Bridge-Crane Hoist Mass Angular Oscillation

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Hung Ly
Faculty of Engineering and Technology
Nguyen Tat Thanh University
Ho Chi Minh City, Vietnam
lthung@ntt.edu.vn

Than Chau*
Faculty of Engineering and Technology
Nguyen Tat Thanh University
Ho Chi Minh City, Vietnam
ctthanh@ntt.edu.vn

Huong Nguyen
Faculty of Engineering and Technology
Nguyen Tat Thanh University
Ho Chi Minh City, Vietnam
ntlhuong@ntt.edu.vn

***Abstract*—The oscillation of hoisting masses in bridge cranes significantly impacts system stability, resulting in adverse effects such as excessive dynamic loads, inaccurate positioning during loading and unloading, and overall instability. Therefore, the angular oscillation of the hoisting mass must be controlled in the design of bridge cranes. This paper presents a mathematical model for the analysis of the angular oscillation of a hoisting load, both during oscillation and trolley movement. The MATLAB/Simulink software was used to simulate the dynamic behavior of the model to gain insights into the oscillation angle and its limitations. The results substantiate the reliability of the proposed model for predicting the angular oscillation of hoisting loads in bridge cranes.**

Keywords— bridge crane, hoisting mass, oscillating angle.

I. INTRODUCTION

Cranes represent a significant means of mechanization in the loading, unloading, and transportation of goods, thereby enhancing labor productivity and reducing production costs. This equipment is undergoing rapid development to meet the increasing demands of customers in terms of quality, transportation productivity, and the ability to lift and transport goods along complex trajectories with accuracy and safety [1]. Cranes are becoming more prevalent in the fields of transportation and construction. The growing demand for faster turnaround times and the necessity of meeting safety requirements have led to an increased demand for more efficient and effective controllers. Container cranes are a common feature of port and port operations infrastructure [2].

A crane is a lifting apparatus that performs repetitive movements, including acceleration, lifting, traveling, and deceleration. Additionally, two combined movements are observed, namely lifting and traveling, as well as lowering and traveling [3]. In the context of heavy-duty crane operations involving substantial loads, the impact of vibrations generated at the outset of lifting, loading, and unloading becomes a significant consideration, necessitating a comprehensive assessment of the crane's structural integrity. Based on the aforementioned factors and our experience, it can be demonstrated that the steel structure of the crane is subjected to robust and intricate vibrations during operation. During the process of moving the trolley and lifting the load, the crane structures will undergo simultaneous changes in mass, damping system, and stiffness. Accordingly, the intricate dynamic responses must be meticulously taken into account when designing the actual crane structure [4].

The crane system comprises three principal components: the lifting mechanism, the moving mechanism, and the reach change mechanism. The classification is based on the number of degrees of freedom of the system, the structural characteristics of the system, and the unity of the assembly of the lifting load, the hook, and the lifting cable in terms of the dynamics of the complex system. The suspension point of the lifting object functions as a pendulum, oscillating in the plane or space. It is therefore of great importance to model, simulate, and calculate the kinematics, dynamics, and control of a crane system to increase labor productivity and ensure high rigidity, safety, and reliability. This is an urgent task for researchers in recent years. This illustrates the necessity for the comprehension and creation of anti-sway solutions for lifting objects [5-6]. Halder [3] also developed a controller based on the feedback signal of the sway angle, intending to suppress the sway motion of the lifting object when the crane is in motion and transporting the object to the desired position.

The function of a crane is to facilitate the transportation of a load from one point to another in the most efficient manner possible, ensuring that the load reaches its destination without any undue deviation from its intended path. In the majority of cases, the task is carried out by a highly proficient operator. During operation, the load is permitted to oscillate in a pendulum-like motion. Should the swing angle exceed a critical threshold, it is imperative that the operator either minimize the swing or halt the operation until the swing is eliminated. This results in a waste of time spent waiting, or a reduction in the availability of the equipment. In response to these challenges, numerous researchers have sought to develop control algorithms to automate crane operations. However, the majority of existing schemes lack the requisite practical implementation capabilities. Consequently, the majority of industrial cranes remain unautomated, continuing to rely on human operators who are occasionally unable to restrict the movement of the load. Another challenge in the automation of cranes is the inherently unstructured nature of their operational environment, which is particularly prevalent in shipyards and factory floors. As the load being lifted fluctuates due to the acceleration of the motion, many researchers have concentrated their efforts on the generation of trajectories that provide the load in the shortest possible time while simultaneously minimizing the sway. The aforementioned trajectories are obtained through the application of optimization techniques. The objective function may be defined in terms of the transfer time, the control action, or the rotation angle [6]. Another significant approach to trajectory generation is input shaping, which involves a series of acceleration and deceleration pulses. These sequences are generated with the objective of eliminating residual motion at the conclusion of the transfer operation. The resulting controller is open-loop, which renders it susceptible to external disturbances and parameter variations [7].

The distinctive feature of crane machines is the integration of a mobile trolley (carriage) that conveys the lifting object. This carriage executes two distinct vertical movements: firstly, the carriage moves in a horizontal direction, and secondly, the lifting object undergoes a rocking motion in the vertical plane, with an amplitude dependent on the characteristics of the vehicle's movement and the lifting mode employed. The

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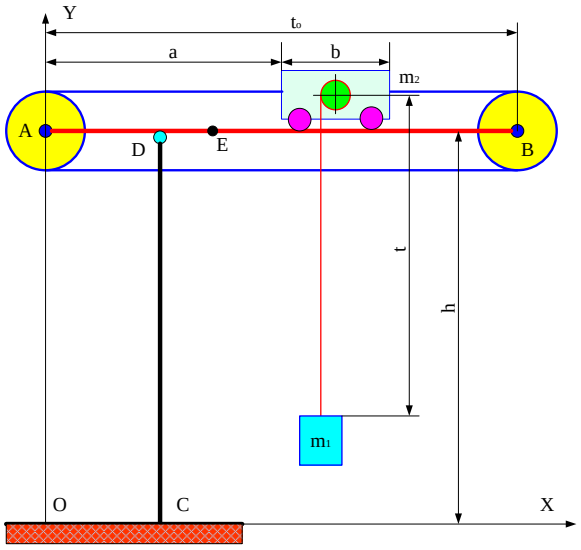
uncontrolled translational movements of the lifting trolley present a significant hazard to the user and numerous other related considerations. Furthermore, these movements have a detrimental impact on the installation of equipment and the loading and unloading of goods.

The oscillatory motion of the lifting object in the crane, which is often caused by the working mechanisms, can result in a number of undesirable effects. These include the application of large dynamic loads to both the machine and the working mechanisms, the difficulty in controlling the object to the correct position, an extension of the working cycle of the machine, and a reduction in the machine's productivity. It is therefore evident that dampening the oscillations of the lifting object during the operation of the machine is a meaningful and beneficial practice in practice, as it serves to stabilize the crane system and ensure safety during operation. The angular velocity and frequency of rotation are contingent upon the length of the pendulum (lift object) and the mass of the payload, or the combined mass of the payload and the lifting-moving transport. The pendulum length demonstrates that the predominant frequencies in the rotation angle responses exhibit a decline with an increase in pendulum length. The amplitude of the rotation angle is observed to increase with an increase in the payload mass. It has been observed that minimal changes are evident in the rotation angle frequency in response to corresponding changes in the payload mass. The swing angle of the lifting load can be significantly reduced if a suitable control strategy is employed. Nayfeh [2] further developed the recently proposed hysteretic position feedback controller for container cranes. The crane was modeled as a simple pendulum comprising a sling and a lifting load block and constructed as a four-bar mechanism. To gain a deeper comprehension of the control and dynamics issues about cranes, it is imperative to undertake a comprehensive review of the existing literature [10].

This paper presents a mathematical model of the swaying process of a lifting load when a trolley is in motion and lifting an object. The model is based on the principle shown in Fig. 1 and uses the MATLAB/Simulink tool to solve a system of motion equations with anti-sway measures for the lifting load. Subsequently, the paper is structured as follows: 2. Modeling the sway angle; 3. Typical case study; 4. Results and discussion; 5. Conclusion and proposed research direction.



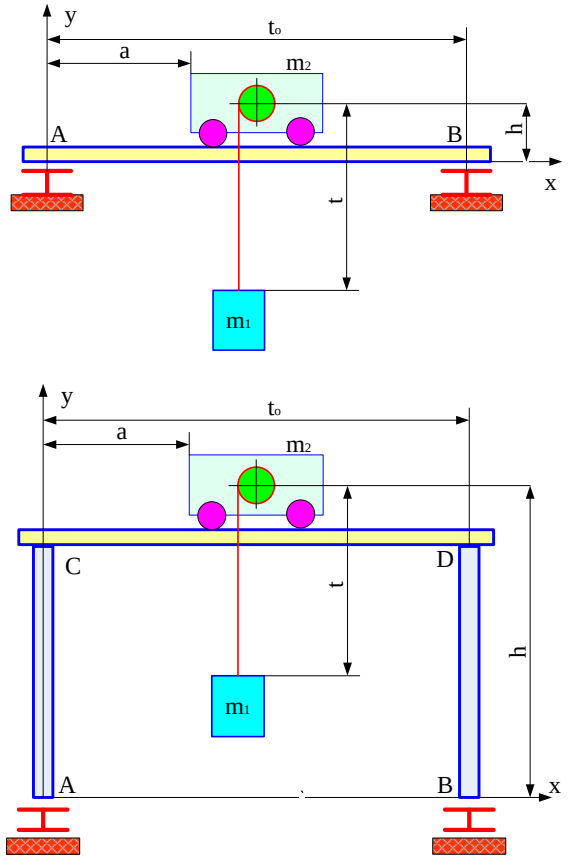


Fig. 1. The diagram illustrates the operational principles of various types of cranes.

II. SWAY ANGLE MODELING

A. Principle and assumptions

The fundamental configuration of the bridge crane is illustrated in Fig. 2. The bridge crane is equipped with two beams that are capable of perpendicular movement along the beam due to the presence of wheels. The trolley, which is responsible for transporting the lifting object through the cable system, functions in a manner analogous to a simple pendulum. It is capable of moving along the central beam, which in turn can traverse both of the primary beams. The load is elevated by the cable drum, which is driven by an electric motor. The control panel is designed to regulate the lifting and lowering processes, the movement of the trolley, and the beams of the bridge crane.

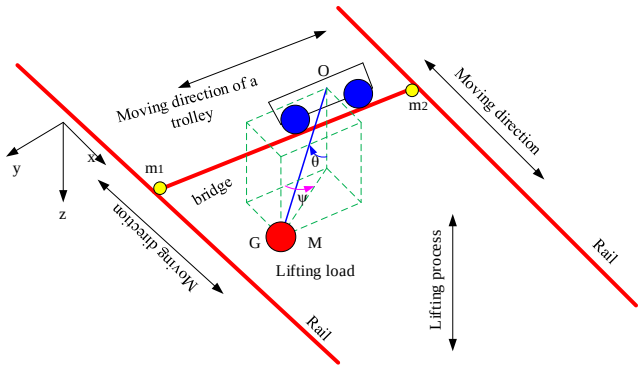


Fig. 2. The fundamental model depicts the movement and lifting of objects in three-dimensional space by a trolley.

The following assumption is employed to streamline the mathematical model of the sway angle while maintaining the crane principle system's overall generality. This assumption

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posits that the cable conveying the lifting object (load) is both inextensible and massless in both static and dynamic states. It can be assumed that the plastic deformation of the crane steel frame (crane) is negligible, as all elements have infinite stiffness (absolute stiffness). The effects of anti-sway and loss due to the driving mechanism are considered to be negligible. The influence of wind force should be disregarded. The applied load, which is considered to be concentrated at a single point, is lifted and lowered by the massless cable. The change in cable length resulting from the swaying of the lifting load is deemed to be insignificant.

B. The swing angle of the lifting load

To relocate the load to the desired position, the crane requires a thrust force. The sole means of generating this force is dependent on the swing angle (θ). Consequently, the phenomenon of oscillation of the lifting object is to be expected during the process of lifting and transporting the load. It is, however, essential to leverage the advantages of the swing angle to create the control force in accordance with the operator's specifications, rather than attempting to suppress the oscillation. The model of a mobile lifting apparatus is illustrated in Fig. 3.

The following symbols are used in the model:

P: Impact force

f: Coefficient of frictional resistance between the moving vehicle and the rail

T: Tension force of the suspension wire

k: Coefficient of air resistance

v: Linear velocity of the lifting load (m/s)

M: Mass of the moving trolley

ℓ: Length of the lifting wire

θ: Swing angle of m

x: Coordinate of the moving trolley.

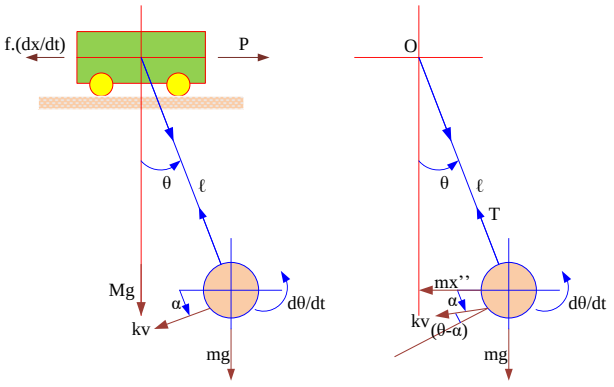

Fig. 3. Model of a trolley moving and lifting objects under wind load.

The model depicts the movement and lifting of objects by a trolley under the influence of wind load. By employing the principles of mechanics, we have derived the following system of equations that describe the sway angle for the crane system:

$$\begin{cases} (M+m)\ddot{x} = P - f\dot{x} - kv\cdot\cos\alpha - ml\ddot{\theta}\cos\theta + ml\dot{\theta}^2\sin\theta \\ ml\ddot{\theta} = -mg\sin\theta - kv\cdot\cos(\theta-\alpha) - m\ddot{x}\cos\theta \end{cases} \quad (1)$$

III. NUMERICAL EXAMPLE

The crane system is surveyed with the following parameters: The mass of the moving trolley is 1 ton (1000 kg), the lifting capacity is m = 100 kg, the length of the cable carrying the lifting object is ℓ = 5 m, the coefficient of friction between the moving trolley and the rail (beam) is $f = 0.3(M + m)$, the acceleration due to gravity is taken as g = 9.81 m/s², the coefficient of air resistance is k = 0.05, and the motor pulling the moving trolley has a rotation speed of 1500 rpm. The sway angle model is solved by implementing the block diagram in the MATLAB/Simulink model, as illustrated in Fig. 4, and the PID control model, as depicted in Fig. 5.

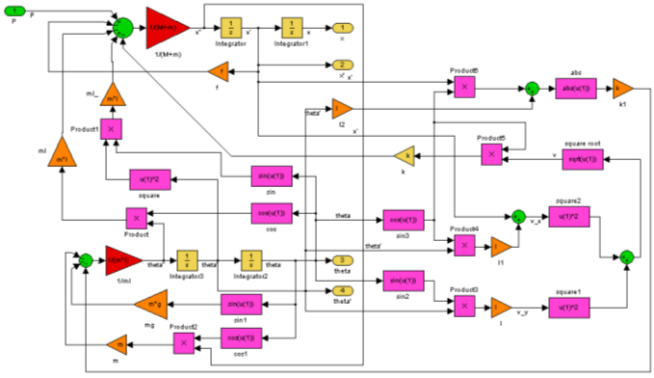

Fig. 4. MATLAB/Simulink model of the swing angle

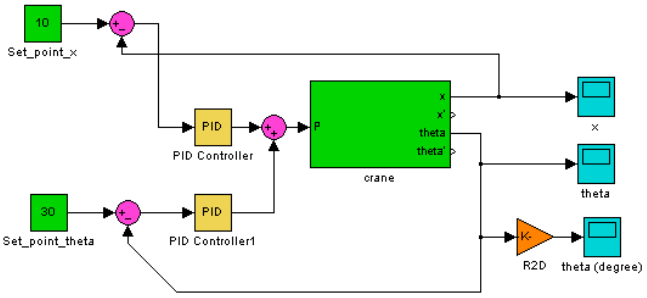

Fig.5. Simulink model using PID

It can be observed that the sway angle of the lifting load is minimal for a 5m-traveling trolley, falling within the range of less than 3 degrees. This demonstrates the efficacy of control in addressing the anti-sway problem and highlights its practical significance. If the trolley is required to traverse a distance of approximately 20 meters, the calculations indicate that the sway angle will be in the range of 15 to 20 degrees (Fig. 6).

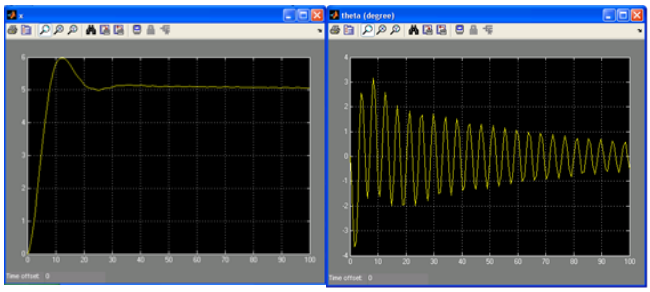

Fig. 6. a. Coordinate position of x b. The swing angle of lifting load

In the MATLAB/Simulink model, varying the length of the rope (denoted as ℓ) reveals that an increase in ℓ, or a longer hanging rope, leads to an elevated oscillation of the lifting object, as indicated by an augmented oscillation angle θ. This results in an increase in instability, which in turn presents challenges in the loading and unloading of goods. In this case, the oscillation angle (θ) is controlled to prevent oscillation, and the length of the rope has a minimal impact on the process of shaking the object. The oscillation angle of the lifting object

is constrained within an allowable range (less than 20 degrees) (see Fig. 7).

+ With the length value ℓ = 10 m (Fig. 7):

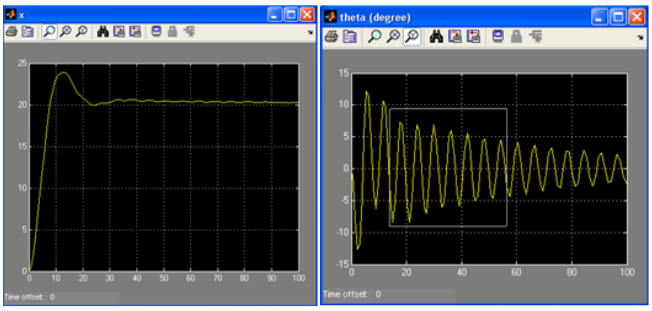

Fig. 7. a. Coordinate position of x (ℓ = 10 m) b. The swing angle of the lifting load (ℓ = 10 m)

IV. RESULTS AND DISCUSSION

The lifting process begins with the cable in a slack position, allowing approximately 2-3 seconds for the lifting drum to wind the cable. Once the cable has been extended, the lifting object exerts a force on the cable that causes it to stretch following the laws of elasticity. This stretching of the cable results in the lifting object making contact with the ground. The lifting object does not ascend rapidly; rather, it moves horizontally at a minimal distance due to the friction between the lifting object and the ground. This is clearly illustrated in the aforementioned graphs, which depict the instability in the lifting process between 0 and 5 seconds. Once the frictional force has been overcome, the lifting object is elevated from the ground and begins to move under the influence of gravity. As the lifting drum winds the lifting object, the length of the cable is reduced, thereby diminishing the amplitude of the object's oscillatory motion. The amplitude of this motion decreases after each cycle. This finding is entirely consistent with the graph illustrating the correlation between the length of the suspended wire and the amplitude of the oscillatory motion. The speed of lifting and the acceleration of the trolley exert a considerable influence on the swing angle of the load being lifted. Accordingly, an increase in lifting speed and a constant trolley speed contribute to limiting the swing angle of the lifted load. This paper paves the way for future research in the following areas: The following research areas have been identified as requiring further investigation: (1) Determination of the dynamic load of the lifting mechanism; (2) Determination of the dynamic load for the entire system; (3) Construction of a model and simulation of soft dynamics, taking into account the elastic deformation of the mechanism during operation; (4) Determination of the law of precise position control for loading and unloading goods; (5) Investigation of stability and anti-sway issues for lifting; (6) Determination of fatigue strength; (7) Reliability of the steel structure system Additionally, future research will consider determining optimal parameters for the system, the relationship between dynamic quantities, controlling the correct trajectory, optimizing the system in terms of time, space, and the dynamic problem of the steel structure system of the crane with the lifting process. Furthermore, optimal conditions for system operation will be determined.

V. CONCLUSION

Cranes are devices that offer significant practical value in the transportation and logistics industry at seaports, particularly in the loading and unloading of containers. The sway angle of the lifting load is of great significance concerning the stability of the crane during operation. This paper has constructed a model of the sway angle of the lifting load and sought to identify a solution for the sway angle based on the block diagram of MATLAB/Simulink, intending to determine reasonable sway angle amplitudes for the model. The sway angle is typically constrained to a value below the allowable threshold of 20 degrees. In the future, the paper will develop a fuzzy control system for the sway angle of port cranes engaged in the loading and unloading of containers.

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