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Impedance control of secondary regulated hydraulic crane in the water entry phase[☆]

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ABSTRACT

During the water entry phase in heavy lift marine operations, the involved crane system must satisfy stringent safety requirements. A hydraulic crane driven by a secondary controlled unit that has the advantage of high dynamic behavior and energy efficiency is investigated in this paper. The detailed models of the crane system and the offshore operating conditions are established. To guarantee functionality of the crane system under harsh sea conditions, a cascade control structure is constructed. In the outer loop, an extended state observer is used to estimate the unmeasured payload states. Then, an impedance controller is designed to realize trajectory tracking and impedance regulation of the payload. In the inner loop, the reference trajectory is provided by the impedance controller. An indirect adaptive robust control approach is applied to maximize the achievable tracking performance and to obtain accurate parameter estimates of the secondary controlled unit in the presence of parametric uncertainties and unknown disturbances. The proposed controller can lower a payload to a desired position, and keep the values of the hydrodynamic forces and wire tension within acceptable bounds at the same time. Comparative simulation results verify the effectiveness of the proposed control method.

1. Introduction

Motivated by the growing demands of oil extraction and subsea exploration, subsea lift and installations of process modules have been one of the major concerns in marine operations. These tasks can be accomplished by using an actively controlled crane placed on a vessel. According to Det Norske Veritas (2011), a typical subsea lift consists of the following main phases: lifting off from deck, lowering through the splash zone, lowering down to seabed and seabed landing. Amongst the aforementioned phases, the most critical phase is the so-called water entry phase during which the payload transits from air to water. In this phase, the payload is subject to potentially large hydrodynamic forces which can cause damage to the load or can lead to a wire break. The possibility of losing or damaging the payload will cause a costly stop of the operations and, most importantly, impair the safety of the operators on board.

Primary actuator of most offshore crane systems is driven by either hydraulic or electric systems. Electric systems have increased in popularity due to their relatively high efficiency (Woodacre et al., 2015). However, high power electric motors tend to be physically large with a correspondingly large moment of inertia, especially in the conditions of heavy lift operations. Compared with electric actuators, hydraulic motors have the advantage of high energy density and small footprint, which is appealing when deck space is limited. Nevertheless, traditional hydraulic drive systems (e.g., valve control and pump control systems) suffer from various shortcomings in marine crane systems. For valve control systems, oil flow through a valve orifice leads to large energy loss. By contrast, pump-control systems view a better energy efficiency with no throttling losses, but the increase of the hydraulic line results in relatively long hydraulic time constant and slow dynamic response. To overcome the drawback of the traditional hydraulic systems, a novel hydrostatic drive technology referred to as "secondary controlled hydraulic drive" (SCHD) have been applied to marine crane systems recently (Dabing et al., 2011; Sun et al., 2005). The idea was originally patented by Nikolaus (1977). In the SCHD system, the constant pressure variable pump is a first-regulated device and the secondary controlled unit (SCU) is a second-regulated device. The constant pressure variable pump is generally adopted with the accumulator to produce a constant pressure and delivering hydraulic oil according to the load requirement. On the constant pressure condition, the hydraulic time constant of the

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system have no influence on the dynamic behavior of the SCU. Besides, the SCU can operate as a pump or a motor, which provides unrestricted four-quadrant operation with the possibility of energy reuse. The energy efficiency of SCHD has been evaluated in Shimoyama et al. (2004) and Shen et al. (2017), where the energy consumption was reduced by 20–50% compared with the traditional hydraulic drive systems.

Several control strategies, e.g., robust control (Berg and Ivantysynova, 1999), bilinear control (Guo et al., 1994) and adaptive fuzzy sliding mode control (Ho and Ahn, 2012) had been utilized to regulate the SCHD system. However, these strategies were based on accurate mathematical models and sensitive to parametric uncertainties. Direct adaptive robust control (DARC) based on the idea of "integrator backstepping" has been developed to attenuate the effect of parametric uncertainties (Guan and Pan, 2008; Polycarpou and Ioannou, 1996; Yao et al., 2000). Although the satisfactory tracking performance has been achieved, the parameters of such a DARC hardly converge to their true values in implementation (Yao and Palmer, 2002). To overcome the defects of DARC design, indirect adaptive robust control (IARC) has recently been proposed (Mohanty and Yao, 2011; Yao and Palmer, 2002), in which the controller and the identifier can be designed separately. The modular design method makes it possible to use a more accurate parameter estimation algorithm for monitoring the health of the SCU. However, these methods need a prior knowledge on the extent of unknown parameters, which may not be realized in the SCU system. A novel estimation law (Wu et al., 2016) based on modified least-squares (LS) algorithm can estimate the parameters without any prior knowledge on bounds of the unknown parameter and disturbances. Nevertheless, this algorithm cannot be applied to closed-loop systems, where the driving signals cannot always be guaranteed to satisfy persistently exciting (PE) condition (Ioannou and Sun, 1996). Besides, the standard backstepping technique may suffer from the problem of 'explosion of terms'. This is due to the requirement of analytic calculation of the derivatives of virtual control functions. It is well documented that, as the order of a system increases, analytic calculation of these derivatives becomes prohibitive (Dong et al., 2012; Swaroop et al., 2000). To overcome this obstacle, dynamic surface control and command filtered backstepping (Dong et al., 2012; Swaroop et al., 2000) were utilized in backstepping design procedure. These approaches use filtering methods to produce certain command signals and their derivatives, which eliminates the requirement of analytic differentiation.

To improve the operability in harsh seas conditions, several heave compensation methods has been proposed recently (Do and Pan, 2008; Küchler et al., 2011; Woodacre et al., 2015). Although these methods can decouple the payload motion from the vessel motion, the influence of the waves is still a big threat in the water entry phase. Thus, a control strategy referred to as "wave synchronization" was proposed to reduce the hydrodynamic forces by minimizing variations of the relative vertical velocity between the payload and the waves. This control method can be realized by either wave amplitude feed-forward control (Johansen et al., 2004) or feedback control (Messineo et al., 2008; Messineo and Serrani, 2009). However, there existed a significant hurdle in wave synchronization since the waves in open sea is stochastic and is difficult to measure. Alternatively, a parallel force/position approach was proposed by Skaare and Egeland (2006) to deal with the water entry problem. The effectiveness of the scheme is ensured by the dominance of the force control loop over the position control loop. For given force and position set points, the force error is asymptotically driven to zero at the expense of a steady-state position error (Chiaverini et al., 1999). Hence, the parallel force/position approach cannot lower a payload to a desired position accurately. Sagatun (2002) proposed a control strategy based on impedance control, which provided a unified framework for both trajectory tracking and regulation of impedance of the payload. When the crane system is compliant to fast hydrodynamic forces, significant improvements in key criteria (e.g., maximum value of hydrodynamic forces and minimum of the wire tension) can be achieved. However, an important limitation of the proposed controller (Sagatun, 2002) is that this scheme neglected the actuator dynamics.

In this paper, a secondary regulated crane is utilized to conduct the lift operation in the water entry phase. Based on the detailed models of the secondary controlled crane system and the offshore operating conditions, a cascade control structure is constructed to ensure tracking performance of the payload and to improve the operability of the crane system. In the outer control loop, an extended state observer is employed to estimate the unmeasured states of the payload. These estimated states are used to develop an impedance controller. In the inner loop, adaptive backstepping designs are utilized to synthesize IARC that achieve not only good output tracking performance but also better parameter estimation processes to obtain accurate parameter estimates. A novel estimation law which combines a modified LS algorithm (Wu et al., 2016) with a gradient algorithm based on fixed σ-modification (Ioannou and Sun, 1996) is proposed to realize parameter estimation without any requirement of the driving signals' PE condition. Furthermore, by employing command filtered adaptive backstepping technique, the proposed schemes can overcome the problem of "explosion of terms" existing in the standard backstepping technique. The remainder of this paper is organized as follows: the system models are described in Section 2. Section 3 develops the control system. Simulation results of the proposed controller and some comparisons with the parallel force position controller are presented and discussed in Section 4. Section 5 provides a brief conclusion of the work.

2. System modeling

The schematic representation of the crane system is illustrated in Fig. 1. A constant pressure variable pump is adopted with the accumulator to produce a constant pressure. On the constant pressure condition, the output torque of the SCU can be controlled by adjusting a displacement control mechanism (DCM) composed of swash plate positioning cylinder, electro-hydraulic servo valve and cylinder position sensor. The SCU drives the winch drum via the gear box to haul in or let out the wire rope, the wire tension is measured by a load pin mounted on the sheave shaft.

2.1. Secondary controlled unit

According to Kim and Lee (1996), Berg and Ivantysynova (1999), the DCM satisfies the following properties: 1) The natural frequency of the servo valve is well above that of the displacement adjusting mechanism of positioning cylinder; 2) The total oil volume of both side chambers is small; 3) The mass of the regulating piston and the viscous friction force are small. Thus, the model of DCM can be simplified to a first-order system

$$\dot{l}(t) = (-l(t) + k_l u(t))/\tau, \tag{1}$$

where $l(t)$ is the cylinder position, τ is the time constant, $u(t)$ is the electric signal control input, k_l is the direct current gain of the DCM. To simplify the notation, from now on the explicit dependence on time t may be sometimes omitted (that is, let $l = l(t)$, $u = u(t)$ and so on). The motion of SCU can be described by the following differential equation

$$J_e\ddot{\varphi} + B_e\dot{\varphi} = \frac{P_s V_{\max}}{l_{\max}} l - \delta - F_t r/i, \tag{2}$$

where φ denotes the angle displacement of SCU, which can be measured by an encoder mounted on the output shaft. J_e is the equivalent moment of inertia and B_e is the damping coefficient converted to the output shaft of the SCU, which can be calculated by

$$J_e = J_m + J_g + J_d r^2/i^2,$$

$$B_e = B_m + B_g + B_d r^2/i^2, \tag{3}$$

Where r is the radius of the winch drum, i is the reduction ratio of the