Simplified open-Loop anti-Sway Technique

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Abstract- **Antisway controllers present in today's market are built around complex technologies. Such technology, for implementation, demands presence of high-end computers, sway sensors, and servo performance drives in the crane electrics. Cost and computational demands for implementation of such controllers makes them unsuitable for use with rail mounted gantry cranes equipped with low-end programmable logic controllers (PLCs) and adjustable speed drives (ASDs). In this paper we had discussed about the theory for open-loop antisway technique. Presented technique is computationally less demanding and does not require fixture of any sway sensor. Discussed technique works well with gantry cranes deploying low-end PLCs with ASDs.**

I. Introduction

Almost every industry uses rail mounted gantry crane for its material handling applications. These rail mounted gantry cranes are mostly equipped with cabled hoisting mechanism, which are prone to the load sway problems. Sway persists even if a skilled operator is operating the crane. Because of danger to the ground staff and ground equipment, these load sways can't be accepted for such material handling applications. Thus, anti-swing control of material handling cranes is becoming a necessity day by day.

In past, numerous research publications had been made on the design of closed loop anti-swing controllers for such material handling cranes. In all such publications [1,2,3,4,5], swing control laws are built around complex technologies like dynamic crane modeling, numerical state space controls, neural networks, and fuzzy logic etc. Such control laws focuses on making real-time crane acceleration corrections based on the input from feedback sway sensors.

Because of money constraints, most industries cannot afford costly higher-end processors and sway sensors that are required for implementation of such technically complex antisway control laws. Conventionally, industrial cranes use low-end PLC with ASDs for crane positioning. These ASDs are not capable to perform on-the-fly acceleration corrections as desired by the present antisway techniques. Hence, a simpler antisway control technique is required which shall meet following objectives

- *Simpler Design:* Control technique must be built across simple mathematical laws, which can be easily implemented on existing low-cost, low-end PLCs using ASDs.
- *Minimal Sway:* Control technique must guarantee minimum sway generation during the starting, progress, and end of entire load travel.
- *Low Cost:* Simple open-loop control is desired, which shall work without costly sway sensors and as well as implementable on existing low-end crane PLCs.

Our work deals with pre-computation of numbers of timed drive reference inputs that shall be applied to the crane drive, in order to move the load to the target position without swing. Such pre-computations for numbers and timings of drive reference inputs were performed one-time, on a high-end computer for various traveling distances. Computation results were then programmed into a crane PLC as a lookup table. In order to move the load to the target position without swing, crane's controller is left with the simple task to choose the numbers and timings from this lookup table and apply it to the crane system. This method reduces computational burden on crane controller to zero, thereby leading to a simplified open-loop minimum sway design as desired in earlier.

Our pre-computation technique is an extension to generalized posicast control technique by [6], and [7]. and deals with elimination of oscillations and overshoot in a lightly damped pendulum systems. Our technique pre-computes drive reference on a high-end computer inline with generalized posicast control theory thereby leading to a simplified open-loop anti-swing control for rail mounted material handling cranes with cabled hoisting mechanisms.

II. Dynamic Behavior Of Suspended Load

Idea behind our technique is to accelerate, move and then decelerate the crane with pre-computed velocity patterns in order to minimize the swing generation. Before looking deeper into the technological implementation, let us first understand the dynamic behavior of the suspended hoist hook with the crane trolley put under accelerating as well as decelerating forces.

With an assumption that under heavy loads hoisting cable can be treated as a rigid body, dynamics of a rail mounted gantry crane can be considered similar to the dynamics of a forced damped pendulum with presence of external wind forces.

It is to be noted that under normal conditions and in a semi-enclosed environments external force on the pendulum like wind forces etc are negligible and may not considered. However we understand that these forces are significant for offshore cranes, where such control techniques cannot be deployed. Hence, for our open-loop system, by neglecting external forces we are only considering an ideal forced pendulum dynamics for the load swing.

Fig. 1 represents a 2-DOF (degree-of-freedom) model of a swinging load for a rail mounted gantry crane, when the crane is put under external accelerating force ($F=Ma$).

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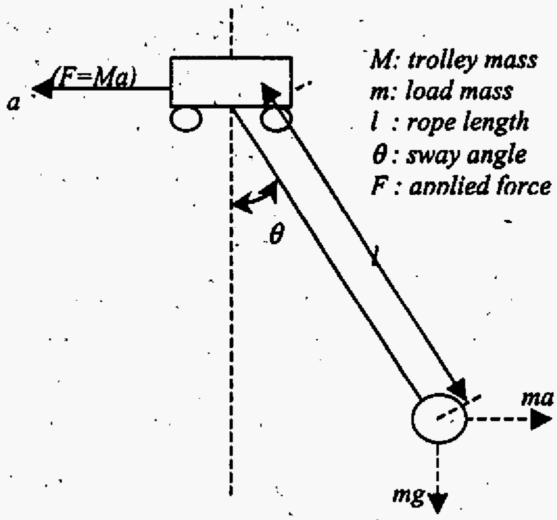

Figure 1. Pendulum With Force Applied On Fulcrum

Moment of inertia (I) of the pendulum shown above can be written as

$$I = ml^2 \tag{1}$$

Neglecting frictional forces and damping, net restoring torque on the pendulum equals

$$\tau_r = (mal\cos\theta - mgl\sin\theta). \tag{2}$$

Applying Newton's law of motion for rotational motion of the pendulum we can write

$$I\frac{d^2\theta}{dt^2} = \tau_r. \tag{3}$$

Using (1), (2), & (3) radial acceleration of the forced pendulum is calculated as

$$\frac{d^2\theta}{dt^2} = \left\{\frac{a\cos\theta - g\sin\theta}{l}\right\}. \tag{4}$$

Considering small sway angle, it can be further assumed that $\cos\theta \approx 1, \textit{and} \sin\theta \approx 0$. Thus above equation further reduces to following second order linear differential equation

$$\frac{d^2\theta}{dt^2} = \frac{(a - g\theta)}{l}. \tag{5}$$

Laplace transform of the above equation provides following transfer function among the applied acceleration and produced sway angle.

$$G(s) = \frac{\theta(s)}{a(s)} = \frac{1}{l(s^2 + \omega^2)}, \text{ where } \omega = \sqrt{g/l}. \tag{6}$$

Equation (6) represents the transfer function of load position with crane trolley put under acceleration. In order to study the behavior of such system under application of external accelerating forces, let us apply a constant acceleration of magnitude 'a' to the system starting from time 't_0' and ending at time 't_1'. (*NOTE: For behavior study under external decelerating forces, reverse the sign of the acceleration pulse to '-a'.*)

Any constant magnitude pulse from time t_0 to time t_1 can be represented in time domain as follows

$$a(t) = \pm a \cdot \{U(t - t_0) - U(t - t_1)\}$$

Where sign of 'a' represents magnitude of the acceleration or deceleration being imparted. Considering 'a' to be +ive for accelerations only, Laplace transform of above input pulse can be written as

$$a(s) = a\left(\frac{e^{-st_0} - e^{-st_1}}{s}\right). \tag{7}$$

Sway angle produced with the application of above input pulse to our pendulum system can be represented by time-domain response of the transfer function and it is given by (6) as

$$\theta(t) = \Im^{-1}\{G(s) \cdot a(s)\}.$$

Where $\Im^{-1}$ represents the inverse-laplace transform. Substituting from (6) & (7) in the above equation and solving for time-domain response, we get output sway angle as

$$\theta(t) = \frac{a}{g}\{U(t - t_0) - \cos\omega(t - t_0) - U(t - t_1) + \cos\omega(t - t_1)\}.$$

After the pulse is over (time $t > t_1$) above solution further reduces to

$$\theta_1(t) = \frac{a}{g}\{\cos\omega(t - t_1) - \cos\omega(t - t_0)\}. \tag{8}$$

Equation (8) represents, the resultant sway angle upon application of accelerating force of magnitude 'a' from time t_0 to time t_1 to the pendulum.

Similarly upon application of same magnitude accelerating force from time t_2 to time t_3, the resultant sway angle can be written as

$$\theta_1(t) = \frac{a}{g}\{\cos\omega(t - t_3) - \cos\omega(t - t_2)\}. \tag{9}$$

System represented by (6) is linear in nature and principle of superposition is applicable. Thus, output of this system upon application of 2 numbers of accelerating force pulses of magnitude 'a' from time t_0 to t_1 and further from time t_2 to t_3 can be written as $\theta(t) = \theta_1(t) + \theta_2(t)$, or simply

$$\theta(t) = \frac{a}{g}\{\cos\omega(t-t_1) - \cos\omega(t-t_0) + \cos\omega(t-t_3) - \cos\omega(t-t_2)\}. \tag{10}$$

Equation (10) represents the load swing behavior with crane put under 2 pulses of accelerating force inputs of same magnitude. If we choose accelerating instances as $t_2=t_0+T/2$ and $t_3=t_1+T/2$, (10) reduces to solution $\theta(t)=0$. From this solution, one can conclude that, "If two numbers of T/2 delayed accelerating pulses of constant magnitude 'a' are applied to any linear pendulum its final swing will be zero." This statement is applicable for decelerating forces also since decelerations can be represented as negative accelerating forces. Such a pair of accelerating or decelerating force input is called a two-pulse input.

Since, the system under consideration is linear in nature and the principle of superposition holds true, we can further concluded that, "Upon application of n-numbers of above such two-pulse inputs the final sway of the pendulum will remain zero."

MATLAB is used to verify the previous statement and to simulate the linear pendulum behavior under application of various combinations of two-pulse input accelerations and decelerations of varied magnitudes. Graphs for trolley acceleration (Fig. 2), and the resultant sway angle (Fig. 3) were plotted against time. From Fig. 3, we see that the final swing of the linear pendulum remains zero under application of any such two-pulse input combinations.

To study suspended load dynamics for our semi-enclosed practical crane, we ignored the external wind forces and pendulum dampening. MATLAB simulation similar to the one mentioned above is repeated for a non-linear pendulum to prove that the, "resultant swing remains zero for a lightly damped non-linear pendulum too, upon application of any combinations of above mentioned two-pulse inputs."

III. Implementation

In order to pick-n-place loads across plant locations, one must follow following trolley movement cycle one after another.

- *STEP-A:* While at the starting position raise the load to the safe traveling height.
- *STEP-B:* Once the load is lifted to the safe height, accelerate the crane trolley to the desired traveling speed (u_{tr}) in the desired load movement direction.
- *STEP-C:* Continue trolley movement close to the target position at fixed traveling speed (acceleration, a=0)
- *STEP-D:* When the target position arrives, apply decelerate the trolley to stop the crane at target position.
- *STEP-E:* Subsequently lower the load at the target position.

Moving load as per the cycle mentioned above, no swing is generated at Step-C because from (10), one can conclude that if a=0, resultant swing $\theta(t)$ is zero. Also, at Steps -A and -E no swing is generated because of load being raised and lowered. From (10), swing is only generated at Steps–B and –D because of applied non-zero acceleration and deceleration. If we apply accelerations and decelerations as two-pulse inputs only, zero load swing is produced at Steps-B and –D both.

With the previous statement our problem of load movement without swing reduces to the problem of determination of number of accelerating or decelerating two-pulse inputs that must be applied to speed up the crane to the desired travel speed or to stop the crane respectively.

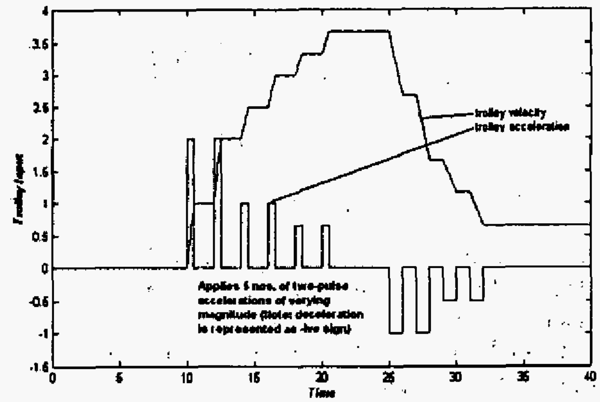

Figure 2. Trolley Acceleration Vs. Time

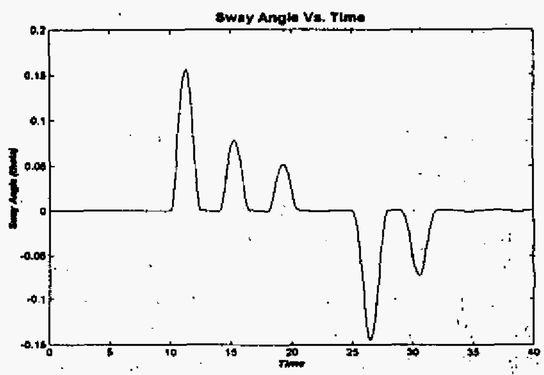

Figure 3. Resultant Sway Angle Vs. Time

In order determine number of two-pulse inputs; let us draw Fig. 4, which represents velocity vs. time curve for a crane with application of five simultaneous two-pulse input accelerations of magnitude 'a', and pulse width 't_p'.

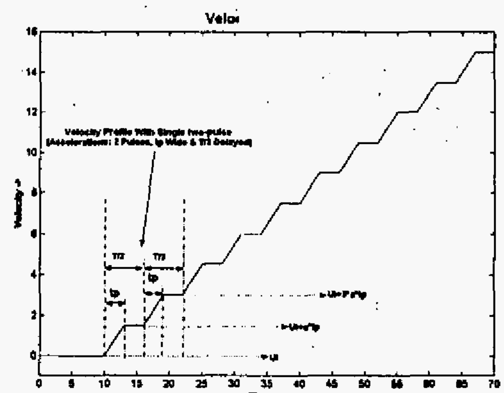

Figure 4. Velocity Time Curve With Application of 5-Numbers of Two Pulse Accelerating Inputs of Constant Magnitude

Looking at Fig. 4, general formula for change in velocity and distance traveled upon application of n-simultaneous two-pulse accelerating inputs of width 't_a' can be derived as

$$\delta v_n = (v_n - u_i) = 2nat_a \text{, and} \tag{11}$$

$$x_n = nu_iT + \left\{\frac{(2n^2+n)at_aT - 2nat_a^2}{2}\right\}. \tag{12}$$

In (11)– (12), u_i is the initial system velocity, which is zero while moving the load from rest and v_n is the final velocity. Similarly, generalized formulae for change in velocity and distance traveled by the crane under application of m-simultaneous two-pulse decelerating inputs of constant magnitude 'd' and pulse width 'td' can be derived as

$$\overline{\delta v_m} = (v_m - u_i) = -2mdt_d \text{, and} \tag{13}$$

$$\overline{x_m} = mu_iT - \left\{\frac{(2m^2+m)dt_dT - 2mdt_d^2}{2}\right\}. \tag{14}$$

In (13)– (14), u_i is the initial traveling velocity and v_m is the final velocity, which is zero for our case. A simple antisway load positioning logic can be built using (11), and (13), which shall produce computed two-pulse accelerations or decelerations output to the crane drive based on the differential speed requirements for load positioning.

In our fully automated system parameters like desired traveling velocity (u_{tr}), trolley acceleration (a), trolley deceleration (d), programmed trolley acceleration time (t_a) and deceleration time (t_d) is known constants for the various known-traveling distances. With these parameters, (11)-(14) can be used to compute number of accelerating (n) and decelerating (m) pulses required to move the crane for each such known-traveling distances. One-time computations of n, and m can be done on a high-end computer and programmed on a PLC as lookup table for every such known-traveling distance.

Now in order to move the load using Steps A to E without swing, PLC is left with the simplified task to select the proper values of n and m from the above mentioned lookup table.

Proposed antisway technique is being implemented at G Blast Furnace, Slag-Pit crane automation project at TATA STEEL.

IV. Conclusion

Antisway technique thus presented above is computationally simple, cost effective and open loop in nature as was the objective. However, for this technique presence of following types of swing in the system was not considered.

- Residual swing as an effect of excessive pendulum damping
- Induced swing because of external factors like wind forces, collision jerks etc
- Initial swing created due to jerky lifting of load off the ground

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