payload (x_P, y_P) is obtained as:

$$x_P = x + (l+u)\sin\theta\,;\; y_P = (l+u)\cos\theta \tag{1}$$

The dynamic model for this 2D crane is derived by using the Lagrange method. After some manipulations, two Lagrange equations of motion reduce to:

$$\begin{aligned} &2\dot{\theta}\dot{u} + \ddot{x}\cos\theta + \ddot{\theta}(l+u) + g\sin\theta = 0 \\ &m\ddot{x}\sin\theta + m\ddot{u} - m\dot{\theta}^2(l+u) + mg(1-\cos\theta) + ku + c\dot{u} = 0 \end{aligned} \tag{2}$$

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The equations (2) have the following non-dimensional form:

$$\begin{aligned} &2\dot{\theta}\dot{u}_n + \ddot{x}_n \cos\theta + \ddot{\theta}(1+u_n) + \sin\theta = 0 \\ &\ddot{x}_n \sin\theta + \ddot{u}_n - \dot{\theta}^2(1+u_n) + (1-\cos\theta) + \alpha^2 u_n + 2\zeta\dot{u}_n = 0 \end{aligned} \tag{3}$$

in which:

$$\omega_s = \sqrt{\frac{g}{l}}; \quad \tau = \omega_s t; \quad \alpha = \frac{\sqrt{k/m}}{\omega_s}; \quad \zeta = \frac{c}{2m\omega_s}; \quad u_n = \frac{u}{l}; \quad x_n = \frac{x}{l}; \tag{4}$$

where ω_s is the natural frequency of payload, τ is the non-dimensional time, α is the ratio of two natural frequencies, ζ respectively is the damping ratios of the damper, u_n is the non-dimensional form of radial movement. The dot operator from now denotes the differentiation with respect to normalized time τ.

The analytical formulas of the optimal parameters in the case of free oscillation were introduced in (La. V.D, 2015) and given the following:

$$\alpha_{opt} = 2; \qquad \zeta_{opt}^2 = \theta_0^2 \frac{12 + 27\zeta_{opt}^2 - 9\zeta_{opt}^4}{8\left(3\zeta_{opt}^2 + 4\right)^2} \tag{5}$$

Here α_{opt} and ζ_{opt} are optimal parameters of the passive damping spring. θ_0 is the initial oscillation angle and the character below the opt denotes the optimal parameters. The optimal damping ratio of the damper ζ_{opt} is obtained by solving the cubic equation for ζ_{opt}^2. The solution of this equation depends on the initial angle θ_0. In the actual calculation, a certain design value of θ_0 needs to be set in advance to calculate ζ_{opt}.

3. On-off damping controller

In the proposed system, we need to maximize the vibrational energy of the radial motion to dissipate the energy in the dampers. This seems logical, because the energy of the swaying oscillation is converted to the energy of the radial motion and dissipated there. The mechanical energy of the dampers is determined as follows:

$$E = \frac{1}{2}m\dot{u}^2 + \frac{1}{2}ku^2 \tag{6}$$

We convert the mechanical energy E to dimensionless form by dividing both sides (6) by ml^2 and using the symbols in (4).

$$\frac{E}{ml^2\omega_s^2} = \frac{1}{2}\dot{u}_n^2 + \frac{1}{2}\alpha^2 u_n^2 \tag{7}$$

Since $ml^2\omega_s^2$ is a constant, we can write:

$$H = \frac{E}{ml^2\omega_s^2} = \frac{1}{2}\dot{u}_n^2 + \frac{1}{2}\alpha^2 u_n^2 \tag{8}$$

The function H represents the mechanical energy (dimensionless) of the radial motion. As argued above, the H function should be as large as possible. To obtain an analytical solution, we approximate the second equation of (3) by Taylor

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expansion of the *Cos* functions and keep only the components up to the second order: $\cos\theta \approx 1 - \frac{\theta^2}{2}$

The u_n is small relative to the unit, i.e.: $1 + u_n \approx 1$

Also consider the case of free oscillation (x = const), the second equation of (3) is simplified to:

$$\ddot{u}_n + 2\zeta\dot{u}_n + \alpha^2 u_n - \dot{\theta}^2 + \frac{\theta^2}{2} = 0. \tag{9}$$

Using the equation (9) we have the expression of the mechanical energy conversion speed as follows:

$$\dot{H} = \dot{u}_n(\ddot{u}_n + \alpha^2 u_n) = \dot{u}_n\left(-2\zeta\dot{u}_n + \dot{\theta}^2 - \frac{\theta^2}{2}\right) \tag{10}$$

$$\dot{H} = -2\zeta\dot{u}_n^2 + \frac{\dot{\theta}^2 + \theta^2}{4}\dot{u}_n + \frac{3}{4}\dot{u}_n\left(\dot{\theta}^2 - \theta^2\right) \tag{11}$$

In (11) $\dot{H}$ is the rate of change of mechanical energy (dimensionless). The component $-2\zeta\dot{u}_n^2$ on the right side is the energy dissipating component. Consider the vibration in the resonance frequency $\theta = \theta_0 \sin\tau$, where θ_0 is the amplitude of the pendulum angle, therefore the component $\dot{\theta}^2 + \theta^2 = \theta_0^2(\cos^2\tau + \sin^2\tau) = \theta_0^2$ is a constant, implying the effect of a static force. The static force only produces small potential energy in the spring. The static nature can not be used to amplify the dynamic radial motion to dissipate energy. Therefore this component should not be the control signal. The last component represents the effect of dynamic force and can be used as the control signal. The power-driven controller is proposed as follows:

$$\zeta = \begin{cases} \zeta_h & \dot{u}_n(\dot{\theta}^2 - \theta^2) < 0 \\ \zeta_l & \dot{u}_n(\dot{\theta}^2 - \theta^2) > 0 \end{cases} \tag{12}$$

The control law (12) can be explained as follows. If $\dot{u}_n(\dot{\theta}^2 - \theta^2)$ is negative, it is likely that the derivative of H is negative, which is not desired since the mechanical energy of radial motion is decreasing. The larger on-damping ζ_h is used to prevent this trend. Otherwise, the smaller off-damping ζ_l will be used.

4. Numerical calculation

In this section, we perform numerical calculations to illustrate the effectiveness of the power-driven controller. Numerical computations are carried out for the full nonlinear differential equations (2). In all cases the ratio of the natural frequencies α is chosen to be 2, which is the optimum value of the free oscillating passive dampers defined in (La. V.D, 2015), the initial conditions are chosen to be zero, except for the initial sway angle θ_0

The calculated values are selected as follows:

- m = 5kg, l = 9m;

- The initial sway angle $\theta_0 = 30^0$.

- On-off damping $\zeta_h = 20\zeta_{opt}$, $\zeta_l = 0.05\zeta_{opt}$;

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$c_h = 20c_{opt}$, $c_l = 0.05c_{opt}$.

- Radial springs, designed for large oscillation angles up to 60^0.

The results of the optimal parameters are calculated by: ζ_{opt} = 0.3296, α_{opt} =2,

$$k_{opt} = 4mg/l; \qquad c_{opt} = 2\zeta_{opt} m\sqrt{g/l}.$$

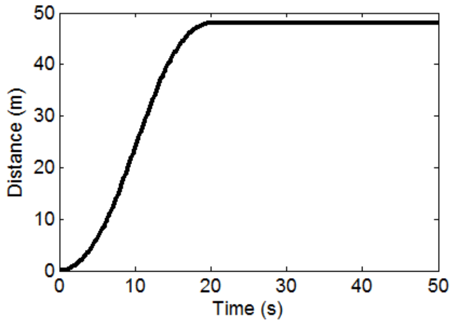
The command to move the trolley is as follows:

- In phase 1, from 0 to 8 seconds, the trolley moves from 0 to 16m, the velocity changes from 0 to 4m/s.

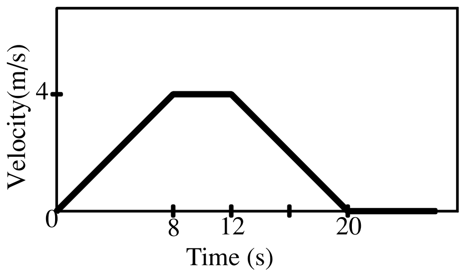
- In phase 2, from 8 to 12 seconds, the trolley moves from 16m to 32m, constant velocity is 4m/s.

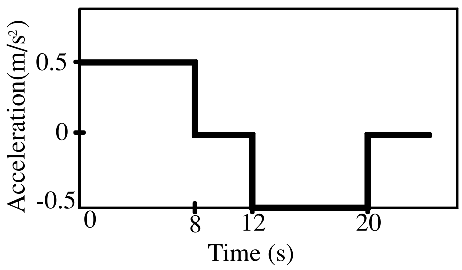
- In phase 3, from 12s to 20s, the trolley moves from 32m to 48m, the velocity decreases from 4m/s to 0.

The operation process of this trolley is shown in figure 2.



a) Horizontal movement of the trolley





b) Velocity command

c) Acceleration command

Fig. 2 The operation process of the trolley

For the convenience of notation, we will consider simulation cases: Case 1: Without a Damper (no spring and no damping); Case 2: Optimal passive; Case 3: On-off damping.

Fig. 3a–d shows the time histories of the pendulum angle θ, the Coriolis force, the on-off damping c, and the payload horizontal displacement, with two cases: the passive dampers use optimal parameters compared to the semi-active dampers.

In figure 3a, the sway angle in the case of on-off damping is smaller than in the case of passive optimal damping. Proved effect of reducing swaying oscillation of the proposed method.

In figure 3b, Coriolis force in case the on-off damping is greater than the case of passive optimal damping. Show that when the damping is controlled, it will increase Coriolis force in the direction of the oscillation, leading to reduce the swaying vibration.

In figure 3d, the horizontal movement of the payload is shown. The swaying angle in the case of the on-off damping is less than in the case of passive optimal damping. This shows that when the damping is controlled, the swaying

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oscillation of the payload has decreased.

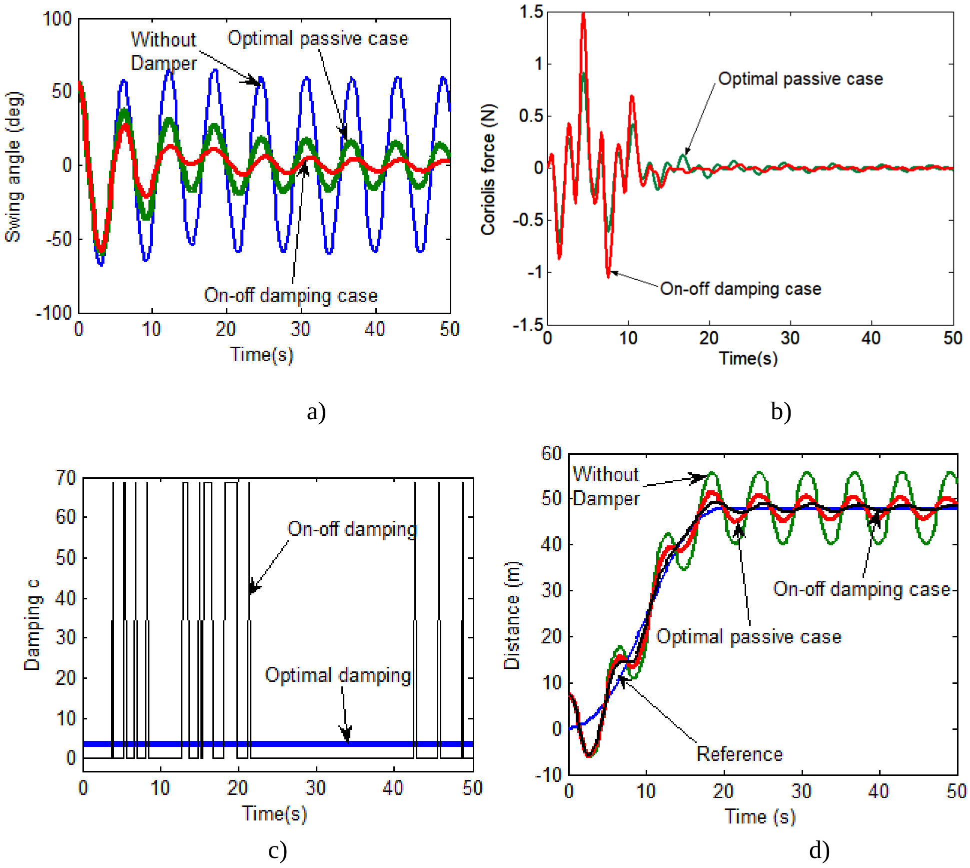

Fig.3 Time histories of: (a) pendulum angle, (b) Coriolis term, (c) damping coefficient, and (d) horizontal movement of payload;

We also calculate the remaining oscillation angle after 4 periods. The comparison results are shown in Table 1.

Table 1 Maximum angle (degree) after 4 periods (percentile beside shows the reduction of swinging).

Case	Without Damper (Case 1)	Optimal passive (Case 2)	On-off damping (Case 3)
θ_0=30^o,c_h=20c_{opt} c_l =0.05c_{opt}	59.81 (0%)	17.84 (70.18%)	5.96 (90.03%)
θ_0=30^o, c_h =5c_{opt} c_l =0.2c_{opt}	59.81 (0%)	17.84 (70.18%)	8.86 (85.18%)
θ_0=30^o, c_h =2c_{opt} c_l =0.5c_{opt}	59.81 (0%)	17.84 (70.18%)	12.02 (79.9%)
θ_0=20^o, c_h=20c_{opt} c_l =0.05c_{opt}	58.85 (0%)	24.23 (58.83%)	9.45 (83.95%)
θ_0=20^o, c_h =5c_{opt} c_l =0.2c_{opt}	58.85 (0%)	24.23 (58.83%)	10.23 (82.61%)
θ_0=10^o,c_h=20c_{opt} c_l =0.05c_{opt}	58.57 (0%)	37.19 (36.5%)	18.51 (68.39%)
θ_0=10^o, c_h =5c_{opt} c_l =0.2c_{opt}	58.57 (0%)	37.19 (36.5%)	20.04 (65.79%)

The results give the following remarks:

- In case 2,3 (with damper, on-off damping), the performance is better if the initial angle is larger (90.03% in comparison with 83.95% and 68.39%).

- In case 3 (with on-off damping), the performance is better if the damping coefficient is turned on and off within a larger range (90.03% in comparison with 85.18% and 79. 9%).

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- In case 3 (with on-off damping), the performance is better than in case 2 (with damper with optimal passive damping) and case 1 (without damper).

In summary, the semi-active dampers are more effective in reducing the swaying oscillation of the payload compared to the passive dampers.

5. The robustness of the proposed controller

In theory, the optimal values of k_{opt} and c_{opt} are determined with certain parameters. However, in practice, the values k_{opt} and c_{opt} often change due to several reasons, for example, the changes of cable length, payload mass, and the fabrication errors of k, c. The errors of k_{opt} and c_{opt} have a significant effect on the efficiency of both passive and on-off dampings. To evaluate the robustness of the proposed controller, in this section, we consider the 25% error of the k_{opt} and c_{opt} in the simulation. Simulation results are shown in Table 2 and 3. Comparing table 2, 3 with table 1, we can see that the effect of reducing oscillation in cases 2 and 3 has decreased (50.89% in comparison with 70.18%, 81.32% in comparison with 90.03%). Because the damper parameters are not optimal. However, case 3 is still more effective than case 2 in all cases. This proves the robustness of the proposed controller.

Table 2 Maximum angle (degree) after 4 periods (percentile beside shows the reduction of swinging)
Case k_{opt} and c_{opt} are increased by 25%

Case	Without Damper (Case 1)	Optimal passive (Case 2)	On-off damping (Case 3)
θ_0=30°,c_h=20c_{opt},c_l =0.05c_{opt}	59.81 (0%)	29.37 (50.89%)	11.17 (81.32%)
θ_0=30°, c_h =5c_{opt} c_l =0.2c_{opt}	59.81 (0%)	29.37 (50.89%)	20.03 (66.52%)
θ_0=10°,c_h=20c_{opt} ,c_l =0.05c_{opt}	58.57 (0%)	46.23 (21.07%)	24.66 (57.9%)
θ_0=10°, c_h =5c_{opt} c_l =0.2c_{opt}	58.57 (0%)	46.23 (21.07%)	34.25 (41.52%)

Table 3 Maximum angle (degree) after 4 periods (percentile beside shows the reduction of swinging)
Case k_{opt} and c_{opt} are reduced by 25%

Case	Without Damper (Case 1)	Optimal passive (Case 2)	On-off damping (Case 3)
θ_0=30°,c_h=20c_{opt}, c_l =0.05c_{opt}	59.81 (0%)	21.25 (64.47%)	7.73 (87.08%)
θ_0=30°, c_h =5c_{opt}, c_l =0.2c_{opt}	59.81 (0%)	21.25 (64.47%)	13.91 (76.75%)
θ_0=10°,c_h=20c_{opt}, c_l =0.05c_{opt}	58.57 (0%)	39.86 (31.94%)	26.23 (55.22%)
θ_0=10°, c_h =5c_{opt}, c_l =0.2c_{opt}	58.57 (0%)	39.86 (31.94%)	31.24 (46.66%)

6. Conclusion

This paper has proposed an on-off damping coefficient controller for the semi-active dampers. This controller is built on the idea of amplifying the motion in the radial direction to speed up the energy dissipation process, thereby reducing the swaying vibration of the crane payload. The results of numerical simulation have shown the effect of the proposed controller in reducing the sway angle of the crane payload compared to the passive method.

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