Dynamics and Zero Vibration Input Shaping Control of a Small-Scale Boom Crane

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Abstract—Crane are vital to many manufacturing and material-handling processes. However, their physical structure leads to flexible dynamic effects that limit their usefulness. Large payload swings induced by either intentional crane motions or external disturbances decrease positioning accuracy and can create hazardous situations. Mobile boom cranes are one of the most dynamically complicated types of cranes. Boom cranes cannot move the payload in a straight line by actuating only one axis of motion because they have rotational joints. This paper presents a nonlinear model of a mobile boom crane. Then, a large range of possible motions is analyzed to investigate the dynamic behavior of the crane. A command-shaping control technique is applied to this nonlinear machine and its effectiveness is analyzed. Experimental results confirm the predicted oscillation phenomenon.

I. INTRODUCTION

Crane are used extensively throughout the world in a variety of applications. However, the flexible nature of their physical design degrades their effectiveness, safety, and throughput. Oscillation induced by both intentional motions of the crane itself and by external disturbances is a major limitation. A small-scale mobile boom crane is shown in Figure 1. The crane has a mobile base, a slewing base, which rotates with respect to the mobile base, and a boom, which also moves by rotation. Like any crane, it also has a suspension cable that lifts the payload.

While numerous researchers have proposed using feedback control to limit crane payload oscillation, success has been limited. Unfortunately, a fundamental conflict exists between computerized feedback control and human operators. Crane operators are feedback controllers; they continually adjust the input command to achieve a desired response. Any additional computer-based feedback control can conflict with the actions of the human operator. A second significant difficulty of using feedback control on cranes is the difficulty of measuring the motion of the payload. For feedback to perform well, accurate measurements of the states of the payload are required. When considering the implementation of such sensors on real machines with varying payloads, locations, disturbances, and operating environments, the task can be very challenging.

Input shaping [1–6], on the other hand, is a control method that is highly compatible with human operators [7, 8] and can drastically reduce motion-induced oscillations. Input shaping is implemented by convolving a sequence of impulses, called the input shaper, with the desired reference command. This process is shown in Figure 2. The shaped command can move the crane without causing oscillations. Input shaping has been successfully applied to bridge [7–12], tower [13–16], boom [17, 18] and container cranes [19].

Boom cranes are both highly complex and useful. Unfortunately, there has been relatively little emphasis placed on understanding and controlling the dynamics of these machines. This paper will analyze and discuss the complex dynamics of boom cranes. The next section presents a nonlinear model of a mobile boom crane. Then, the slewing and luffing dynamics are investigated in detail. Input shaping is applied to these motions and its effectiveness is analyzed. Experimental results confirm the predicted oscillation phenomenon.
II. MOBILE BOOM CRANE MODEL

Figure 3 shows a sketch of a mobile boom crane. It has the same basic components as the small-scale mobile boom crane shown in Figure 1. The bottom portion is composed of two rectangular bodies. The lower, dark-colored rectangle is the mobile base. It is treated as a car with rear-wheel drive and front-wheel steering. The distance between the two axles is $l_{axle}$. The top, light-colored rectangle is the slewing base that rotates with respect to the mobile base. The distance from the geometric center of the lower base to the slewing center of rotation is $l_b$. The boom is attached to this slewing base and is also capable of rotation in a vertical plane that is perpendicular to the slewing base. The distance from the slewing center of rotation to the attachment point of the boom is $l_h$. The payload is supported by a cable attached near the end of the boom. The distance from the boom rotation point to the cable suspension point is $l_{boom}$.

The inputs to the model are the velocity of the base, $v$, and the accelerations for the steering angle, $\psi$, the slewing angle about the mobile base, $\theta$, the luffing angle of the boom, $\gamma$, and the suspension cable length, $\ell$. The rotational velocity of the mobile base, $\omega$, and the Cartesian components of the mobile base velocity, $\dot{x}$ and $\dot{y}$, are obtained as functions of $v$ and $\psi$. The important outputs are the swing angles in the radial direction, $\phi$, and the tangential direction, $\beta$. The model assumes the suspension cable is massless and inelastic. The body of the crane is significantly more massive than the payload, so that the payload is unable to affect the motion of the crane base. There is no damping in the cable swing. Motor and transmission dynamics are also not modeled.

Using a commercial dynamics package, the equations of motion for this system were developed. The equations for the radial swing angle, $\phi$, and the tangential swing $\beta$ are:

\[
\begin{align*}
-2\ell\cos(\beta)\dot{\phi} &= 2g\sin(\phi) + 4\ell\cos(\beta)\dot{\phi} - \sin(\beta)\cos(\phi)(\omega + \dot{\theta}) + 2\cos(\phi)\sin(\omega + \dot{\theta})\dot{\phi} + 2\cos(\phi)\cos(\omega + \theta)\dot{x} + 2\ell\sin(\beta)\sin(\phi)(\omega + \dot{\theta}) - \sin(\beta)\cos(\phi)(\omega + \dot{\theta}) - 2h_{boom}\cos(\gamma - \phi)(\gamma^2 + \cos(\gamma^2)(\omega + \dot{\theta})^2 - 2f_{boom}\cos(\omega + \theta) - 2f_{boom}\cos(\beta)\cos(\phi)(\omega + \dot{\theta}) - 2(\beta + \sin(\phi)(\omega + \dot{\theta})) - 2f_{boom}\sin(\gamma - \phi)(\sin(\gamma)\cos(\gamma)(\omega + \dot{\theta})^2 + \gamma).
\end{align*}
\]

III. BOOM CRANE DYNAMICS

In the next three subsections, the dynamics of the boom crane are investigated. The slewing and luffing motions of the crane are introduced and analyzed. Input shaping is also applied to these motions and its effectiveness is quantified.

A. Slewing

Slewing motion of the boom crane is defined as the rotation, $\theta$, of the slewing base about the mobile base. During numerical simulation of the boom crane, the maximum slewing velocity is limited to 10 deg/sec and the maximum acceleration is 25 deg/sec². The baseline reference command is a trapezoidal-velocity profile (bang-coast-bang acceleration). The residual stage is defined as the time frame from the end of the decelerating pulse to the end of the simulation. The residual vibration amplitude is the maximum displacement of the payload relative to the overhead suspension point.

Figure 4 shows the location of the payload relative to the suspension point during a 10° slewing motion. The luffing angle was held constant at 45° and a constant suspension cable length of 1 meter was used. Figure 4 demonstrates that the payload residual oscillation precesses. This effect occurs because the centripetal force of the slewing motion causes the payload to oscillate in the radial direction. When this radial oscillation is combined with the tangential oscillation caused by the base slewing acceleration, the payload precesses during the residual stage.

In order to better understand the dynamics of the slewing motion, the relationship between residual vibration amplitude, slewing distance, and luff angle were investigated. The
crane started from rest at the 0° slew position (with the boom pointing directly forward). Then, it was slewed with a constant luff angle using trapezoidal velocity commands. Figure 5 shows how the residual vibration amplitude changes as a function of the luff angle and slewing distance. As the luff angle decreases, the end of the boom extends farther away from the base and the maximum residual vibration amplitude increases. When the payload is farther from the rotating base, it travels faster and covers a longer distance for a given slewing command. The higher velocity requires higher tangential and centripetal forces, causing more swing. This relationship; however, is not linear. When the boom is pointing straight up, with a luff angle of 90°, the oscillation is at its smallest; however, it is not zero. This is because the center of slewing and luffing are a small distance, $l_b$, apart. Hence, there is a small amount of oscillation even when the boom is straight up.

The relationship between residual vibration amplitude and slewing distance is complex, as shown in Figure 6. After a 90° slew, the radial and tangential directions have exactly switched. As a result, the residual vibration amplitude is almost symmetrical about a slewing distance of 90°. There are numerous peaks and troughs as the slewing distance is varied. Although the slewing motion is nonlinear, the trends in the peaks and troughs can be explained by using a simple linear second-order model. Assume that the input to such a system is two pulses in acceleration that form a bang-coast-bang command (trapezoidal velocity). Each acceleration pulse induces oscillation. For a linear system, the magnitude of oscillation caused by each pulse is equal in magnitude and sometimes in phase and sometimes out of phase with each other. The amplitude of residual vibration will then contain peaks and troughs as it is plotted versus the move distance. In the nonlinear slewing motion; however, the swings produced by the acceleration and deceleration are not quite equal in magnitude. The peaks in residual vibration amplitude arise when the pulse responses are in phase and add up to produce more swing. The troughs occur when the two responses are out of phase and partially cancel each other, resulting in low residual swing. Figure 6 shows how these two scenarios alternate with slewing distance.

An iterative routine was performed to find the residual vibration amplitude for a representative subset of all possible slewing commands. The slewing motion was simulated for distances between 0° and 180°, using constant luffing angles between 0° and 90°. The suspension cable length was held constant at 1 m. To investigate the effectiveness of input shaping on controlling the oscillation induced by the nonlinear slewing motion, the same maneuvers were repeated, but the reference commands were convolved with a two-impulse Zero Vibration (ZV) input shaper [1, 2]. The ZV shaper was designed for a natural frequency of 3.13 rad/sec, which corresponds to a simple pendulum with a 1 m long suspension cable. Figure 7 shows the residual vibration amplitude of each motion for the unshaped and ZV-shaped commands. The residual vibration amplitude from the shaped commands is shown by the solid surface. The residual vibration amplitude from the shaped commands is shown by the solid surface. The residual vibration amplitude from the trapezoidal velocity commands has been overlaid using a mesh. Figure 7 demonstrates the effectiveness of input shaping on this nonlinear slewing motion. The residual vibration amplitude was reduced for every slewing distance and luff angle. Over the entire space shown in Figure 7, input shaping reduced residual vibration by an average of 95%.

B. Luffing

Luffing motion of the mobile boom crane is defined as the rotation, $\gamma$, of the boom in a vertical plane. The maximum velocity of the luffing motion was limited to 6.67
deg/sec and the maximum acceleration was limited to 83.33 deg/sec^2.

Unlike the slewing motion, luffing results only in oscillation in the radial direction. For example, the payload was luffed upward from an initial angle of 30° to a final angle of 60°. The slewing angle was set to zero and the suspension cable length was held constant at 1 meter. The payload simply swings back and forth with approximately 13 cm displacement. The residual vibration amplitude caused by luffing motions is generally smaller than those produced by slewing commands.

The luffing dynamics are complicated because the magnitude of payload oscillation caused by a specific move distance is dependent on both the initial luff angle and the final luff angle. The direction of motion, upward or downward, can also be significant because the effect of gravity changes. The changes in the net applied forces not only change the oscillation amplitude, but also the oscillation frequency during accelerations [20]. However, this only becomes significant if the luffing acceleration is a substantial fraction of the gravitational acceleration.

Figure 8 shows the relationship between residual vibration amplitude and luffing distance for upward motion from four different initial luff angles. The reason all lines do not span the entire Luffing-Distance axis is that the luffing angles were limited to between 0° and 90°. For example, if the initial luff angle is 75°, then the maximum allowable upward luffing distance is only 15°, as shown by the dotted line in Figure 8.

Peaks and troughs appear in the data for the luffing motion. As explained before, these peaks and troughs appear because the oscillation caused by the starting and stopping forces add up when in phase to create the peaks and partially cancel to create the troughs when out of phase.

Figure 9 shows the relationship between residual vibration amplitude and initial luff angle for upward luffing. In general, larger initial luff angles lead to larger residual vibration, as shown by the 5° and 45° luffing distance lines. This occurs because for small initial luff angles, the boom is mainly in the horizontal plane. In this configuration, the starting motion is mostly vertical and does not contribute significantly to the radial motion of the payload. For large initial luff angles, the starting motion has a larger component in the radial direction and therefore results in larger radial payload oscillation. For some luffing distances, such as 27° and 54°, this relationship is reversed. These luffing distances correspond to the troughs of Figures 8. For these luffing distances, the starting and stopping forces are out of phase. At higher initial luff angles, the stopping force is able to cancel more of the starting force, leading to lower residual vibration.

For equal luffing distances, luffing upward and downward will produce approximately the same magnitude of residual vibration if the initial and final luff angles are exactly reversed. This is because the luffing angle determines the magnitude of the radial component of the acceleration and deceleration. For example, luffing upward from a luffing angle of 30° to 60° produces 6.4 cm of residual vibration. Luffing downward from a luffing angle of 60° to 30° produces 6.7 cm of residual vibration.

Iterative routines were carried out for the luffing motion to further investigate the relationship between residual vibration amplitude, move distance, initial luffing angle, and direction of luffing. The suspension length was set to 1 m and the luffing distance and initial luff angle were varied between 0° and 90°. Figure 10 shows the residual vibration amplitude from upward luffing for unshaped and ZV-shaped commands. The ZV shaper was again designed for a natural frequency of 3.13 rad/sec. Input shaping reduced residual vibration.
by an average of 97%. ZV-shaped commands were able to significantly reduce the residual vibration amplitude for luffing upward for all initial luff angles and luffing distances.

C. Slewing and Luffing

The boom crane dynamics become more complex when two rotations are performed simultaneously. Figure 11 shows the location of the payload during an upward luffing of 30°, from an initial angle of 45°, and a simultaneous 10° slewing. The suspension cable length was kept constant at 1 meter. The payload motion during the transient stage is a complicated function of the radial, tangential, and centripetal accelerations caused by the slewing and luffing commands. However, once the slewing and luffing commands are complete, the payload precesses, similar to the motion caused by slewing commands alone.

In order to analyze the effectiveness of input shaping on the combined luffing and slewing motions, both unshaped and ZV-shaped commands were used to drive the boom crane model for luffing distances between 0° and 55°, from an initial luff angle of 35°, and slewing distances between 0° and 90°. Figure 12 shows the residual vibration amplitude as a function of slewing and luffing distances. There are small variations in residual vibration amplitude with respect to changes in luffing distance. However, there are much larger variations in residual vibration amplitude with respect to slewing distance. As seen in the previous data, slewing commands produce larger residual vibration and are responsible for the majority of the residual vibration amplitude. Even given the complicated nonlinear dynamics of multiple-axis motion, ZV-shaping is still able to substantially reduce the residual vibration amplitude for all possible combinations of slewing and luffing commands. For the space shown in Figure 12, input shaping reduced residual vibration by an average of 93%.

The velocity and acceleration limits used in these simulations are relatively low. For higher accelerations, nonlinear effects would become more significant and ZV input shaping would not be as effective. However, the velocities and accelerations used here correspond reasonably well with those used to drive large boom cranes.

IV. Experimental Verification

The mobile boom crane shown in Figure 1 was used for experimental verification of the dynamics and control presented in this paper. The body of the crane is approximately 115 x 50 cm. The top slewing base can rotate with respect to the bottom mobile base and is capable of 340° slewing rotation. The slewing rotation is made possible with the use of a turntable bearing and 4 stud-mount, ball transfers placed between the two layers.

The 200 cm boom is attached to the slewing base. The payload is moved in and out with respect to the base by luffing the boom inward or outward. The luffing angle is controlled via a cable attached to a motor. The suspension cable length is controlled via a different motor. A Siemens digital camera is mounted at the tip of the boom and records the swing deflection of the payload. (The camera is not shown in picture.) All actuation of the crane is done by Siemens synchronous, AC servomotors. The motors are controlled via Siemens SINAMICS motor drives with PI feedback control on the desired velocity.

Experiments were performed to verify two important aspects of the simulation results: i) the effectiveness of input shaping at reducing the residual oscillation and, ii) the alternating peaks and troughs in the residual oscillation amplitude as a function of move distance. To verify the alternating peaks and troughs in the residual oscillation amplitude, the crane was luffed at increments of 2.5° starting from 20° until a final angle of 75°. At the initial state, the payload was 20 cm above the ground. The boom crane has a physical design that provides automatic level luffing as the boom is luffed. Level luffing keeps the payload the same height above the ground during luffing motions. The level luffing is optimized to work best in the range of 40° to 70°. Outside of that range, the payload height does not stay perfectly level. However, the suspension cable length does change to provide some degree of height compensation.

Figure 13 shows the residual oscillation amplitudes obtained from the experiments. This figure is similar to a slice through Figure 10 at the point where the initial luff angle is 20°. However, Figure 10 does not include the
level-luffing effect present in the experimental setup. The residual vibration amplitude increases and decreases as the luffing distance is increased, similar to the results obtained through simulation. Figure 13 also shows the shaped residual vibration amplitude for the same luffing distances. The ZV shapers were designed for the average natural frequency during the luffing motion. As the luffing distance increases, the ZV-shaper’s performance degrades. This occurs because for longer luffing distances, the variation in the vibration frequency is larger and the ZV shaper is not robust enough to completely compensate for these changes. Nonetheless, the ZV-shaped commands reduced the residual amplitude by an average of 76%.

V. CONCLUSIONS

Boom cranes exhibit complex nonlinear dynamic behavior because they have two rotational axes that cause the payload to experience centripetal forces. The complex dynamic responses were investigated for a wide range of slewing and luffing motions. As the move distances increase, the residual oscillation amplitude varies in a periodic manner. This variation in the residual vibration occurs because the vibration caused by the stopping deceleration can be either in phase or out of phase with the vibration induced by the initial acceleration. Even on such a nonlinear crane, Zero Vibration input shaping was successful in reducing the residual oscillation.

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