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MODELLING AND CONTROL OF OFFSHORE CRANE SYSTEMS

by

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Certificate of Authorship/Originality

I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as a part of the requirements for other degree except as fully acknowledged within the text.

I also certify that this thesis has been written by me. Any help that I have received in my research work and in the preparation of the thesis itself has been fully acknowledged. In addition, I certify that all information sources and literature used are quoted in the thesis.

R.M.T. Raja Ismail

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ABSTRACT

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Cranes are widely used in transportation, construction and manufacturing. Suspended payloads in crane system are caused to swing due to actuator movement, external disturbance such as wind flows, and motion of the crane base in the case of portable cranes. Recently, offshore cranes have become a new trend in stevedoring and in offshore construction as they can help to avoid port congestion and also to exploit ocean engineering applications. For crane operations, it is important to satisfy rigorous requirements in terms of safety, accuracy and efficiency. One of the main challenges in crane operations has been identified as the sway motion control, which is subject to underactuation of crane drive systems and external disturbances. Particularly in offshore cranes, the harsh conditions can produce exogenous disturbances during the load transfer at various scenarios of offshore crane operations in practice. Therefore, it is interesting as to how to design robust controllers to guarantee high performance in the face of disturbances and parameter variations in offshore cranes.

The motivation for this thesis is based on recent growing research interest in the derivation of dynamic models and development of control techniques for offshore cranes in the presence of, for example, the rope length variation, sway, ocean waves and strong winds in offshore crane systems. Accordingly, the work for this thesis has been conducted in the two main themes, namely analytical modelling and control design, for which new results represent its contributions.

Dynamic models of two types of offshore crane systems, namely the offshore gantry crane and offshore boom crane, are derived in the presence of vessel's ocean wave-induced motion. The effect of wind disturbances on the payload sway is also considered in the modelling. In the control context, sliding mode control techniques for a generic form of underactuated mechanical Lagrangian systems are presented, including the conventional first-order, second-order and adaptive fuzzy sliding mode controllers. The major component in this part of the thesis is the design of sliding mode control laws based on the developed offshore crane models for trajectory tracking problems, in the presence of persistent disturbances in severe open-sea conditions. Extensive simulation results are presented to demonstrate the efficacy of the models and robustness of the designed controllers.

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List of Publications

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5. **R.M.T. Raja Ismail** and Q.P. Ha (2013), *Trajectory tracking control for offshore boom cranes using higherorder sliding modes*, The 30st International Symposium on Automation and Robotics in Construction and Mining, Montreal, Canada, pp. 894-904.

6. **R.M.T. Raja Ismail** and Q.P. Ha (2012), *Second-order sliding mode control for offshore container cranes*, The 22nd Australasian Conference on Robotics and Automation, Wellington, New Zealand, 7p.

7. **R.M.T. Raja Ismail**, Nguyen D. That, and Q.P. Ha (2012), *Observer-based trajectory tracking for a class of underactuated Lagrangian systems using higher-order sliding modes*, The 8th IEEE International Conference on Automation Science and Engineering, Seoul, Korea, pp. 1200-1205.

8. Nguyen D. That, Nguyen K. Quang, **R.M.T. Raja Ismail**, P.T. Nam and Q.P. Ha (2012), *Improved reachable set bounding for linear systems with discrete and distributed delays*, The 1st International Conference on Control, Automation and Information Sciences, Ho Chi Minh, Vietnam, pp. 137-141.

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Nomenclature and Notation

Throughout the thesis, the following nomenclatures and notations are used:

- 1-SMC: First-order sliding mode control
- 2-SMC: Second-order sliding mode control
- 2-D: Two-dimensional
- 3-D: Three-dimensional
- AFSMC: Adaptive fuzzy sliding mode control
- DOF: Degree of freedom
- HOSM: Higher-order sliding modes
- LQR: Linear quadratic regulator
- LMI: Linear matrix inequality
- LTI: Linear time-invariant
- MIMO: Multi input multi output
- SISO Single input single output
- SMC: Sliding mode control
- SVD: Singular value decomposition
- UMS: Underactuated mechanical system
- VSC: Variable structure control
- $\mathbb{R}$: Field of real numbers
- $\mathbb{R}^n$: n-dimensional space
- $\mathbb{R}^{n\times m}$: Space of all matrices of $(n \times m)$-dimension
- A^T: Transpose of matrix A
- A^{-1}: Inverse of matrix A
- I_n: Identity matrix of $(n \times n)$-dimension
- $0_{n\times m}$: Zero matrix of $(n \times m)$-dimension
- C_θ: $\cos\theta$

- S_θ: $\sin\theta$
- $\lambda(A)$: Set of all eigenvalues of matrix A
- $\lambda_{\min}(A)$: Smallest eigenvalue of matrix A
- $\lambda_{\max}(A)$: Largest eigenvalue of matrix A
- $\mathrm{diag}(\lambda_1, \cdots, \lambda_i, \cdots, \lambda_n)$: Diagonal matrix with diagonal entries λ_i, $i = 1, \cdots, n$
- $\mathrm{rank}(A)$: Rank of matrix A
- $\mathrm{sign}(\cdot)$: Signum function
- $||\cdot||$: Euclidean norm of a vector or spectral norm of a matrix
- $\forall$: For all

Chapter 1

Introduction

1.1 Background

Cranes are used for transportation of heavy loads such as containers and construction materials on land as well as at open sea. They are widely used in construction sites, warehouses and harbours (Figure 1.1) due to their ability to handle hefty objects.

In general, cranes can be regarded as underactuated Lagrangian system. Firstly, crane systems are underactuated because they have fewer independent control actuators than degrees of freedom (DOF) to be controlled. For example, in a 2-D gantry crane system, both cart position on the girder and payload sway are controlled by a single motor. Secondly, cranes are classified as Lagrangian systems because their equations of motion can be obtained based on the formulation of Lagrangian mechanics like robotic manipulators. Basically, a crane system consists of a support mechanism, which is a part of its structure, and a hoisting mechanism. The hoisting mechanism of a crane often exhibits an oscillatory behaviour due to the underactuation of the system. For this reason, it is important for a crane operation to meet stringent safety requirement.

A large number of studies on the development of control strategies to improve the efficiency and safety of crane operations has been seen over the past few decades. The hoisting mechanism that typically consists of cable, hook and payload assembly has high compliance. Hence, certain exogenous excitations at the suspension point can produce high amplitude of oscillations to the payload. The inertia forces due to

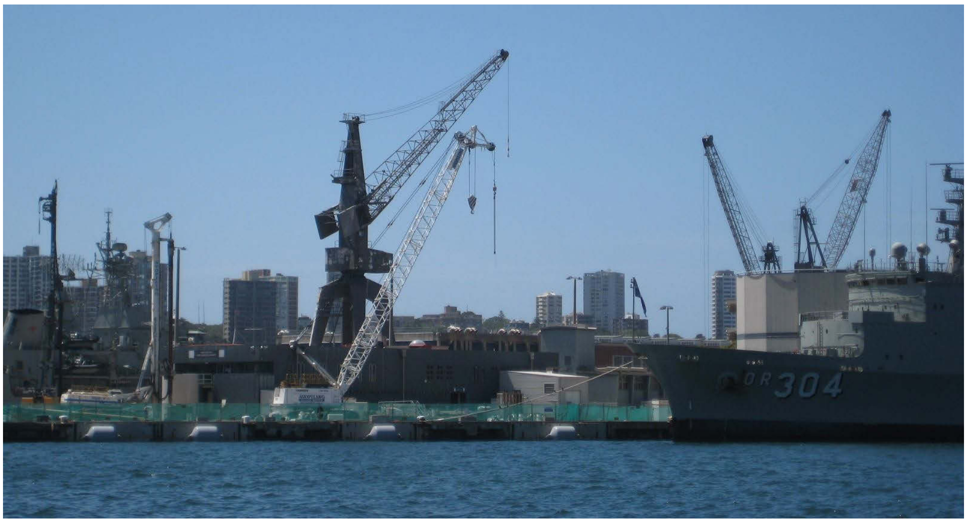

Figure 1.1 : Panoramic view of Sydney Harbour.

the motion of the crane can induce significant payload pendulations as well. This problem occurs because cranes are typically lightly damped. In other words, any transient oscillation response in crane systems takes a long time to dampen out.

As the research on conventional cranes becomes well established, researchers have explored a more complex problem, namely the offshore cranes. The growing usage of ocean facilities in many segments such as shipping of containers, oil and gas exploration, and offshore wind farm construction have necessitated certain operations occur in open sea conditions. These activities require the application of offshore cranes to transfer loads between vessels or to place loads from a vessel to an offshore site.

In general, offshore cranes operations can be categorised into two types, namely the stevedoring and the moonpool operations. Offshore stevedoring or lightering is the process of transferring containers between vessels, and moonpool operation is the activity of a payload placement underwater or on the seabed for the purpose of underwater installation. The advantage of offshore stevedoring operation is it can avoid marine traffic congestion in a port. The transfer of containers between two vessels requires a crane equipment on one of the vessel (Figure 1.2) or a third vessel (Figure 1.3). The common types of offshore cranes used in this operation is gantry crane (Figure 1.4) and boom crane (Figure 1.5).

Port congestion has become a major issue over the last few years due to rapid developments of logistics industry causing a substantial increase in the trading volume [50, 74, 144]. Some ambitious plans of port expansion have been proposed to overcome this problem, but it is not a feasible solution due to land constraints. Consequently, a new method of transportation, namely, the ship-to-ship cargo transfer operation, is introduced [81]. This method, emerging to become a promising solution to improve ports' efficiency and productivity and reduce operational costs, could enable the ports to stay competitive.

Despite all the necessities and benefits of offshore transportation and installation, the presences of persistent disturbances in the crane operations due to harsh sea condition are inevitable. Ocean waves can induce motions to vessels or ships where cranes are located. These motions include roll, pitch, yaw, surge, sway and heave. Besides, wind drag or buoyancy of seawater can produce exogenous forces on the payload, whenever it is suspended in the air or submerged. For this reason, it is necessary to have an element of robustness in the offshore cranes control system to deal with the aforementioned disturbances.

In particular, motivated by a large amount of significant practical problems, the control of underactuated nonlinear systems has become an important subject of research. Intuitively, the control synthesis for underactuated systems is more complex than that for fully actuated systems. Control of underactuated systems is currently an active field of research due to their broad applications in robotics, land and aerospace vehicles, surface vessels and crane automation. Based on recent surveys, control of general underactuated systems is a major open problem. Since the

Figure 1.2 : Ships lightering operation [126].

Figure 1.3 : An offshore crane transferring containers between a ship and a vessel [135].

presence of uncertainties and parameter variations always aroused in underactuated nonlinear systems, the implementation of robust control approach on the systems is appealing.

Sliding mode control (SMC) is a well-known control methodology belonging to the variable structure systems which are characterised by their robustness with re-

Figure 1.4 : Container gantry crane mounted on a vessel [113].

Figure 1.5 : Ship-mounted boom cranes near Port Botany, Sydney.

spect to parameter variations and external disturbances. The basic idea of the sliding mode is to drive the system trajectories into a predetermined hyperplane or surface and maintain the trajectory on it for all subsequent time. During the ideal sliding motion, the system is completely insensitive to uncertainties or external disturbances. The dynamics and performance of the systems then depend on the selection of the sliding surface. In SMC design, a sliding surface is first constructed to meet existence conditions of the sliding mode. Then, a discontinuous control law is synthesized to drive the system state to the sliding surface in a finite time and maintain it thereafter on that surface. However, the effects of the discontinuous nature of the control, known as the chattering phenomenon have originated a certain scepticism about such an approach. The common practice to overcome the chattering phenomena is by changing the system dynamics in a small vicinity of the discontinuity surface. This modification can avoid real discontinuity while preserving the main properties of the whole system. However, robustness of the sliding mode were partially lost.

The introduction of higher-order sliding modes (HOSM) can practically attenuate the chattering if properly designed. The chattering attenuation can be achieved because the HOSM acting on the higher order derivatives of the system deviation from the constraint (e.g., sliding function), instead of influencing the first deviation derivative that occurs in standard or first-order sliding modes. HOSM preserve the main advantages of the original approach, as well as totally remove the chattering effect. Besides, HOSM can provide higher accuracy in realization of the control system. Second-order sliding mode control (2-SMC) algorithms recently developed have produced satisfactory results for single-input systems. The extension of second-order sliding mode to multi-input systems, as in general, most of the underactuated systems are, is nontrivial.

Fuzzy logic control has been an active research topic in automation and control. The basic concept of fuzzy logic control is to utilize the qualitative knowledge of a system for designing a practical controller. Generally, fuzzy logic control is applicable to plants that are ill-modelled, but qualitative knowledge of an experienced operator is available. The principle of SMC has been introduced in designing fuzzy logic controllers. This combination which is known as adaptive fuzzy sliding mode control (AFSMC) provides the mechanism to design robust controllers for nonlinear systems with uncertainty. Adaptive fuzzy has been either used to adjust the control gain of the sliding mode or approximate the system dynamics to construct the sliding function or the control law. The development of AFSMC for uncertain mechanical systems to tackle more generic problems have been an active research topic in recent years.

The motivation for this thesis is based on recent growing research interest in the derivation of dynamic models and development of control techniques for offshore cranes subject to exogenous disturbances and parameter variations. This research has been conducted in the analytical modelling and control design for offshore crane systems, for which new results represent its contributions.

1.2 Research objectives

The main objectives of this research are:

i. To develop dynamic models of 2-D and 3-D offshore cranes in the presence of system disturbances due to open-sea condition.

ii. To formulate the generalisation of sliding mode control for a class of nonlinear underactuated mechanical systems with bounded uncertainties.

iii. To construct the robust first-order and second-order sliding mode controllers for offshore crane systems subject to system disturbances.

1.3 Thesis organization

This thesis is organised as follows:

- *Chapter 2*: This chapter presents a survey of the underactuated mechanical systems, cranes dynamics and control, the recent development of offshore crane control systems, and sliding mode control approaches.

- *Chapter 3*: The dynamic models of offshore crane systems subject to disturbances and uncertainties based on the Euler-Lagrange formulation are derived in this chapter.

- *Chapter 4*: This chapter presents the sliding mode control designs for a generic form of underactuated mechanical systems. The proposed controllers have been implemented to conventional crane systems.

- *Chapter 5*: In this chapter, the problem of robust sliding mode control is investigated for trajectory tracking problem of offshore crane systems with bounded disturbances.

- *Chapter 6*: In this chapter, a second-order sliding mode control law is proposed for offshore gantry crane and boom crane, making use of its capability of chattering alleviation while achieving high tracking performance and preserving strong robustness.

- *Chapter 7*: A brief summary of the thesis contents and its contributions are given in the final chapter. Recommendation for future works is given as well.

Chapter 2

Literature Survey

In this chapter, a brief survey of underactuated mechanical systems, cranes dynamics and control, and sliding mode control including the general first- and the second-order control laws, is presented.

2.1 Underactuated mechanical system

Underactuated mechanical systems (UMS) are systems that have fewer independent control actuators than degrees of freedom (DOF) to be controlled. UMS arise in a broad range of real-life applications, and this class of systems have been the subject of active scientific research. In general, the control of UMS is a more challenging task as compared to the control of fully actuated systems because the former presents additional restrictions on the control design. Besides, it gives rise to complex theoretical problems that may not found in fully actuated systems, and that may not be solved using classical control techniques. The control of UMS has been studied for a long period in the control literature and has been attracting more attention in recent years due to the growing interests in new theories and applications. This section provides brief survey of the most recent studies on UMS from control point of view and focuses on its application to marine vehicles and crane systems. A more detail survey of crane control strategies is provided in Section 2.2.2.

Research on UMS can trace back to twenty years ago when control of nonholonomic mechanical systems were of great interest to researchers, e.g., [9, 21], and references therein. Studies on this class of systems have gained more attention years after, and they have been widely used in robotics, aircrafts and marine vehicles. Various control strategies for UMS have been proposed in the literature, including intelligent control, backstepping, sliding mode, and many more. The most recent review paper on UMS has been reported by Liu and Yu [90], in which a comprehensive survey of UMS is presented from its history to the state-of-the-art research on modelling, classification, and control.

Numerous studies have attempted to give a classification and a generalisation of these systems with the aim of proposing a systematic control design method for UMS. Several researchers have formulated the stabilisation problems of UMS by using controlled Lagrangian methods [19, 20, 26], passivity-based control [106], equivalent-input-disturbance approach [123], and Lyapunov-based method [117]. Sliding mode control (SMC) is one of the most popular methods in the control designs for a generic form of UMS. These include studies on reachability [102], stabilisation [100], and sliding surface design techniques [91]. The works on generic SMC control design have been reported for two DOF UMS [94, 95] and also for UMS without any restriction on the number of DOF [8, 121, 140, 142]. Other control methods proposed for a generic form of UMS are hybrid control [55], adaptive control [27], and passivity-based control [31].

Marine vehicles and cranes are among UMS, which attracted many research interests as the topic of control problems. The challenge in the underactuated ship and surface vessel control systems is to solve the trajectory tracking problem in the presence of ocean waves disturbance. Among the control strategies that have been proposed for underactuated ships and surface vessels are state feedback control [82], backstepping method [38, 58, 59, 111], adaptive control [37, 39], Lyapunov's method [36, 42, 72], SMC [7, 49, 146], and cooperative control [41, 57]. Studies on underactuated gantry cranes, which have similar equations of motion with cart-pole systems, are also have been reported by many researchers. Most recent works on underactuated crane motion control can be referred to [52, 130, 131, 147].

2.1.1 Equations of motion

The dynamics of UMS is formulated based on Lagrangian mechanics. In general, the equations of motion of UMS can be written in the form of Euler-Lagrange equation as follows [90, 91]:

$$\frac{d}{dt}\left(\frac{\partial \mathcal{L}}{\partial \dot{q}}\right) - \frac{\partial \mathcal{L}}{\partial q} = E(q)u, \tag{2.1}$$

where $\mathcal{L} = \mathcal{K}(q,\dot{q}) - \mathcal{P}(q)$ is the Lagrangian, $\mathcal{K}(q,\dot{q})$ is the total kinetic energy, $\mathcal{P}(q)$ is the total potential energy, $q \in \mathbb{R}^n$ is the vector of generalised coordinates, $u \in \mathbb{R}^m$ is the vector of actuator input, and $E(q) \in \mathbb{R}^{n\times m}$ is the matrix of external forces, with $1 \leq m < n$. The kinetic energy $\mathcal{K}(q,\dot{q})$ is a quadratic function of the vector $\dot{q}$ of the form

$$\mathcal{K}(q,\dot{q}) = \frac{1}{2}\sum_{i,j}^{n} m_{ij}(q)\dot{q}_i\dot{q}_j = \frac{1}{2}\dot{q}^T M(q)\dot{q},$$

where $M(q) \in \mathbb{R}^{n\times n}$ is the inertia matrix and $m_{ij}(q)$ is an element of the matrix. The Euler-Lagrange equation (2.1) can be written as [128]:

$$\sum_j m_{kj}(q)\ddot{q}_j + \sum_{i,j} c_{ijk}(q)\dot{q}_i\dot{q}_j + \frac{\partial \mathcal{P}(q)}{\partial q_k} = p_k^T E(q)u, \quad k = 1, \cdots, n, \tag{2.2}$$

where p_k is the kth standard basis in $\mathbb{R}^n$ and $c_{ijk}(q)$ are known as Christoffel symbols, defined as

$$c_{ijk}(q) = \frac{1}{2}\left(\frac{\partial m_{kj}(q)}{\partial q_i} + \frac{\partial m_{ki}(q)}{\partial q_j} - \frac{\partial m_{ij}(q)}{\partial q_k}\right).$$

Finally, by defining $G_k(q) = \partial\mathcal{P}(q)/\partial q_k$ or

$$G(q) = \frac{\partial \mathcal{P}(q)}{\partial q},$$

(2.2) can be expressed in matrix form as

$$M(q)\ddot{q} + C(q,\dot{q})\dot{q} + G(q) = E(q)u, \tag{2.3}$$

where $C(q,\dot{q}) \in \mathbb{R}^{n\times n}$ is the centrifugal-Coriolis matrix, $c_{kj} = \sum_{i=1}^{n} c_{ijk}(q)\dot{q}_i$ is an element of $C(q,\dot{q})$, and $G(q) \in \mathbb{R}^n$ is the vector of gravity. The dynamics (2.3) has some important properties that facilitate control analysis and design. Among these properties, the following are often used in control development [90]:

i. The inertia matrix $M(q)$ is positive definite, symmetric, and bounded such as for $M(q) \in \mathbb{R}^{n\times n}$,

$$k_1 I_n \leq M(q) = M^T(q) \leq k_2 I_n,$$

where $k_1, k_2 > 0$.

ii. A skew symmetric relationship exists between the inertia matrix $M(q)$ and the centrifugal-Coriolis matrix $C(q,\dot{q})$ such as for $M(q), C(q,\dot{q}) \in \mathbb{R}^{n\times n}$

$$\nu^T \left(\dot{M}(q) - 2C(q,\dot{q})\right)\nu = 0, \quad \forall \nu \in \mathbb{R}^n.$$

iii. Define the total energy of the system as

$$\mathcal{W}(q,\dot{q}) = \mathcal{K}(q,\dot{q}) + \mathcal{P}(q) = \frac{1}{2}\dot{q}^T M(q)\dot{q} + \mathcal{P}(q).$$

Then the time derivative of the total energy is

$$\dot{\mathcal{W}}(q,\dot{q}) = \frac{1}{2}\ddot{q}^T M(q)\dot{q} + \frac{1}{2}\dot{q}^T M(q)\ddot{q} + \frac{1}{2}\dot{q}^T \dot{M}(q)\dot{q} + \dot{q}^T \frac{\partial \mathcal{P}(q)}{\partial q}$$

Since $\ddot{q}^T M(q)\dot{q} = \dot{q}^T M^T(q)\ddot{q} = \dot{q}^T M(q)\ddot{q}$ (Property i) and $\dot{q}^T \dot{M}(q)\dot{q} = 2\dot{q}^T C(q,\dot{q})\dot{q}$ (Property ii), the above equation becomes

$$\begin{aligned}\dot{\mathcal{W}}(q,\dot{q}) =& \dot{q}^T M(q)\ddot{q} + \dot{q}^T C(q,\dot{q})\dot{q} + \dot{q}^T G(q)\\ =& \dot{q}^T \left(M(q)\ddot{q} + C(q,\dot{q})\dot{q} + G(q)\right)\\ =& \dot{q}^T E(q)u,\end{aligned}$$

which implies that the system is passive with respect to the input u and output $\dot{q}$. The passivity is an important character of UMS which shows that the system has a stable origin.

If the matrix $E(q)$ is assumed to be $E(q) = [I_m \; 0^T_{(n-m)\times m}]^T$, the vector of generalised coordinates can be partitioned as $q = [q_a^T, q_u^T]^T$. By letting $f(q,\dot{q}) = C(q,\dot{q})\dot{q} + G(q)$ and partitioning the vector as $f(q,\dot{q}) = [f_a^T(q,\dot{q}), f_u^T(q,\dot{q})]^T$, (2.3) can be expressed in the following form [8, 100, 102, 142]:

$$\begin{bmatrix} M_{aa}(q) & M_{au}(q) \\ M_{au}^T(q) & M_{uu}(q) \end{bmatrix} \begin{bmatrix} \ddot{q}_a \\ \ddot{q}_u \end{bmatrix} + \begin{bmatrix} f_a(q,\dot{q}) \\ f_u(q,\dot{q}) \end{bmatrix} = \begin{bmatrix} I_m \\ 0_{(n-m)\times m} \end{bmatrix} u, \tag{2.4}$$

where $q_a \in \mathbb{R}^m$, $q_u \in \mathbb{R}^{n-m}$, $M_{aa}(q) \in \mathbb{R}^{m\times m}$, $M_{au}(q) \in \mathbb{R}^{m\times(n-m)}$, $M_{uu}(q) \in \mathbb{R}^{(n-m)\times(n-m)}$, $f_a(q,\dot{q}) \in \mathbb{R}^m$, and $f_u(q,\dot{q}) \in \mathbb{R}^{n-m}$.

2.1.2 Feedback linearisation

The feedback linearisation approach generalised the concept of inverse dynamics of Lagrangian systems. The basic concept of feedback linearisation is to construct a nonlinear control law as an inner-loop control (see Figure 2.1). In ideal case, the inner-loop control exactly linearises the nonlinear system after a proper state space change of coordinates. One can then design a second stage or outer-loop control in the new coordinates to satisfy the control design specifications.

Consider the dynamics equation in the form of (2.4). The idea of inverse dynamics is to seek a nonlinear feedback control law

$$u = U(q,\dot{q})$$

which, when substituted into (2.4), results in a linear closed-loop system. If we choose the control u according to the equation

$$u = \begin{bmatrix} I_m & -M_{au}(q)M_{uu}^{-1}(q) \end{bmatrix} \begin{bmatrix} f_a(q,\dot{q}) \\ f_u(q,\dot{q}) \end{bmatrix} + \Big(M_{aa}(q) - M_{au}(q)M_{uu}^{-1}(q)M_{au}^T(q)\Big)v, \tag{2.5}$$

then, since the matrix $M(q)$ as well as its partitions $M_{aa}(q)$ and $M_{uu}(q)$ are invert-

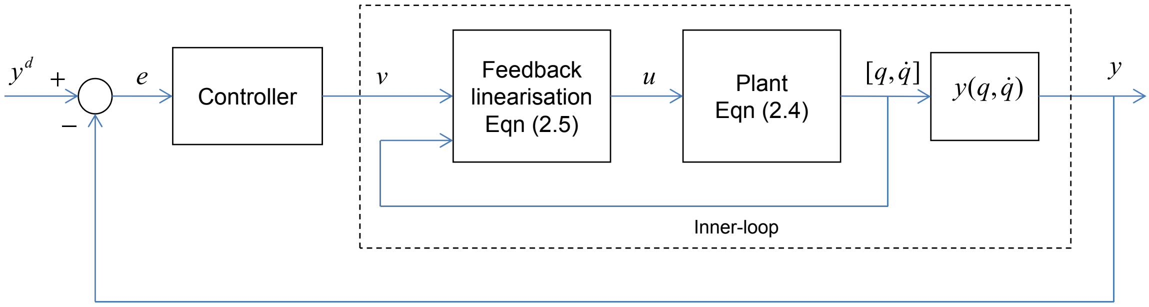

Figure 2.1 : Architecture of feedback linearisation control.

ible, the combined system (2.4) and (2.5) becomes

$$\begin{bmatrix} M_{aa}(q) & M_{au}(q) \\ M_{au}^T(q) & M_{uu}(q) \end{bmatrix} \begin{bmatrix} \ddot{q}_a \\ \ddot{q}_u \end{bmatrix} = \begin{bmatrix} I_m & -M_{au}(q)M_{uu}^{-1}(q) \\ 0_{(n-m)\times m} & 0_{(n-m)\times(n-m)} \end{bmatrix} \begin{bmatrix} f_a(q,\dot{q}) \\ f_u(q,\dot{q}) \end{bmatrix}$$
$$- \begin{bmatrix} f_a(q,\dot{q}) \\ f_u(q,\dot{q}) \end{bmatrix} + \begin{bmatrix} M_{aa}(q) - M_{au}(q)M_{uu}^{-1}(q)M_{au}^T(q) \\ 0_{(n-m)\times m} \end{bmatrix} v$$
$$= - \begin{bmatrix} M_{au}(q)M_{uu}^{-1}(q) \\ I_{n-m} \end{bmatrix} f_u(q,\dot{q})$$
$$+ \begin{bmatrix} M_{aa}(q) - M_{au}(q)M_{uu}^{-1}(q)M_{au}^T(q) \\ 0_{(n-m)\times m} \end{bmatrix} v.$$

Eventually, one can show that the last equation can be reduced to

$$\begin{bmatrix} \ddot{q}_a \\ \ddot{q}_u \end{bmatrix} = \begin{bmatrix} 0_{m\times 1} \\ -M_{uu}^{-1}(q)f_u(q,\dot{q}) \end{bmatrix} + \begin{bmatrix} I_m \\ -M_{uu}^{-1}(q)M_{au}^T(q) \end{bmatrix} v. \tag{2.6}$$

The term v represents a new input to the system which is yet to be chosen. However, practical implementation of the inverse dynamics control law (2.5) requires both that the parameters in the dynamic model of the system be known precisely and also that the complete equations of motion be computable in real time [128].

2.2 Crane dynamics and control

Cranes can be categorised based on the DOF the support mechanism offers the payload suspension point, e.g., gantry cranes (rectiliner translations in a horizontal plane); tower cranes (translation and rotation in a horizontal plane); and boom cranes (rotations around two orthogonal axes). From the control literature, cranes can be classified as UMS. The payload swing angles are considered as unactuated coordinates in cranes dynamics, e.g., longitudinal and lateral sways in a gantry crane system, and tangential and radial sways in a boom crane system. Abdel-Rahman et al. [1] have conducted a comprehensive literature review of crane modelling and control starting in 1961. In the following, we provide a survey of the most recent works on crane dynamics and control.

2.2.1 Crane dynamics

The most common crane modelling approaches are the lumped-mass and distributed-mass approaches. In the distributed-mass approach, the hoisting cable is modelled as a distributed-mass system and the hook and payload, lumped as a point mass, are applied as a boundary condition for this distributed-mass system [32, 114]. The lumped-mass approach is the most widely used method to crane modelling. The hoisting line is modelled as a massless rigid cable. The payload is lumped with the hook and modelled as a point mass.

The complexities of dealing with a nonlinear model of the crane systems drive many works on crane control to make-do with linearized approximations of the model. This simplification, however, may reduce controller robustness, in which linear controller may provide acceptable performance only within a small fixed operating range around the equilibrium point of the pendulation angles. As a result, there has been an increasing interest in the design of crane control strategies based on nonlinear crane models.

The general form of crane dynamics are presented in the extended form of underactuated mechanical systems (2.3) as follows:

$$M(q)\ddot{q} + C(q,\dot{q})\dot{q} + G(q) + d(t) = E(q)u, \tag{2.7}$$

where $d(t)$ can be regarded as the system uncertainty which may contains frictions and external disturbances. Conventional and offshore container crane dynamics are often presented in the form of (2.7), e.g., see [5, 53, 103]. Besides, in the case of offshore cranes, the matrices $M(q)$, $C(q,\dot{q})$ and $G(q)$ may contain the disturbance terms from the vessel's motion.

The offshore crane system in moonpool operation has a different form of dynamics as compared to (2.7), in which, the system is modelled as a fully-actuated system. In this case, the crane dynamics are modelled based on the vertical motion of the payload. Besides, most of the studies on offshore cranes in moonpool operation neglected the sway of the payload as well as the vessel's rolling motion [76, 97, 98, 120, 125]. However, the complexity of the model lies in the system disturbance, namely the hydrodynamic forces due to the effects of buoyancy and added mass. The crane dynamics in this working environment is represented in the following form [98, 125]:

$$m\ddot{z} + mg = f(t,z,\dot{z}) + u, \tag{2.8}$$

where $z \in \mathbb{R}$ is the payload position, m is the payload mass, $u \in \mathbb{R}$ is the cable force, and $f(t,z,\dot{z})$ is the hydrodynamic forces.

2.2.2 Crane control

In the following, we discuss the survey on crane control strategies. Due to some major differences between conventional and offshore cranes, each type is discussed separately.

Conventional crane control

A significant research effort has been devoted to the development of control strategies to improve the efficiency and safety of crane operations. From the crane control literature, the control techniques can be categorised into two types, namely, the open-loop and closed-loop control. In the conventional crane systems, the open-loop control methods for sway suppression are input-shaping [18, 56] and filtering techniques [68]. However, these methods are limited by the fact that they are sensitive to external disturbances. On the other hand, various closed-loop control methods have been proposed for trajectory tracking and sway suppression of conventional cranes. These methods include linear control [104], adaptive control [52, 143], fuzzy logic control [25], optimal control [132], delayed-feedback technique [67], and nonlinear control. The nonlinear control methods for conventional cranes can further be classified as Lyapunov's direct method [51, 131], first-order sliding mode control [5], and second-order sliding mode control [16]. In addition, many studies on crane trajectory planning have been reported, for example, [130, 147].

Offshore crane control

Over the past two decades, research on cranes' control and automation has focused on addressing challenges in their offshore operations. The synthesis of feedback control for offshore cranes remains a challenge because the systems involve the presence of parameter variations, e.g., changes of load during the process of loading/unloading, and the presence of disturbances, e.g., wave- and wind-induced motion. Besides, the presence of obstacles in the environment, such as harbour and vessel, must be taken into consideration for the path planning of load transfer.

Because of the facts as mentioned earlier, some researchers have proposed the modifications of offshore cranes' mechanical parts to change the properties of the systems. The new mechanical properties can avoid resonance in the system during

Table 2.1 : Summary of previous offshore crane models and control methods.

Author	**Modelling approach**	**Control method**	**Limitation**
Osinski & Wojciech (1998) [107]	– Boom crane – Euler-Lagrange formulation – Flexible boom	– Input shaping	– Only consider vessel's heave
Driscoll et al. (2000) [43]	– Underwater conveying – Finite-element lumped-mass model	– Passive heave compensation – Impedance control	– Only consider vessel's heave
Kimiaghalam et al. (2002) [75]	– Boom crane – Stevedoring operation – Use Maryland rigging – 2 DOF crane, 1 DOF vessel	– Feedforward control	– 2-D model – Only consider vessel's roll angle
Sagatun (2002) [120]	– Moonpool operation – Vertical motion – Consider hydrodynamics force	– Passive heave compensation – Impedance control	– Only consider vessel's heave

Author	Modelling approach	Control method	Limitation
Johansen et al. (2003) [73]	– Moonpool operation – Vertical motion – Consider hydrodynamics force	– Active heave compensation – Feedforward control	– Only consider vessel's heave
Masoud et al. (2004) [96]	– Boom crane – Stevedoring operation – 4 DOF crane, 3 DOF vessel	– Delayed-feedback control	– Fixed rope length – Vessel's motion is not included in the dynamics model
Skaare & Egeland (2006) [125]	– Moonpool operation – Vertical motion – Consider hydrodynamics force	– Active heave compensation – Parallel force/position control	– Only consider vessel's heave
Al-Sweiti & Söffker (2007) [4]	– Boom crane – Stevedoring operation – Use Maryland rigging	– Variable gain state-feedback control	– 2-D model – Only consider vessel's roll angle

Author	**Modelling approach**	**Control method**	**Limitation**
Hatleskog & Dunningan (2007) [66]	– Moonpool operation – Vertical motion	– Passive heave compensation – Impedance control	– 2-D model – Only consider vessel's heave
Parker et al. (2007) [110]	– Boom crane – Stevedoring operation – Use rider block tagline – Inverse kinematic	– Utilise the rider block tagline (no specific control method)	– 2-D model – Lack of analytic modeling in system dynamics
Do & Pan (2008) [40]	– Underwater conveying – Planar motion – Use electro-hydraulic system	– Active heave compensation – Lyapunov's direct method	– 2-D model – Only consider vessel's heave
Messineo et al. (2008) [97]	– Moonpool operation – Vertical motion	– Feedback compensator	– 2-D model – Only consider vessel's heave

Author	Modelling approach	Control method	Limitation
Schaub (2008) [122]	– Boom crane – Stevedoring operation – Velocity-based inverse kinematic – 3 DOF crane, 3 DOF vessel	– Rate-based control	– Ignore payload sway/oscillation
Messineo & Serrani (2009) [98]	– Moonpool operation – Vertical motion – Consider hydrodynamics force	– Heave compensation – Adaptive control	– Only consider vessel's heave
Küchler et al. (2011) [76]	– Underwater conveying – 2 DOF crane, 1 DOF vessel	– Prediction algorithm – Input/output linearisation	– Assume fully actuated system

Author	Modelling approach	Control method	Limitation
Ngo & Hong (2012) [103]	– Gantry (container) crane – Stevedoring operation – Euler-Lagrange formulation – 3 DOF crane, 3 DOF vessel	– Sliding mode control	– Assume fixed rope length
Fang et al. (2014) [53]	– Boom crane – Stevedoring operation – Euler-Lagrange formulation – 3 DOF crane, 2 DOF vessel	– Lyapunov method	– 2-D model

load transfer. For example, the rider block tagline system has been utilised to change the natural frequency of offshore crane's pendulation [110]. The Maryland rigging system was also proposed to change the properties of offshore crane systems, which can dissipate the payload sway with some additional control strategies, such as feedforward control [75] and state feedback control [4]. However, the introduction of the additional mechanism to crane systems leads to a higher complexity in the analysis of the crane dynamics.

Recently, the works on offshore crane systems have utilised their available control inputs rather than introducing additional mechanism. A common approach to deal with the complexity and underactuation of offshore crane dynamics is to break up the system into several parts and then analyse only a part or a decoupled part of the system states. Some researchers have first separated payload motion in the vertical direction and then designed a controller to vary the cable length. For example, the application of input shaping to minimise the cable deformation during payload vertical motion has been reported in [107]. Later, researchers have introduced the heave compensation approach to assign the payload to move at a constant vertical velocity in an earth-fixed reference frame in order to reduce the variations of the cable tension. This approach can further be categorised into passive and active heave compensations. Passive heave compensations can be constructed using augmented impedance control laws by utilising the tension of the cable as the only control input [43, 66, 120]. Some active compensation approaches have been constructed by using various type of control strategies, for example, feedforward scheme [73], feedback scheme [97], parallel force/position control [125], adaptive control [98], and Lyapunov's direct method [40]. Besides, active heave compensation approach combined with prediction algorithm for vessel motion has been reported in [76].

On the other hand, research on offshore crane dynamics and control has been devoted to considering a higher number of DOF in the systems' model to achieve satisfactory control performance. In [96], a delayed-feedback controller has been proposed to place the payload by using a linearised offshore crane system. A rate-based control strategy by using the measurements from onboard sensors has been reported in [122]. The most recent work on the dynamics analysis and nonlinear control has been reported in [53]. In this work, a Lyapunov's based controller has been designed for a simplified two-dimensional offshore crane model.

In terms of control strategies, sliding mode control (SMC) has been recognised as a strong control methodology for Lagrangian systems. Most recent works on SMC for offshore cranes have been reported in [103], [115] and [116]. However, the selections of SMC parameters to deal with the bounded disturbances were not fully addressed in [103, 115] while model uncertainties and practical scenarios were not adequately detailed in [116]. For this reason, the control designs for offshore crane systems remains an open problem. The previous study on offshore crane models and control methods are summarised in Table 2.1.

2.3 Sliding mode control

Research on variable structure control (VSC) systems has progressed from the pioneering work in Russia of Emel'yanov and Barbashin in the early 1960s. The idea of VSC only wide-spread outside of Russia during mid 1970s when a book by Itkis in 1976 and a survey paper by Utkin [137] in 1977 were published in English. VSC concepts have subsequently been utilised in the design of robust regulators, model-reference systems, adaptive schemes, tracking systems, state observers and fault detection schemes. The ideas have successfully been applied to problems as diverse as automatic flight control, control of electric motors, chemical processes, helicopter stability augmentation systems, space systems and robots [44]. One of the earliest survey paper on VSC was written by Hung et al. [69] that provides many references to the application of sliding mode ideas in various engineering problems. Years later, the generalisation of VSC for a class of uncertain systems have been developed, for example, in [29, 45]. The VSC approaches have been further distinguished to two types of controls, i.e., (i) VSCs that switch between different parameters and (ii) systematic further development of the methods which is known as soft variable structure controls (soft VSC) that continuously vary controllers' parameters or structures and achieve nearly time-optimal control performance. A survey paper on soft VSC has been reported in [3].

Sliding mode control (SMC) belonging to the VSC systems became popular because of its application to a broad class of systems containing discontinuous control

Chapter 1

Introduction

1.1 Background

Cranes are used for transportation of heavy loads such as containers and construction materials on land as well as at open sea. They are widely used in construction sites, warehouses and harbours (Figure 1.1) due to their ability to handle hefty objects.

In general, cranes can be regarded as underactuated Lagrangian system. Firstly, crane systems are underactuated because they have fewer independent control actuators than degrees of freedom (DOF) to be controlled. For example, in a 2-D gantry crane system, both cart position on the girder and payload sway are controlled by a single motor. Secondly, cranes are classified as Lagrangian systems because their equations of motion can be obtained based on the formulation of Lagrangian mechanics like robotic manipulators. Basically, a crane system consists of a support mechanism, which is a part of its structure, and a hoisting mechanism. The hoisting mechanism of a crane often exhibits an oscillatory behaviour due to the underactuation of the system. For this reason, it is important for a crane operation to meet stringent safety requirement.

A large number of studies on the development of control strategies to improve the efficiency and safety of crane operations has been seen over the past few decades. The hoisting mechanism that typically consists of cable, hook and payload assembly has high compliance. Hence, certain exogenous excitations at the suspension point can produce high amplitude of oscillations to the payload. The inertia forces due to

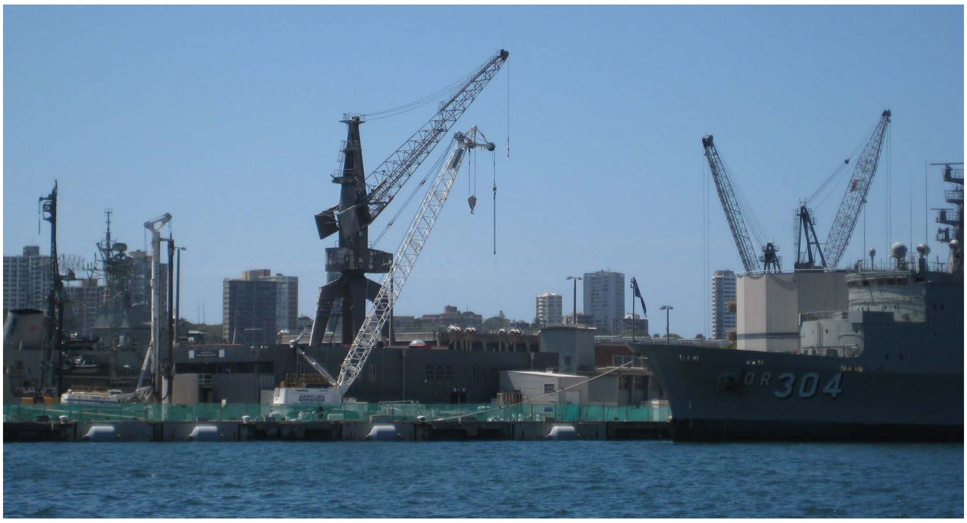

Figure 1.1 : Panoramic view of Sydney Harbour.

the motion of the crane can induce significant payload pendulations as well. This problem occurs because cranes are typically lightly damped. In other words, any transient oscillation response in crane systems takes a long time to dampen out.

As the research on conventional cranes becomes well established, researchers have explored a more complex problem, namely the offshore cranes. The growing usage of ocean facilities in many segments such as shipping of containers, oil and gas exploration, and offshore wind farm construction have necessitated certain operations occur in open sea conditions. These activities require the application of offshore cranes to transfer loads between vessels or to place loads from a vessel to an offshore site.

In general, offshore cranes operations can be categorised into two types, namely the stevedoring and the moonpool operations. Offshore stevedoring or lightering is the process of transferring containers between vessels, and moonpool operation is the activity of a payload placement underwater or on the seabed for the purpose of underwater installation. The advantage of offshore stevedoring operation is it can avoid marine traffic congestion in a port. The transfer of containers between two vessels requires a crane equipment on one of the vessel (Figure 1.2) or a third vessel (Figure 1.3). The common types of offshore cranes used in this operation is gantry crane (Figure 1.4) and boom crane (Figure 1.5).

Port congestion has become a major issue over the last few years due to rapid developments of logistics industry causing a substantial increase in the trading volume [50, 74, 144]. Some ambitious plans of port expansion have been proposed to overcome this problem, but it is not a feasible solution due to land constraints. Consequently, a new method of transportation, namely, the ship-to-ship cargo transfer operation, is introduced [81]. This method, emerging to become a promising solution to improve ports' efficiency and productivity and reduce operational costs, could enable the ports to stay competitive.

Despite all the necessities and benefits of offshore transportation and installation, the presences of persistent disturbances in the crane operations due to harsh sea condition are inevitable. Ocean waves can induce motions to vessels or ships where cranes are located. These motions include roll, pitch, yaw, surge, sway and heave. Besides, wind drag or buoyancy of seawater can produce exogenous forces on the payload, whenever it is suspended in the air or submerged. For this reason, it is necessary to have an element of robustness in the offshore cranes control system to deal with the aforementioned disturbances.

In particular, motivated by a large amount of significant practical problems, the control of underactuated nonlinear systems has become an important subject of research. Intuitively, the control synthesis for underactuated systems is more complex than that for fully actuated systems. Control of underactuated systems is currently an active field of research due to their broad applications in robotics, land and aerospace vehicles, surface vessels and crane automation. Based on recent surveys, control of general underactuated systems is a major open problem. Since the

Chapter 3

Modelling of Offshore Crane Systems

3.1 Introduction

The chapter begins with the generalisation of cranes dynamics by using the Lagrangian mechanics as the preliminary to the model derivations. From the Euler-Lagrange formulation, the dynamic models of offshore gantry crane and boom crane are derived by considering the vessels' motion. For each crane types, 2-D and 3-D models are developed with full system dynamics with respect to system dimensions. To facilitate the first-order sliding mode control designs in the latter chapter, we provide the linearised forms of 2-D offshore crane models.

3.2 Euler-Lagrange equation for cranes

In this section, we provide the generalisation of offshore crane dynamics based on Lagrangian mechanics. The offshore crane models are derived based on the following assumptions:

i. The payload is considered as a point mass.

ii. The crane's support mechanism (girder or boom) has even mass distribution.

iii. The rope or cable is massless and there always exists strain in the rope so that the rope will not bend under the motion of vessel or crane.

Consider a crane system consisting of r links and suppose the mass of link k is m_k. Let center of mass m_k has position vector $p_k \in \mathbb{R}^3$. Thus, the kinetic energy of the system is

$$\mathcal{K} = \frac{1}{2}\sum_{k=1}^{r} m_k \|\dot{p}_k\|^2$$

and the potential energy of the system is

$$\mathcal{P} = \sum_{k=1}^{r} m_k [0 \;\; 0 \;\; g] p_k$$

where g is the gravitational acceleration. It follows that, the Lagrangian of the system can be obtained as

$$\mathcal{L} = \mathcal{K} - \mathcal{P}.$$

Let $q \in \mathbb{R}^n$ be the vector of generalised coordinates and $\tau \in \mathbb{R}^n$ be the corresponding generalised forces. By applying the Euler-Lagrange formulation

$$\frac{d}{dt}\left(\frac{\partial \mathcal{L}}{\partial \dot{q}}\right) - \frac{\partial \mathcal{L}}{\partial q} = \tau, \tag{3.1}$$

the equation of motion of the system can be expressed in the following form:

$$M(q)\ddot{q} + C(q,\dot{q})\dot{q} + G(q) + d(t) = \tau, \tag{3.2}$$

where $M(q) \in \mathbb{R}^{n\times n}$ is the inertia matrix, $C(q,\dot{q}) \in \mathbb{R}^{n\times n}$ is the centrifugal-Coriolis matrix and $G(q) \in \mathbb{R}^n$ is the vector of gravity. Vector $d(t) \in \mathbb{R}^n$ may consist of frictions, uncertainty and disturbance terms. For simplicity, (3.2) can also be written as

$$M(q)\ddot{q} + f(q,\dot{q}) = \tau, \tag{3.3}$$

where $f(q,\dot{q}) = C(q,\dot{q})\dot{q} + G(q) + d(t)$.

Now consider an underactuated mechanical system with m number of inputs such that $1 \leq m < n$. By partitioning vector of generalised coordinates as $q = [q_a^T \;\; q_u^T]^T$ and vector of generalised forces as $\tau = [\tau_a^T \;\; 0_{1\times(n-m)}]^T$, (3.3) can be expressed in the following form:

$$\begin{bmatrix} M_{aa}(q) & M_{au}(q) \\ M_{au}^T(q) & M_{uu}(q) \end{bmatrix} \begin{bmatrix} \ddot{q}_a \\ \ddot{q}_u \end{bmatrix} + \begin{bmatrix} f_a(q,\dot{q}) \\ f_u(q,\dot{q}) \end{bmatrix} = \begin{bmatrix} \tau_a \\ 0_{(n-m)\times 1} \end{bmatrix}, \tag{3.4}$$

where $q_a \in \mathbb{R}^m$, $q_u \in \mathbb{R}^{n-m}$, $\tau_a \in \mathbb{R}^m$, $M_{aa}(q) \in \mathbb{R}^{m\times m}$, $M_{au}(q) \in \mathbb{R}^{m\times(n-m)}$, $M_{uu}(q) \in \mathbb{R}^{(n-m)\times(n-m)}$, $f_a(q,\dot{q}) \in \mathbb{R}^m$, and $f_u(q,\dot{q}) \in \mathbb{R}^{n-m}$. It follows from the second row of (3.4) that

$$\ddot{q}_u = M_{uu}^{-1}(q)[-M_{au}^T(q)\ddot{q}_a - f_u(q,\dot{q})]. \tag{3.5}$$

Substituting (3.5) into the first row of (3.4) yields

$$\ddot{q}_a = (M_{aa}(q) - M_{au}(q)M_{uu}^{-1}(q)M_{au}^T(q))^{-1} \times \big[-f_a(q,\dot{q}) + M_{au}(q)M_{uu}^{-1}(q)(f_u(q,\dot{q})) + \tau_a\big]. \tag{3.6}$$

3.3 Modelling of offshore gantry cranes

In this section, we present the models of two-dimensional (2-D) and three-dimensional (3-D) offshore gantry cranes. The 2-D model is presented in the form of uncertain LTI system, and the 3-D model is presented as the extended model of [103] with full DOF in the crane coordinates.

3.3.1 2-D model

The offshore crane system considered in this study consists of a gantry crane mounted on a ship vessel as visualize in Figure 3.1, where $\{O_G x_G y_G z_G\}$, $\{O_B x_B y_B z_B\}$ and $\{O_N x_N y_N z_N\}$ are the coordinate frames of the ground, the container ship, and the cart's starting point, respectively. The offshore crane system motion is represented by three generalized coordinates, i.e., the position of the cart, y, the length of the rope measured from the cart to the payload, l, and the sway angle induced by the motion of the cart, θ. Let h_t denote the vertical position of the cart from O_B, and d_y denote the distance of the cart's starting point from z_B-axis. The masses of the cart and payload are denoted by m_c and m_p, respectively. Let $\zeta(t)$ be the heaving and $\phi(t)$ be the rolling angular displacement of the vessel. Thus, the position vectors of the cart and the payload with respect to the ground coordinate