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An optimal performance control scheme for a 3D crane

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ABSTRACT

This paper presents an optimal performance control scheme for control of a three dimensional (3D) crane system including a Zero Vibration shaper which considers two control objectives concurrently. The control objectives are fast and accurate positioning of a trolley and minimum sway of a payload. A complete mathematical model of a lab-scaled 3D crane is simulated in Simulink. With a specific cost function the proposed controller is designed to cater both control objectives similar to a skilled operator. Simulation and experimental studies on a 3D crane show that the proposed controller has better performance as compared to a sequentially tuned PID-PID anti swing controller. The controller provides better position response with satisfactory payload sway in both rail and trolley responses. Experiments with different payloads and cable lengths show that the proposed controller is robust to changes in payload with satisfactory responses.

1. Introduction

Cranes are important and used in many industries, factories and ware houses to transfer heavy loads from one place to another [1]. One of the significant factors affecting productivity and efficiency of the industrial systems is speed. However it is obvious for a crane, fast manoeuvres resulted in significant payload oscillation and residual sway that negatively affects performance of the system. At higher speeds, these sway angles prevent the payload to settle down during movement and unloading. This problem will be crucial particularly for industrial applications where operators should manipulate the cranes [2]. Vast applications of cranes have encouraged many researchers around the globe to reduce the motion-induced oscillation of these structures [3]. One of the most useful approaches to reduce the motion-induced oscillations of oscillatory systems is input shaping [4].

Initially, the input shaping theory was introduced in the late 1950s [5] and it was named as "Posicast Control". The well-known input shaping technique was firstly introduced by Singer and Seering [6]. Researchers have provided and utilised different types of input shapers including Zero Vibration (ZV) and Zero Vibration Derivative (ZVD) [7], negative shapers [8], multi-hump extra intensive shapers [9], two mode shapers [10] and unity magnitude shapers [11]. The first time input shaping implemented on a real gantry crane was at Savanah Research Technology Centre [12] and researchers proved their approach was successful to move the cart without sway during hoisting. Recently, implementing methods of input shaping on cranes have been improved by researchers [13].

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Despite the advent of many control theories and techniques, PID feedback controller is still one of the most widely used control algorithm in industries [14]. They are easy to implement, low cost and effective and thus they have a significant role in industrial applications. Many type of PID based control schemes to control the swing motion and cart position of the cranes have been presented by researchers including neural network based PID controller [15], PSO based PID controllers [16], genetic algorithm based PID controllers [17] and fuzzy based PID controllers [18–20]. Combination of input shaping and PID controller has been previously proposed [21]. Some researchers also investigated this combination carefully [22] and a closed-loop application of this combination has been also proposed [23]. Moreover, a comprehensive analysis on closed-loop applications of input shaping has been done [24]. Recently, PID and S-shaped input are utilised to reduce the sway of a rotary crane [25]. Some researchers discussed optimal closed loop PID–PD control of gantry crane without input shaping [26]. However, an optimal performance closed-loop control scheme including both PID and input shaping has not been proposed.

In this paper, a control scheme is presented for an automatic crane based on optimal design of PID and ZV shaper in a closed loop system. A specific cost function to solve two control objectives concurrently is proposed. The objective is to design a practical anti-swing controller without utilising the sway reduction loop of PID–PID control scheme. In the other words, the PID controller gains of a closed-loop system are optimised based on a specific cost function which considers actual position and sway of the payload. Thus a simple architecture with only one loop can satisfy the control requirements in real time experiment. Initially, a complete mathematical model of the system is obtained based on Newtonian techniques. The closed loop system including the ZV shaper, saturation block and PID controller is designed in Simulink. PID controller's gains are then optimised to satisfy a specific cost function based on the absolute value of the difference between desired position of payload and its actual position.

A PID–PID anti-swing controller [18] augmented with ZV shaper is considered for comparison. The result shows that the proposed optimal performance controller performs better than the PID–PID anti-swing controller in several aspects. Although, the proposed controller needs an accurate model for designing purpose and cannot handle environmental disturbances, its simplicity and good performance may be useful. Based on this scheme, a crane can be moved to a predefined destination with a reduced motion-induced sway of a payload.

2. Three dimensional (3D) crane system

Fig. 1 shows a lab-scaled 3D crane used in this study. The crane is capable of transferring a load from any location to a desired place in a restricted three dimensional space. The system hardware consists of three main components: a cart, a rail and a pendulum. The mathematical model is obtained based on the given characteristics of the crane by the manufacturer and the study by Pauluk et al. [27]. The obtained model is simulated using Simulink to investigate dynamic behaviour of the system.

A schematic diagram of the 3D crane system is shown in Fig. 2 with *XYZ* as the coordinate system. m_p, m_t and m_r are the payload mass, trolley mass (including gear box, encoders and DC motor) and moving rail respectively. l represents the length of the lift-line, α represents the angle of lift-line with Y axis and β represents angle between negative part of Z axis and projection of the payload cable onto the XZ plane. T is a reaction force in the payload cable acting on the trolley, F_x and F_y are the forces driving the rail and trolley respectively, F_z is a force lifting the payload and f_x, f_y and f_z are corresponding friction forces. By defining

$$\mu_1 = \frac{m_p}{m_t},\ \mu_2 = \frac{m_p}{m_t + m_r};\quad u_1 = \frac{F_x}{m_t},\ u_2 = \frac{F_y}{m_t + m_r},\ u_3 = \frac{F_z}{m_p};\quad f_1 = \frac{f_x}{m_t},\ f_2 = \frac{f_y}{m_t + m_r}, f_3 = \frac{f_z}{m_p}$$
$$K_1 = u_1 - f_1,\ K_2 = u_2 - f_2,\ K_3 = u_3 - f_3$$

Fig. 1. A lab-scaled 3D crane.