Vibration suppression of fixed-time jib crane maneuvers

Gordon G. Parker, Ben Petterson, Clark R. Dohrmann, Rush D. Robinett

Sandia National Laboratories
P.O. Box 5800 Mail Stop 0439
Albuquerque, NM 87185-0439

ABSTRACT

A jib crane consists of a pendulum-like end line attached to a rotatable jib. Within this general category of cranes there exist devices with multiple degrees of freedom including variable load-line length and variable jib length. These cranes are commonly used for construction and transportation applications. Point-to-point payload maneuvers using jib cranes are performed so as not to excite the spherical pendulum modes of their cable and payload assemblies. Typically, these pendulum modes, although time-varying, exhibit low frequencies. The resulting maneuvers are therefore performed slowly, contributing to high construction and transportation costs. The crane considered here consists of a spherical pendulum attached to a rigid jib. The other end of the jib is attached to a direct drive motor for generating rotational motion. A general approach is presented for determining the open-loop trajectories for the jib rotation for accomplishing fixed-time, point-to-point, residual oscillation free, symmetric maneuvers. These residual oscillation free trajectories purposely excite the pendulum modes in such a way that at the end of the maneuver the oscillatory degrees of freedom are quiescent. Simulation results are presented with experimental verification.

Keywords: Rotary Crane, Control, Swing-Free, Oscillation-Free, Input Shaping

1. INTRODUCTION

Construction and transportation cranes can generally be grouped into one of two categories based on their configuration. The first category consists of overhead, gantry cranes. These systems incorporate a trolley which translates in the horizontal plane. Attached to the trolley is a load-line for payload attachment. Typically, they have varying load-line length capabilities. The second category consists of rotary cranes. As the name implies, the load-line attachment point undergoes rotation. Other degrees of freedom may exist such as translation of the load-line attachment point along the jib, variable load-line length, or if the jib is replaced by a boom, the characteristic boom rotational motion, known as luffing*.

Auernig and Troger¹ consider time optimal payload maneuvers of a gantry crane undergoing trolley translation and load-line length change. The coupled, nonlinear equations of motion and adjoint equations, obtained from the application of Pontryagin's maximum principle, are solved analytically for the cases of constant and variable hoisting speeds. In both cases the maneuvers are developed such that the payload is residual oscillation free. Moustafa and Ebeid² demonstrate a state feedback controller for damping load sway for a gantry crane configured to move along two orthogonal directions in the horizontal plane. This work is expanded on by Ebeid et. al.³ to incorporate actuator dynamics into the crane model. Fliess et. al.⁴ investigate a feedback linearizing controller for a one-dimensional gantry crane. Trolley traversal and load-line length changes are considered. Simulation results indicate the ability of the closed-loop controller to control load sway for relatively slow maneuvers. This same system is examined by Nguyen⁵ where simulation and experimental results of a nonlinear state-feedback controller is used. Small motions are assumed about a specified operating point. This allows for decoupled equations of motion and decoupled controller design.

Sakawa and Nakazumi⁶ investigate a rotary crane undergoing hub and boom rotation and load hoisting using a combined open and closed-loop approach. The open-loop input to the crane is designed based on a postulated set of functions such that the sway motion of the load is excited minimally. The closed-loop controller is then switched on when the maneuver is near the end, providing significant sway damping. Vaha et. al.⁷ generate suboptimal, minimum time inputs for a rotary crane. Tracking is achieved via a state feedback control law. However, radial sway, due to centripetal acceleration of the payload, is not compensated. Souissi and Koivo⁸ consider a rotary crane undergoing a boom-rotation-boom maneuver using a proportional-

*Boom rotational motion, referred to as luffing, causes the tip of the boom to move vertically.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

integral-derivative controller similar to Fliess et.al. ⁴. The simulation model considers both radial and tangential payload sway, however, the control strategy used results in residual load oscillation.

In this paper a procedure for generating open-loop inputs for a rotary jib crane is introduced. The primary maneuver of interest is the residual oscillation, point-to-point movement of a payload. The input angular acceleration to the jib motor is postulated as having a pulse-coast-pulse form. A numerical optimization technique is used to generate the parameters which define this pulse sequence such that the maneuver is residual oscillation free. Of secondary interest are maneuvers where the payload oscillates in a specified manner at the end of the maneuver. Again, the pulse sequences are designed via numerical optimization. Experimental results for these maneuvers verify the results of the numerical optimization procedure.

2. CRANE DESCRIPTION

The crane considered here consists of a rotatable jib with a load-line attached to the end. A mass, representing a payload, is attached to the end of the load-line. This apparatus is shown in Fig. 1

The $\hat{a}_1, \hat{a}_2, \hat{a}_3$ coordinate system is attached to the jib and rotates about the hub with angular rate $\dot{\gamma}$. The rotation angles θ_1, θ_2 are defined as rotations of the load-line about the $\hat{a}_1, \hat{a}_2$ axes respectively. The attachment point of the load-line to the jib is at a distance x from the center of rotation of the hub. The load-line has length L and the payload has mass m . The two rotations will from now on be referred to as the radial (θ_2) and the tangential (θ_1). The input to this system is the hub rotation trajectory $\gamma(t)$ and the outputs are the rotation angles of the load-line, $\theta_1(t), \theta_2(t)$.

The experimental system uses a constant velocity, computer controlled, direct-drive, motor. Physical values for x, m, L are given in Table 1.

Figure 1. The rotary jib crane.

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

Parameter	Units	Value
x	m	0.659
L	m	0.251
m	kg	0.05

Table 1. Physical quantities used for the experimental apparatus.

As mentioned in the introduction, several maneuvers are examined all of which consist of a 90° jib rotation. The first is designed such that the payload is residual oscillation free. The second allows only tangential oscillation at the end of the maneuver, while the third allows only radial oscillation of the payload.

3. EQUATIONS OF MOTION

Using the rotating $\hat{a}_1, \hat{a}_2, \hat{a}_3$ coordinate system, the vector from the center of rotation of the hub to the payload is given by

$$\begin{aligned} \underline{p}(t) = & (x - L \cos[\theta_1(t)] \sin[\theta_2(t)]) \hat{a}_1 + \\ & L \sin[\theta_1(t)] \hat{a}_2 - \\ & L \cos[\theta_1(t)] \cos[\theta_2(t)] \hat{a}_3 \end{aligned} \quad (1)$$

Lagrange's equations

$$\frac{d}{dt} \left(\frac{\partial \mathcal{L}}{\partial \dot{\theta}_i} \right) - \frac{\partial \mathcal{L}}{\partial \theta_i} = 0 \quad (2)$$

are applied to this system where the Lagrangian $\mathcal{L}$

$$\mathcal{L} = T - V \quad (3)$$

is the difference between the kinetic energy T and the potential energy V .

With the aid of Fig. 1 and Eq. 1, the kinetic energy can be found as

$$\begin{aligned} T = & \frac{1}{2} m L^2 \left[\dot{\theta}_1^2 + \left(\dot{\theta}_2 \cos \theta_1 \right)^2 - 2 \dot{\theta}_1 \dot{\theta}_2 \sin \theta_2 + (\dot{\gamma} \sin \theta_1)^2 + \right. \\ & 2 \dot{\gamma} \dot{\theta}_2 \sin \theta_1 \cos \theta_1 \cos \theta_2 + \left(\frac{\dot{\gamma} x}{L} \right)^2 + (\dot{\gamma} \cos \theta_1 \sin \theta_2)^2 - \\ & \left. \frac{2 \dot{\gamma}^2 x}{L} \cos \theta_1 \sin \theta_2 + \frac{2}{L} \dot{\gamma} x \dot{\theta}_1 \cos \theta_1 \right] \end{aligned} \quad (4)$$

and the potential energy is

$$V = -mgL \cos \theta_1 \cos \theta_2 \quad (5)$$

Substituting Eq. 3, Eq. 4, and Eq. 5 into Eq. 2 yields the two equations of motion for the oscillatory degrees of freedom θ_1, θ_2

$$\begin{bmatrix} \ddot{\theta}_1 \\ \ddot{\theta}_2 \end{bmatrix} + 2 \begin{bmatrix} 0 & -\dot{\gamma} \\ \dot{\gamma} & 0 \end{bmatrix} \begin{bmatrix} \dot{\theta}_1 \\ \dot{\theta}_2 \end{bmatrix} + \begin{bmatrix} \left(\frac{g}{L} - \dot{\gamma}^2\right) & -\ddot{\gamma} \\ \ddot{\gamma} & \left(\frac{g}{L} - \dot{\gamma}^2\right) \end{bmatrix} \begin{bmatrix} \theta_1 \\ \theta_2 \end{bmatrix} = -\frac{x}{L} \begin{bmatrix} \dot{\gamma} \\ \dot{\gamma}^2 \end{bmatrix} \quad (6)$$

where small angle assumptions have been exploited for θ_1 , θ_2 and their derivatives.

In the following, a numerical solution for the hub rotation $\gamma(t)$ yielding the desired residual motion of the payload, is found relying on the equations of motion of Eq. 6.

4. MANEUVER GENERATION

Maneuvers are generated by first postulating the jib angular acceleration profile $\ddot{\gamma}(t)$ having the pulse-coast-pulse form shown in Fig. 2.

The choice of this form is based on similar work regarding residual oscillation free slewing of a flexible rod¹⁰. In that study, the slewing was performed in the horizontal plane. However, due to gravitational and centrifugal effects, vibration was excited in both the horizontal and vertical directions. It was shown how a rotation profile could be generated such that vibration in both directions was suppressed at the end of the maneuver. While the physics of the pendulum problem considered here and the slewing rod are similar, the coupling between modes of oscillation is significantly different.

The angular acceleration profile of Fig. 2 is uniquely described by the amplitude of the first pulse, A_1 , the duration of the first pulse, T_1 , the coast duration, T_2 and the duration of the second pulse, T_3 . The amplitude of the second pulse is determined by the requirement of zero jib angular velocity at the end of the maneuver, or

$$A_2 = \frac{A_1 T_1}{T_3} \quad (7)$$

These four parameters are obtained using the recursive quadratic programming numerical optimization code developed by Hopper¹¹. The minimized cost function has the form

$$J = w_1 \theta_1^2(t_f) + w_2 \dot{\theta}_1^2(t_f) + w_3 \theta_2^2(t_f) + w_4 \dot{\theta}_2^2(t_f)$$

Figure 2. Postulated jib angular acceleration history.

where the weighting coefficients w_i are chosen based on the type of residual oscillation desired. Equality constraints are used to ensure the completion of the 90° jib rotation. Evaluation of the cost function and the constraint equations requires the integration of the equations of motion in Eq. 6. This is performed using fourth order Runge-Kutta integration with a time step of 0.001 seconds.

5. RESULTS

Table 2 shows the parameters obtained via the optimization process which result in the three maneuvers of interest. Plots of jib rotation are shown in Fig. 3 through Fig. 5. The angular acceleration profiles implemented on the hardware, described in Section 2, produce the desired maneuvers with no modification of the inputs. This implies an excellent ability to model and simulate this system. The absence of residual oscillation was determined by observation during testing. Due to the agreement between simulation and experiment, simulation plots are presented.

Parameter	No Oscillation	Radial Oscillation Only	Tang. Oscillation Only
A_1 (rad/s/s)	1.2798	2.3333	1.3528
T_1 (s)	0.90684	0.52440	0.88550
T_2 (s)	0.44655	0.58515	0.50637
T_3 (s)	0.90684	0.86981	0.72376

Table 2. Jib acceleration parameters for the maneuvers considered.

The residual oscillation free maneuver of Fig. 3 uses an acceleration profile symmetric in time about the half period of the maneuver. This is consistent with the results obtained by Petterson and Robinett¹⁰. Allowing pendulum sway in either the radial or tangential directions only, requires nonsymmetric acceleration histories as in Fig. 4 and Fig. 5. Plots of the pendulum sway are shown in Fig. 6 through Fig. 8. Centrifugal effects during the maneuver are illustrated by Fig. 9 through Fig. 11 which show the projection of the payload and the jib end onto the horizontal surface. In all cases the load is observed moving initially outward from the tip path.

Figure 3. No oscillation maneuver.

Figure 4. Radial oscillation only maneuver.

Figure 5. Tangential oscillation only maneuver.

Figure 6. No oscillation maneuver.

Figure 7. Radial oscillation only maneuver.

Figure 8. Tang. oscillation only maneuver.

Figure 9. No oscillation maneuver.

Figure 10. Radial oscillation only maneuver.

Figure 11. Tang. oscillation only maneuver.

The acceleration profiles developed here purposely excite the oscillatory motion of the load. This is illustrated by investigating the Fourier series of the acceleration history. Examination of the quasi-static stiffness matrix of Eq. 6 throughout the residual oscillation-free maneuver indicates only slight changes in the quasi-static natural frequencies of the load. Specifically, the nominal (nonrotating) natural frequency in both radial and tangential directions is 0.995 Hz . The initial acceleration pulse, with zero velocity, results in a slight increase of this frequency. However, the softening effect of the angular velocity during the maneuver causes the quasi-static natural frequency to decrease to 0.978 Hz at the half-period of the maneuver. This change in quasi-static natural frequency is symmetric in time about the half-period.

Fig. 12 shows a 20-term Fourier sine series expansion of the input acceleration history. The first 5 coefficients of the jib angle history, obtained from integrating the acceleration history, are given in Table 3.

Frequency (Hz)	Fourier Coefficient
4.424×10^{-1}	1.912×10^{-1}
8.849×10^{-1}	8.915×10^{-3}
1.327	2.784×10^{-3}
1.770	2.950×10^{-3}
2.212	6.152×10^{-07}

Table 3. First 5 terms of the Fourier expansion.

The second and third terms are near the quasi-static frequency of the system, indicating excitation of that frequency.

Figure 12. Acceleration and Fourier expansion.

Finally, some mention of the robustness of this technique is in order. It is expected that the performance of the system will suffer due to inaccuracies of the model. Sensitivity analyses indicate that the most important model parameter is the load-line length. Small errors in this quantity can lead to residual vibration, whereas similar errors in jib length can be tolerated without significant degradation of performance. Sensitivity of the response due to the discontinuous nature of the acceleration profile can be reduced by using a smoothed profile such as a versine function. This technique has been used for vibration suppression of a flexible rod, and is applicable here.¹²

6. SUMMARY

A method for generating residual oscillation free maneuvers for a rotary jib crane has been presented. This method uses the model of the system in conjunction with a numerical optimization process to generate parameters describing the jib angular acceleration profile. Maneuvers obtained using this procedure were experimentally verified demonstrating not only the usefulness of this method but also the ability to accurately simulate such a system. The input profiles were also shown to purposely excite the quasi-static frequencies of the payload in order to decrease the operating time of the jib crane (i.e., decrease construction costs).

7. ACKNOWLEDGMENTS

This work performed at Sandia National Laboratories is supported by the U.S. Department of Energy under contract DE-AC04-94AL85000.

8. REFERENCES

1. J.W. Auernig and H. Troger, "Time Optimal Control of Overhead Cranes with Hoisting of the Load," *Automatica*, Vol. 23, No. 4, pp. 437-447, 1987.
2. K.A.F. Moustafa and A.M. Ebeid, "Nonlinear Modeling and Control of Overhead Crane Load Sway," *Journal of Dynamic Systems, Measurement, and Control*, Vol. 110, pp. 266-271, 1988.
3. A.M. Ebeid, K.A.F. Moustafa and H.E. Emara-Shabaik, "Electromechanical Modelling of Overhead Cranes," *International Journal of Systems Science*, Vol. 23, No. 12, pp. 2155-2169, 1992.
4. M. Fliess, J. Levine P. Rouchon, "A Simplified Approach of Crane Control via a Generalized State-Space Model," *IEEE Proceedings of the 30th Conference on Decision and Control*, Brighton, England, pp. 736-741, 1991.
5. H.T. Nguyen, "State-Variable Feedback Controller for an Overhead Crane," *Journal of Electrical and Electronics Engineering*, Australia, Vol. 14, No. 2, pp. 75-84, 1994.
6. Y. Sakawa and A. Nakazumi, "Modeling and Control of a Rotary Crane," *Journal of Dynamic Systems, Measurement, and Control*, Vol. 107, pp. 200-206, 1985.
7. P. Vaha, S. Pieska and E. Timonen, "Robotization of an Offshore Container Crane," *Robots: Coming of Age, Proceedings of the 19th ISIR International Symposium*, pp. 637-648, 1988.
8. R. Souissi and A.J. Koivo, "Modeling and Control of a Rotary Crane for Swing-Free Transport of Payloads," *The First IEEE Conference on Control Applications*, Dayton, OH, pp. 782-787, 1992.
9. C.T. Chen, "Linear System Theory and Design", Holt, Rinehart and Winston, New York, NY, 1984.
10. B.J. Petterson and R.D. Robinett, "Model-Based Damping of Coupled Horizontal and Vertical Oscillations in a Flexible Rod," *Journal of Intelligent and Robotic Systems*, Vol. 4, pp. 285-299, 1991.
11. M.J. Hopper, Harwell Subroutine Library, AERE-R9185, UKAEA, Harwell, Oxon., UK, 1978.
12. G.G. Parker, R. Eisler, J. Phelan, and R.D. Robinett, "Input Shaping for Vibration-Damped Slewing of a Flexible Beam Using a Heavy-Lift Hydraulic Robot," Presented at the *1994 American Control Conference*, Baltimore, Maryland, 1994.